KARPAGAM ACADEMY OF HIGHER EDUCATION
Faculty of Engineering
Department of Mechanical Engineering

SYLLABUS

16BEME602 DESIGN OF TRANSMISSION SYSTEMS 3 0 0 3 100

OBJECTIVES

1. To gain knowledge on the principles and procedure for the design of Mechanical power transmission components.
2. To understand the standard procedure available for Design of Transmission of Mechanical elements
3. To learn to use standard data and catalogues

UNIT I DESIGN OF TRANSMISSION SYSTEMS FOR FLEXIBLE ELEMENTS 9

Design of V belts and pulleys – Selection of Flat belts and pulleys – Wire ropes and pulleys – Selection of Transmission chains and Sprockets – Design of sprockets.

UNIT II DESIGN OF SPUR AND HELICAL GEARS 9

Gear Terminology – Speed ratios and number of teeth–Force analysis – Tooth stresses – Dynamic effects – Fatigue strength – Factor of safety – Gear materials – Module and Face width–power rating calculations based on strength and wear considerations – Parallel axis Helical Gears – Pressure angle in the normal and transverse plane– Equivalent number of teeth–forces and stresses – Estimating the size of the helical gears.

UNIT III DESIGN OF BEVEL AND WORM 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 DESIGN OF GEAR BOXES 9

Geometric progression – Standard step ratio – Ray diagram, kinematics layout –Design of sliding mesh gear box –Constant mesh gear box. – Design of multi speed gear box.

UNIT V DESIGN OF CLUTCHES AND BRAKES 9

Design of plate clutches –axial clutches–cone clutches–internal expanding rim clutches–internal and external shoe brakes.

TOTAL 45

(Permitted to use PSG design data book in the examination)

TEXT BOOKS

S. No.	Author(s) Name	Title of the book	Publisher	Year of Publication
1	Robert C. Juvinall, Kurt M. Marshek	Fundamentals of Machine Component Design	John Wiley and Sons., London	2017
2	Bhandari V B	Design of Machine Elements	Tata McGraw Hill	2016

REFERENCES

S. No.	Author(s) Name	Title of the book	Publisher	Year of Publication

1	Maitra G.M., Prasad L.V	Hand book of Mechanical Design	Tata McGraw–Hill, New Delhi	2009
2	Shigley J.E, Mischke C.R	Shigley's Mechanical Engineering Design 10e	McGraw–Hill International Editions, New Delhi	2015
3	Gope P C	Machine Design : Fundamentals And Applications	PHI learning, India	2012

WEB REFERENCES

1. http://en.wikipedia.org/wiki/Gear
2. http://www.physicsforums.com/showthread.php?t=292163
3. http://www.seminarprojects.com/Thread–design–and–fabrication–of–gearbox–full–report
4. http://www.cs.cmu.edu/~rapidproto/mechanisms/chpt6.htm

KARPAGAM ACADEMY OF HIGHER EDUCATION

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FACULTY OF ENGINEERING

DEPARTMENT OF MECHANICAL ENGINEERING

LESSON PLAN

Subject Name : **Design of Transmission Systems**

Subject Code : **16BEME602** **(Credits - 3)**

Name of the Faculty : **S. ARAVIND**

Designation : **Assistant Professor**

Year/Semester : **III / VI**

Branch : **Mechanical Engineering**

UNIT – I: DESIGN OF TRANSMISSION SYSTEMS FOR FLEXIBLE ELEMENTS

Sl. No.	No. of Periods	Topics to be Covered	Support Materials
1.	1	Introduction to Transmission systems, Design of V belts and pulleys	T [1] R [2]
2.	1	Problems from Design of V belts and pulleys	T [2]
3.	1	Selection of Flat belts and pulleys	T [1] R [2]
4.	1	Problems from Design of Flat belts and pulleys	T [2]
5.	1	Tutorial I - Problems from Design of Flat belts and V-belts	T [2]
6.	1	Design of Wire ropes and pulleys	T [1] W [1]
7.	1	Problems from Design of Wire ropes and pulleys	T [2]
8.	1	Selection of Transmission chains and Sprockets, Design of sprockets.	T [1] R [2]
9.	1	Tutorial II - Problems from Design of Wire ropes and Chains	T [2]
10.	1	Discussion on Competitive Examination Related Questions / University previous year questions	
		Total No. of Hours Planned for Unit - I	**10**

UNIT – II: DESIGN OF SPUR AND HELICAL GEARS

Sl. No.	No. of Periods	Topics to be Covered	Support Materials
11.	1	Gear Terminology – Speed ratios and number of teeth–Force analysis – Tooth stresses	T [1] R [1]

12.	1	Dynamic effects – Fatigue strength – Factor of safety – Gear materials	T [1] R [1]
13.	1	Module and Face width–power rating calculations based on strength and wear considerations	T [1] R [1]
14.	1	Problems from Design of Spur gears	T [2]
15.	1	Tutorial III - Problems from Design of Spur gears	T [2]
16.	1	Parallel axis Helical Gears – Pressure angle in the normal and transverse plane	T [1] W [2]
17.	1	Equivalent number of teeth–forces and stresses – Estimating the size of the helical gears	T [1] R [1]
18.	1	Problems from design of helical gears	T [2]
19.	1	Tutorial IV - Problems from design of helical gears	T [2]
20.	1	Discussion on Competitive Examination Related Questions / University previous year questions	
		Total No. of Hours Planned for Unit - II	**10**

UNIT – III: DESIGN OF BEVEL AND WORM GEARS

Sl. No.	No. of Periods	Topics to be Covered	Support Materials
21.	1	Straight bevel gear: Tooth terminology, tooth forces and stresses, equivalent number of teeth.	T [1] R [1]
22.	1	Estimating the dimensions of pair of straight bevel gears.	T [1] R [1]
23.	1	Problems from design of bevel gears	T [2]
24.	1	Tutorial V - Problems from design of bevel gears	T [2]
25.	1	Worm Gear: Merits and demerits– terminology – Thermal capacity, materials–forces and stresses, efficiency	T [1] W [3]
26.	1	Estimating the size of the worm gear pair	T [1] R [1]
27.	1	Problems from design of worm gear pair	T [2]
28.	1	Cross helical: Terminology–helix angles–Estimating the size of the pair of cross helical gears.	T [1] R [1]
29.	1	Tutorial VI – Problems from design of worm gear pair and cross helical gear	T [2]
30.	1	Discussion on Competitive Examination Related Questions / University previous year questions	
		Total No. of Hours Planned for Unit - III	**10**

UNIT – IV: DESIGN OF GEAR BOXES

Sl. No.	No. of Periods	Topics to be Covered	Support Materials
31.	1	Geometric progression – Standard step ratio	T [1] W [4]
32.	1	Ray diagram, kinematics layout	T [1]
33.	1	Problems from Ray diagram, kinematics layout	R [3]
34.	1	Problems from Ray diagram, kinematics layout	R [3]
35.	1	Tutorial VII - Problems from Ray diagram, kinematics layout	R [3]
36.	1	Design of sliding mesh gear box –Constant mesh gear box. – Design of multi speed gear box.	T [1]
37.	1	Problems from design of gear box	R [3]
38.	1	Problems from design of gear box	R [3]
39.	1	Tutorial VIII - Problems from design of gear box	R [3]
40.	1	Discussion on Competitive Examination Related Questions / University previous year questions	
Total No. of Hours Planned for Unit - IV			**10**

UNIT – V: DESIGN OF CLUTCHES AND BRAKES

Sl. No.	No. of Periods	Topics to be Covered	Support Materials
41.	1	Design of plate clutches –axial clutches	T [1] R [2]
42.	1	Cone clutches–internal expanding rim clutches	T [1] W [5]
43.	1	Problems from design of clutches	T [2]
44.	1	Problems from design of clutches	T [2]
45.	1	Tutorial IX - Problems from design of clutches	T [2]
46.	1	Design of internal and external shoe brakes.	T [1] R [2]
47.	1	Problems from design of internal shoe brakes	T [2]
48.	1	Problems from design of external shoe brakes	T [2]
49.	1	Tutorial X- Problems from design of internal & External shoe brakes	T [2]
50.	1	Discussion on Competitive Examination Related Questions / University previous year questions	
Total No. of Hours Planned for Unit - V			**10**

TOTAL PERIODS : 50

TEXT BOOKS

S. No.	Author(s) Name	Title of the book	Publisher	Year of Publication
T [1]	Robert C. Juvinall, Kurt M. Marshek	Fundamentals of Machine Component Design	John Wiley and Sons., London	2017
T [2]	Bhandari V B	Design of Machine Elements	Tata McGraw Hill	2016

REFERENCES

S. No.	Author(s) Name	Title of the book	Publisher	Year of Publication
R [1]	Maitra G.M., Prasad L.V	Hand book of Mechanical Design	Tata McGraw–Hill, New Delhi	2009
R [2]	Shigley J.E, Mischke C. R	Shigley's Mechanical Engineering Design 10e	McGraw–Hill International Editions, New Delhi	2015
R [3]	Gope P C	Machine Design: Fundamentals and Applications	PHI learning, India	2012

WEBSITES

W [1] - www.usbr.gov/ssle/safety/RSHS/appD.pdf
W [2] - www.slideshare.net/AMIR92671/design-of-helical-gear-box
W [3] - www.scribd.com/document/122664213/worm-gear-pdf
W [4] - www.slideshare.net/Abhi23396/design-of-gear-box-66549551
W [5] - www.scribd.com/document/288456986/Cone-Clutch

JOURNALS

J [1] - Hong, Tran Thi, et al. "Calculating Optimum Gear Ratios of Mechanical Driven Systems Using Worm-Helical Gearbox and Chain Drive." International Journal of Applied Engineering Research 14.14 (2019): 3211-3218.
J [2] - Singh, Raju, and Mousam Sharma. "Design and Optimization of tooth Profile Modification of Spur Gear by FEA Approach." Journal of Automobile Engineering and Applications 5.2 (2018): 15-24.
J [3] - Rai, Paridhi, and Asim Gopal Barman. "Design of bevel gears using accelerated particle swarm optimization technique." IOP Conference Series: Materials Science and Engineering. Vol. 361. No. 1. IOP Publishing, 2018.
J [4] - Gunti, Karan, et al. "Innovative Layout of Gears for the Optimum Design of Gearbox of a Levelling Machine." International Conference on Advances in Thermal Systems, Materials and Design Engineering (ATSMDE2017). 2017.
J [5] - Srinath, Nitin, et al. "Smart Clutch: An Automated Clutch That Can Be Used in Manual Transmission Driven Vehicles." ASME 2016 International Mechanical Engineering Congress and Exposition. American Society of Mechanical Engineers Digital Collection, 2016.

UNIT	Total No. of Periods Planned	Lecture Periods	Tutorial Periods
I	10	8	2
II	10	8	2
III	10	8	2
IV	10	8	2
V	10	8	2
TOTAL	50	40	10

I. CONTINUOUS INTERNAL ASSESSMENT : 40 Marks

(Internal Assessment Tests: 25, Attendance: 5, Assignment:5, Seminar: 5)

II. END SEMESTER EXAMINATION : 60 Marks
TOTAL : 100 Marks

UNIT I

DESIGN OF FLEXIBLE DRIVE SYSTEMS

Design of V belts and pulleys – Selection of Flat belts and pulleys – Wire ropes and pulleys – Selection of Transmission chains and Sprockets – Design of sprockets.

Belt Drive

Belt drive is a mechanical drive made up of flexible material used to transmit power from one shaft (driving shaft) to another shaft (driven shaft) which are parallel to each other and run at same (or) different speeds. The selection of belt drive depends on some important factors which include, the speed of driving and driven shaft, power transmitted, speed reduction ratio, centre distance between the two shafts, space available and so on.

Types of Belt drives

The belt drives are classified based on their specific applications. They are

(a) Light duty belt drives

These are used to transmit less power (approx 5 kW) and at belt speeds upto 10 m/s. The main applications of these type of drives are in agricultural purposes (pumps, blowers, fans, etc.,)

(b) Medium duty belt drives

These type of belt drives are used to transmit medium powers (approx. 5 kW to 20 kW) and speed varies from 10 m/s to 20 m/s. The main applications of these type of belt drives include, machine tools, generators, etc.,

(c) Heavy duty belt drives

These types of belt drives are used for transmitting heavy power (ie) above 20 kW. The main applications of these type of belt drives include, crushers, bucket elevators, marine engines, etc.

Types of Belts

The belts can be classified based on cross-section and represented in Fig. They are (i) Flat belts (ii) V – belts (iii) Circular belts (iv) Toothed belts

i) Flat belts

The Flat belt as shown in Fig. (a) is mostly used in farming, mining and logging applications. The Flat belt is a simple system of power transmission. It can deliver high power at high speeds (370 kw at 50 m/s).

(ii) V - belts

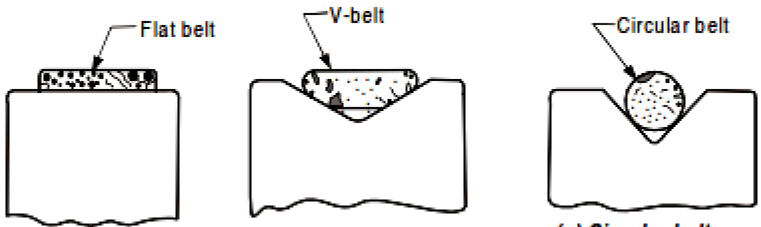
The V - belt as shown in Fig. (b) are generally endless and their cross-section shape is trapezoidal. The V - shape of belt tracks in a mating groove in the shaft, with the result that the belt cannot slip off.

(iii) Circular belts

The Circular belt as shown in Fig (c) are circular cross section belt to run in a pulley with a 60° V – groove round belts are used in case of low torque requirements.

(iv) Toothed belt

The toothed belt is made as a flexible belt with teeth moulded onto its inner surface. It runs over matching toothed pulleys (or) sprockets. Since toothed belts can also deliver more power that a friction drive belt, they are also used for high power transmissions. They include primary drive of some motor cycles.



(a) Flat belt (b) V-belt (c) Circular belt

Materials used for belts

The materials used for belts have to be strong, flexible and durable. They should have high coefficient of friction. They are cotton fabrics, leather, rubber, silk, etc.

Flat Belt Drives

Flat belt drives can be used for transmitting large amount of power and there is no upper limit of distance between the pulleys. These drives are efficient at high speeds and they offer noiseless running. Flat belts are available for a wide range of width, thickness, weight and material.

Advantages of Flat-belt drive

1. Different velocity ratios can be obtained by using a stepped cone pulley.

2. A belt drive can be used as a clutch, by shifting the belt from fast pulley to loose pulley.

3. Design of flat belt drive is simple.

4. Flat belt drive is relatively cheap and easy to maintain.

5. Flat belt drives are flexible, which gives protection.

6. Close casing is not required, like a gear box.

7. Flat belt drives can be used for long centre distances (upto 15 metres)

Disadvantages

1. Since velocity ratio is not constant, flat belt drive is not a positive drive

2. Flat belt drives have larger dimensions and occupy more space.

3. Flat belt drive is not suitable for smaller centre distance (less than 1 metre).

Design Based on Manufacturer's Data

Step 1: Diameter of Driver (or) Driven pulley:

The diameter of the pulley and angle of contact can be selected from the table given in data book pg.no. 7.52 using the given belt speed and assuming the number of piles , minimum pulley diameter chosen.

Step 2: Calculation of design power in kW

Calculate the design power in kW using the relationship given below

$$Design\ kW = \frac{rated\ kW \times load\ correction\ factor\ (K_s)}{Arc\ of\ conatct\ factor\ (K_\alpha) \times Small\ pulley\ diameter\ (K_d)}$$

The load correction factor is selected from the table given in data book pg. no. 7.53 and the arc of contact factor is selected from the table in data book pg. no. 7.54 for calculated actual arc of contact,

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

Small pulley factor is selected from the table given in data book pg.no 7.62.

Step 3: Selection of a belting

The load rating of fabric belts per mm width per ply at 180° arc of contact at 10 m/s belt speed from table from data book pg. no 7.54.

Step 4: Load rating correction

The load rating is corrected to actual speed of the belt using the relation given below (data book pg. no. 7.54)

$$load\ rating\ at\ V\ {}^{\mathrm{m}}/_{s} = \ load\ rating\ at\ 10\ {}^{m}/_{s} \times \frac{V}{10}$$

Step 5: Determination of belt width

$$Belt\ width = \frac{Design\ power}{Load\ rating \times No.\ of\ plies}$$

The calculated belt width is rounded off to the standard belt width from table given from data book pg. no. 7.52.

Step 6: determination of pulley width:

The pulley width is selected from the tables given in data book pg. no. 7.54 and 7.55.

Step 7: Calculation of belt length:

From data book pg. no. 7.53

For open belt drive

$$L = 2\,C + \left(\frac{\pi}{2}\right)\,(D + d) + \frac{(D - d)^2}{4C}$$

For cross belt drive

$$L = 2\,C + \left(\frac{\pi}{2}\right)\,(D + d) + \frac{(D + d)^2}{4C}$$

Problem:

Design a belt drive to transmit 20 kW at 720 rpm to an aluminium rolling machine, the speed ratio being 3. The distance between the pulleys is 3 m. Diameter of the rolling machine pulley is 1.2 m.

Solution:

Step 1: Diameter of Driver (or) Driven pulley:

The diameter of the pulley and angle of contact can be selected from the table given in data book pg.no. 7.52 using the given belt speed and assuming the number of piles , minimum pulley diameter chosen.

The driven pulley diameter, D = 1200 mm

$$W.K.T, the\ speed\ ratio, i = \frac{D}{d} = \frac{N_1}{N_2}$$

$$d = \frac{D}{i} = \frac{1200}{3} = 400\ mm$$

Step 2: Calculation of design power in kW

Calculate the design power in kW using the relationship given below

$$Design\ kW = \frac{rated\ kW \times load\ correction\ factor\ (K_s)}{Arc\ of\ conatct\ factor\ (K_\alpha) \times Small\ pulley\ diameter\ (K_d)}$$

The load correction factor is selected from the table given in data book pg. no. 7.53 and the arc of contact factor is selected from the table in data book pg. no. 7.54 for calculated actual arc of contact,

$$arc\ of\ contact = 180° - \left(\frac{D - d}{C}\right) \times 60° = 180° - \left(\frac{1200 - 400}{3000}\right) \times 60° = 164°$$

Small pulley factor is selected from the table given in data book pg.no 7.62.

$$k_d = 0.8$$

$$Design\ kW = \frac{20 \times 1.5}{1.06 \times 0.8} = 35.377\ kW$$

Step 3: Selection of a belting

The load rating of fabric belts per mm width per ply at 180° arc of contact at 10 m/s belt speed from table from data book pg. no 7.54.

FORT duck belting is selected, its capacity is around 0.0289 kW/mm/ply

Step 4: Load rating correction

The load rating is corrected to actual speed of the belt using the relation given below (data book pg. no. 7.54)

$$load\ rating\ at\ V\ ^{\mathrm{m}}/_{\mathrm{s}} = load\ rating\ at\ 10\ ^{m}/_{s} \times \frac{V}{10}$$

$$Velocity\ of\ the\ belt, V = \frac{\pi d\ N_1}{60} = \frac{\pi \times 0.4 \times 720}{60} = 15.08\ m/s$$

Therefore, load rating

$$load\ rating\ at\ V\ ^{\mathrm{m}}/_{\mathrm{s}} = 0.0289 \times \frac{15.08}{10} = 0.04358\ kW/mm/ply$$

Step 5: Determination of belt width

The number of plies is selected from table given in data book pg. no. 7.52.and it is found **to be 6**

$$Belt\ width = \frac{Design\ power}{Load\ rating \times No.of\ plies} = \frac{35.377}{0.04358 \times 6} = 135.29\ mm$$

The calculated belt width is rounded off to the standard belt width from table given from data book pg. no. 7.52.

$$Std\ width\ of\ the\ belt = 152\ mm$$

Step 6: determination of pulley width:

The pulley width is selected from the tables given in data book pg. no. 7.54 and 7.55.

$$Pulley\ width = belt\ width + 25\ mm = 152 + 25 = 177\ mm$$

Step 7: Calculation of belt length:

From data book pg. no. 7.53

For open belt drive

$$L = 2\,C + \left(\frac{\pi}{2}\right)(D + d) + \frac{(D - d)^2}{4C} = 8566.6\ mm$$

DESIGN OF V-BELT DRIVE

Introduction

V-Belt is a type of flexible connector used for transmitting power from one pulley to another pulley having a centre distance upto 3 metres. V-Belts are used with electric nylon to drive different equipments like blowers, compressor, machine tool, industries machinery, etc. The belts are operated on grooved pulley called sheaves. The sheaves have V – shaped grooves or two inclined sides with flat bottom. The belt makes contact with the sheaves on the sides and clearance at the bottom.

Usually V - Belts are endless ie., each belt is made in a circular form with various cross section which may be differentiated by different grades. It is made in trapezoidal section and the power is transmitted by the wedging action between the belt and the V -groove of the pulley or sheave. A properly installed V - belt should fit tightly against the sides of the pulley grooves without making any projection beyond the rim and should have efficient clearance bottom of the groove. The materials used for V-belts are cotton fabric and cards moulded in rubber and coined with fabric and rubber.

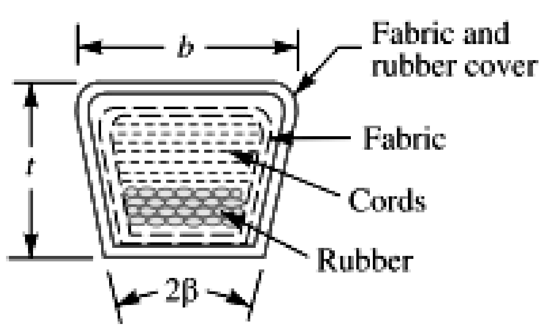
Advantages

1. High velocity ratio (upto 7 and in some cases 10 also)
2. Smaller centre distance.
3. Reliability of the drive, in any position; even with vertical shafts.
4. Replacement is easy, because V-belts are available in standard sizes.
5. Smooth operation.

Disadvantages

1. Design of V-belt drive is more complicated.
2. Cannot be used for larger centre distance.

The Cross-section of V-belt



Standard Pitch Lengths of V-belts

According to IS: 2494-1974, the V-belts are designated by its type and nominal inside length. For example, a V-belt of type A and inside length 914 mm is designated as A 914–IS: 2494. According to IS: 2494-1974, the pitch length is defined as the circumferential length of the belt at the pitch width (i.e. the width at the neutral axis) of the belt. The value of the pitch width remains constant for each type of belt irrespective of the groove angle. The pitch lengths are obtained by adding to inside length: 36 mm for type A, 43 mm for type B, 56 mm for type C, 79 mm for type D and 92 mm for type E. The following table shows the standard pitch lengths for the various types of belt.

Standard pitch lengths of V-belts according to IS: 2494-1974.

Type of belt	*Standard pitch lengths of V-belts in mm*
A	645, 696, 747, 823, 848, 925, 950, 1001, 1026, 1051, 1102 1128, 1204, 1255, 1331, 1433, 1458, 1509, 1560, 1636, 1661, 1687, 1763, 1814, 1941, 2017, 2068, 2093, 2195, 2322, 2474, 2703, 2880, 3084, 3287, 3693.
B	932, 1008, 1059, 1110, 1212, 1262, 1339, 1415, 1440, 1466, 1567, 1694, 1770, 1821, 1948, 2024, 2101, 2202, 2329, 2507, 2583, 2710, 2888, 3091, 3294, 3701, 4056, 4158, 4437, 4615, 4996, 5377.
C	1275, 1351, 1453, 1580, 1681, 1783, 1834, 1961, 2088, 2113, 2215, 2342, 2494, 2723, 2901, 3104, 3205, 3307, 3459, 3713, 4069, 4171, 4450, 4628, 5009, 5390, 6101, 6863, 7625, 8387, 9149.
D	3127, 3330, 3736, 4092, 4194, 4473, 4651, 5032, 5413, 6124, 6886, 7648, 8410, 9172, 9934, 10 696, 12 220, 13 744, 15 268, 16 792.
E	5426, 6137, 6899, 7661, 8423, 9185, 9947, 10 709, 12 233, 13 757, 15 283, 16 805.

Design procedure for V-belt drive

Step 1: Selection of belt section:

The belt section is selected from the table given in data book pg.no. 7.58 based on the power to be transmitted.

Step 2: Selection of pulley diameters

The small pulley diameter is selected from the table given in data book pg.no. 7.58. Then by using the speed ratio, the larger pulley diameter is calculated.

Step 3: Selection f center distance:

It is selected from the table given in data book pg.no. 7.61 based on speed ratio. The minimum and maximum center distanc are given by the relations

$$C_{mini} = 0.55\ (D + d) +\ T \quad and\ C_{maxi} = 2\ (D + d)$$

Step 4: Determination of nominal pitch length:

The length of the belt L which is also known as nominal inside length is determined using therelation

$$L = 2\,C + \left(\frac{\pi}{2}\right)\,(D + d) + \frac{(D-d)^2}{4\,C}$$

Then select the next standard pitch length from the table given in data book pg.no. 7.58.

Step 5: Selection of various modification factors:

In order to calculate the design power, the following modification factors have to be determined

(i) Length correction factor (F_c)

It is selected from the data book pg.no 7.58 to 7.60

(ii) Correction factor for arc of contact (F_d)

Based on the arc of contact of the smaller pulley, the correction factor is selected from data book pg.no. 7.68

$$Arc\ of\ contact = 180° - \left(\frac{D-d}{C}\right) \times 60°$$

(iii) service factor, F_a

It is selected from the table given in data book pg.no. 7.69

Step 6: Calculation of maximum power capacity:

The maximum power capacity in kW is calculated using the formula given in data book pg.no. 7.62.

The formula is selected based on the equivalent pitch diameter

$$d_e = d_p \times F_b$$

Where, d_p is the pitch diameter of the smaller pulley, and F_b small diameter factor from the table given in data book pg.no. 7.62 based on the speed ratio.

Step 7: determination of number of belts (n_b)

From the relation taken from data book pg.no. 7.70, the number of belts is calculated

$$n_b = \frac{P \times F_a}{kW \times F_c \times F_d}$$

Step 8: Calculation of actual center distance:

The actual center distance is calculated using the relation given in data book pg.no. 7.61

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

Where

$$A = \frac{L}{4} - \pi\left[\frac{D+d}{8}\right] \quad and\ B = \frac{(D-d)^2}{8}$$

Problems:

Design a V-belt drive to the following specifications:

Power to be transmitted	=	7.5 kW
Speed of the driving wheel	=	1440 rpm
Speed of the driven wheel	=	400 rpm
Diameter of driving wheel	=	300 mm
Centre distance	=	1000 mm
Service	=	16 hours/day

Solution:

Step 1: Selection of belt section:

The belt section is selected from the table given in data book pg.no. 7.58 , B section is selected for power rating of 7.5 kW.

Step 2: Selection of pulley diameters

The small pulley diameter is selected from the table given in data book pg.no. 7.58. The small pulley diameter given is 300 mm, but the preferred smaller pulley diameter is 315 mm.

$$Speed\ ratio, i = \frac{D}{d} = \frac{N_1}{N_2} = \frac{1140}{400} = 3.6$$

Then by using the speed ratio, the larger pulley diameter is calculated.

$$Larger\ pulley\ diameter, D = i \times d = 3.6 \times 315\ mm = 1134\ mm$$

From the table, preferred diameter is D = 1250 mm

Step 3: Selection of center distance:

$$C = 1000\ mm, given$$

Step 4: Determination of nominal pitch length:

The length of the belt L which is also known as nominal inside length is determined using the relation

$$L = 2\,C + \left(\frac{\pi}{2}\right)(D + d) + \frac{(D-d)^2}{4\,C} = 2 \times 1000 + \left(\frac{\pi}{2}\right)(1250 + 315) + \frac{(1250 - 315)^2}{4 \times 1000}$$

$$L = 4676.85\ mm$$

Then select the next standard pitch length from the table given in data book pg.no. 7.58.

$$L = 4996\ mm$$

Step 5: Selection of various modification factors:

In order to calculate the design power, the following modification factors have to be determined

(i) Length correction factor (F_c)

It is selected from the data book pg.no 7.58 to 7.60

$$F_c = 1.18$$

(ii) Correction factor for arc of contact (F_d)

Based on the arc of contact of the smaller pulley, the correction factor is selected from data book pg.no. 7.68

$$Arc\ of\ contact = 180° - \left(\frac{D-d}{C}\right) \times 60° = 180° - \left(\frac{1250-315}{1000}\right) \times 60° = 123.9°$$

$$F_d = 0.83$$

(iii) service factor, F_a

It is selected from the table given in data book pg.no. 7.69

$$F_a = 1.3$$

Step 6: Calculation of maximum power capacity:

The maximum power capacity in kW is calculated using the formula given in data book pg.no. 7.62.

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

Where, S – belt speed

$$S = \frac{\pi d\, N_1}{60} = \frac{\pi \times 0315 \times 1440}{60} = 23.75\ m/s$$

The formula is selected based on the equivalent pitch diameter

$$d_e = d_p \times F_b = 315 \times 1.14 = 359.1\ mm$$

Where, d_p is the pitch diameter of the smaller pulley, and F_b small diameter factor from the table given in data book pg.no. 7.62 based on the speed ratio of i = 3.6.

The maximum value of d_e in the formula is 175 mm

$$kW = \left(0.79 \times 23.75^{-0.09} - \frac{50.8}{175} - 1.32 \times 10^{-4} \times 23.75^2\right) 23.75$$

$$kW = 5.445\ kW$$

Step 7: determination of number of belts (n_b)

From the relation taken from data book pg.no. 7.70, the number of belts is calculated

$$n_b = \frac{P \times F_a}{kW \times F_c \times F_d} = \frac{7.5 \times 1.3}{5.445 \times 1.18 \times 0.83} = 1.828 \approx 2\ belts$$

Step 8: Calculation of actual center distance:

The actual center distance is calculated using the relation given in data book pg.no. 7.61

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

Where

$$A = \frac{L}{4} - \pi \left[\frac{D+d}{8}\right] = \frac{4996}{4} - \pi \left[\frac{1250 + 315}{8}\right] = 634.42$$

$$and\ B = \frac{(D-d)^2}{8} = \frac{(1250 - 315)^2}{8} = 109278$$

$$C_{actual} = 634.42 + \sqrt{634.42^2 - 109278} = 1175.92\ mm$$

DESIGN OF ROPE DRIVES

The rope drives are widely used where a large amount of power is to be transmitted, from one pulley to another, over a considerable distance. It may be noted that the use of flat belts is limited for the transmission of moderate power from one pulley to another when the two pulleys are not more than 8 metres apart. If large amounts of power are to be transmitted, by the flat belt, then it would result in excessive belt cross-section.

The ropes drives use the following two types of ropes:

1. Fibre ropes, and 2. Wire ropes.

The fibre ropes operate successfully when the pulleys are about 60 metres apart, while the wire ropes are used when the pulleys are upto 150 metres apart.

Fibre Ropes

The ropes for transmitting power are usually made from fibrous materials such as hemp, manila and cotton. Since the hemp and manila fibers are rough, therefore the ropes made from these fibers are not very flexible and possesses poor mechanical properties. The hemp ropes have less strength as compared to manila ropes. When the hemp and manila ropes are bent over the sheave, there is some sliding of the fibers, causing the rope to wear and chafe internally. In order to minimize this defect, the rope fibers are lubricated with a tar, tallow or graphite. The lubrication also makes the rope moisture proof. The hemp ropes are suitable only for hand operated hoisting machinery and as tie ropes for lifting tackle, hooks etc.

The cotton ropes are very soft and smooth. The lubrication of cotton ropes is not necessary. But if it is done, it reduces the external wear between the rope and the grooves of its sheaves. It may be noted that the manila ropes are more durable and stronger than cotton ropes. The cotton ropes are costlier than manila ropes.

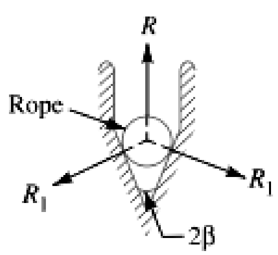
Advantages of Fibre Rope Drives

The fibre rope drives have the following advantages:

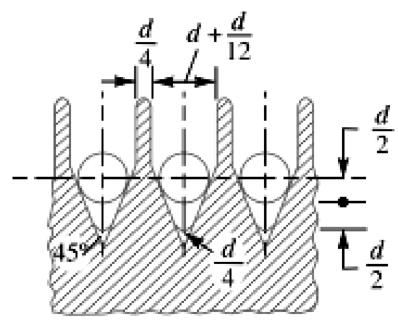
1. They give smooth, steady and quiet service.
2. They are little affected by outdoor conditions.
3. The shafts may be out of strict alignment.
4. The power may be taken off in any direction and in fractional parts of the whole amount.
5. They give high mechanical efficiency.

Sheave for Fibre Ropes

The fibre ropes are usually circular in cross-section as shown in Fig. The sheave for the fibre ropes is shown in Fig. The groove angle of the pulley for rope drives is usually 45°.



(a) Cross-section of a rope.



(b) Sheave (grooved pulley) for ropes.

The grooves in the pulleys are made narrow at the bottom and the rope is pinched between the edges of the V-groove to increase the holding power of the rope on the pulley. The grooves should be finished smooth to avoid chafing of the rope. The diameter of the sheaves should be large to reduce the wear on the rope due to internal friction and bending stresses. The proper size of sheave wheels is 40 d and the minimum size is 36 d, where d is the diameter of rope in cm.

When a large amount of power is to be transmitted over long distances from one pulley to another (i.e. when the pulleys are upto 150 metres apart), then wire ropes are used. The wire ropes are widely used in elevators, mine hoists, cranes, conveyors, hauling devices and suspension bridges.

The wire ropes run on grooved pulleys but they rest on the bottom of the *grooves and are not wedged between the sides of the grooves. The wire ropes are made from cold drawn wires in order to have increase in strength and durability. It may be noted that the strength of the wire rope increases as its size decreases. The various materials used for wire ropes in order of increasing strength are wrought iron, cast steel, extra strong cast steel, plough steel and alloy steel. For certain purposes, the wire ropes may also be made of copper, bronze, aluminium alloys and stainless steels.

Advantages of Wire Ropes

The wire ropes have the following advantages as compared to fibre ropes.

1. These are lighter in weight,
2. These offer silent operation,
3. These can withstand shock loads,
4. These are more reliable,
5. These are more durable,
6. They do not fail suddenly,
7. The efficiency is high, and
8. The cost is low.

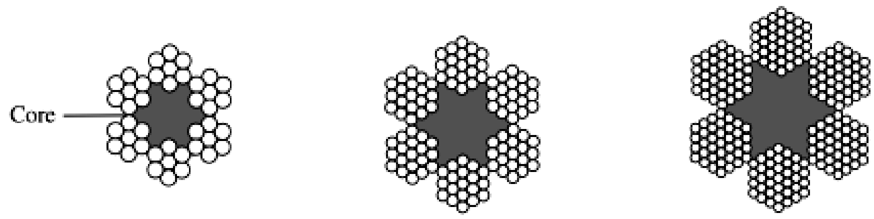
Construction of Wire Ropes

The wire ropes are made from various grades of steel wire having a tensile strength ranging from 1200 to 2400 MPa as shown in the following table :

Grade and tensile strength of wires

Grade of wire	120	140	160	180	200
Tensile strength range (MPa)	1200 – 1500	1400 – 1700	1600 – 1900	1800 – 2100	2000 – 2400

The wires are first given special heat treatment and then cold drawn in order to have high strength and durability of the rope. The steel wire ropes are manufactured by special machines. First of all, a number of wires such as 7, 19 or 37 are twisted into a strand and then a number of strands, usually 6 or 8 are twisted about a core or centre to form the rope as shown in Fig. 20.7. The core may be made of hemp, jute, asbestos or a wire of softer steel. The core must be continuously saturated with lubricant for the long life of the core as well as the entire rope. The asbestos or soft wire core is used when ropes are subjected to radiant heat such as cranes operating near furnaces. However, a wire core reduces the flexibility of the rope and thus such ropes are used only where they are subjected to high compression as in the case of several layers wound over a rope drum.



(a) 6 × 7 rope. (b) 6 × 19 rope. (c) 6 × 37 rope.

Classification of Wire Ropes

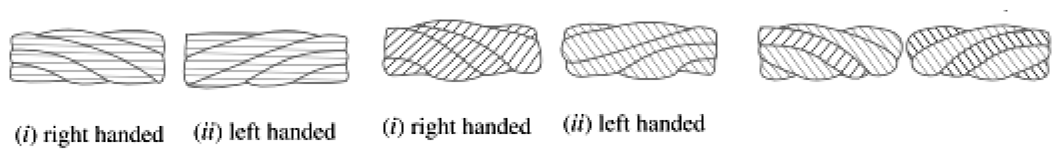
V-belt and Rope Drives

According to the direction of twist of the individual wires and that of strands, relative to each other, the wire ropes may be classified as follows:

1. Cross or regular lay ropes. In these types of ropes, the direction of twist of wires in the strands is opposite to the direction of twist of the stands, as shown in Fig. (a). Such type of ropes are most popular.

2. Parallel or lang lay ropes. In these type of ropes, the direction of twist of the wires in the strands is same as that of strands in the rope, as shown in Fig. (b). These ropes have better bearing surface but is harder to splice and twists more easily when loaded. These ropes are more flexible and resists wear more effectively. Since such ropes have the tendency to spin, therefore these are used in lifts and hoists with guide ways and also as haulage ropes.

3. Composite or reverse laid ropes. In these types of ropes, the wires in the two adjacent strands are twisted in the opposite direction, as shown in Fig. (c).



(a) Cross or regular lay ropes. (b) Parallel or lang lay ropes. (c) Composite or reverse laid ropes.

Designation of Wire Ropes

The wire ropes are designated by the number of strands and the number of wires in each strand. For example, a wire rope having six strands and seven wires in each strand is designated by 6 × 7 rope. Following table shows the standard designation of ropes and their applications:

Standard designation	Application
6 × 7 rope	It is a standard coarse laid rope used as haulage rope in mines, tramways, power transmission.
6 × 19 rope	It is a standard hoisting rope used for hoisting purposes in mines, quarries, cranes, dredges, elevators, tramways, well drilling.
6 × 37 rope	It is an extra flexible hoisting rope used in steel mill laddles, cranes, high speed elevators.
8 × 19 rope	It is also an extra flexible hoisting rope.

Design procedure for wire rope:

Step 1: Selection of suitable wire rope

The suitable type of wire rope is selected based on the application.

Step 2: Calculation of design load:

The design load is calculated by assuming a larger factor of safety say about 15 or 2to 2.5 times the factor of safety given in table from data book pg.no. 9.1.

$$Design\ load = load\ to\ be\ lifted \times assumed\ factor\ of\ safety$$

Step 3: selection of wire rope diameter

It is selected from the table given in data book pg.no. 9.5 & 9.6 by taking the design load as breaking load

Step 4: Calculation of sheave diameter

The sheave diameter is calculated by consulting the table given in data book pg.no. 9.1.

Step 5: selection of area of useful cross section of the rope:

It is calculated using the following relation

$$6 \times 7\ rope, A = 0.38\ d^2\ , for\ 6 \times 19\ rope, A = 0.4\ d^2\ and\ for\ 6 \times 37\ rope, A = 0.4\ d^2$$

Step 6: Calculation of wire diameter

It is calculated using the relation

$$d_w = \frac{d}{1.5\sqrt{i}}$$

Where, i is the number of wires in the rope = no of strands x no of wires in each strand

Step 7: Selection of weight of rope (Wr)

It is selected from the table given in data book pg.no. 9.5 & 9.6

Step 8: Calculation of various loads

(i) Direct load

$$W_d = W + W_r$$

(ii) Bending load

$$W_b = E_r \times \frac{d_w}{D} \times A$$

(iii)Acceleration load due to change in the speed of hoisting

$$W_a = \left[\frac{W + W_r}{g}\right] \times \left(\frac{v_2 - v_1}{t}\right)$$

(iv) Starting or stopping load

a) when there is no slack in the rope

$$W_{st} = 2 \times W_d$$

b) when there is slack in the rope

$$W_{st} = \sigma_{st} \times A = (W + W_r)\left[1 + \sqrt{1 + \frac{2\, a_s\, h\, E_r}{\sigma_d\, l\;\, g}}\right]$$

Step 9: calculation of effective load

(i) Effective load on the rope during normal working,

$$W_{en} = W_d + W_b$$

(ii) Effective load on the rope during acceleration of the load,

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

(iii) Effective load on the rope during starting,

$$W_{est} = W_b + W_{st}$$

Step 10: Calculation of working or actual factor of safety

$$FS_w = \frac{Breaking\ load\ selected}{Effective\ load\ during\ acceleration, W_{ea}}$$

Step 11: Check for safe design

The calculated factor of safety is compared with the recommended FOS, n' given in data book pg.no. 9.1. If FOS working is greater than the recommended, then the design is safe.

If it is not so, then select the next type of rope or increase the breaking strength or increase the number of ropes.

Step 12: calculation of number of ropes:

$$Number\ of\ ropes = \frac{recommended\ factor\ of\ safety, n'}{Working\ fcator\ of\ safety, FS_w}$$

Design of Chain Drives

The chains are made up of number of rigid links which are hinged together by pin joints in order to provide the necessary flexibility for wrapping round the driving and driven wheels. These wheels have projecting teeth of special profile and fit into the corresponding recesses in the links of the chain as shown in Fig. The toothed wheels are known as sprocket wheels or simply sprockets. The sprockets and the chain are thus constrained to move together without slipping and ensures perfect velocity ratio. The chains are mostly used to transmit motion and power from one shaft to another, when the centre distance between their shafts is short such as in bicycles, motor cycles, agricultural machinery, conveyors, rolling mills, road rollers etc. The chains may also be used for long centre distance of upto 8 metres. The chains are used for velocities up to 25 m / s and for power upto 110 kW. In some cases, higher power transmission is also possible.

Advantages and Disadvantages of Chain Drive over Belt or Rope Drive

Following are the advantages and disadvantages of chain drive over belt or rope drive:

Advantages

1. As no slip takes place during chain drive, hence perfect velocity ratio is obtained.

2. Since the chains are made of metal, therefore they occupy less space in width than a belt or rope drive.

3. It may be used for both long as well as short distances.

4. It gives a high transmission efficiency (upto 98 percent).

5. It gives less load on the shafts.

6. It has the ability to transmit motion to several shafts by one chain only.

7. It transmits more power than belts.

8. It permits high speed ratio of 8 to 10 in one step.

9. It can be operated under adverse temperature and atmospheric conditions.

Disadvantages

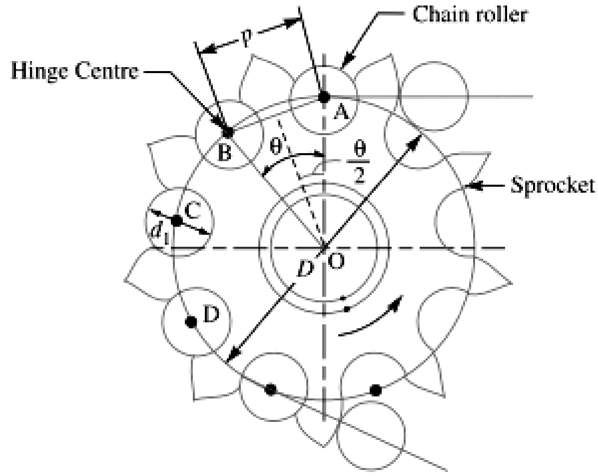
1. The production cost of chains is relatively high.

2. The chain drive needs accurate mounting and careful maintenance, particularly lubrication and slack adjustment.

3. The chain drive has velocity fluctuations especially when unduly stretched.

The following terms are frequently used in chain drive.

1. Pitch of chain. It is the distance between the hinge centre of a link and the corresponding hinge centre of the adjacent link, as shown in Fig. It is usually denoted by p.



2. Pitch circle diameter of chain sprocket. It is the diameter of the circle on which the hinge centers of the chain lie, when the chain is wrapped round a sprocket as shown in Fig. The points A, B, C, and D are the hinge centers of the chain and the circle drawn through these centers is called pitch circle and its diameter (D) is known as pitch circle diameter.

Classification of Chains

The chains, on the basis of their use, are classified into the following three groups:

1. Hoisting and hauling (or crane) chains,

2. Conveyor (or tractive) chains, and

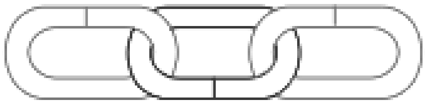
3. Power transmitting (or driving) chains.

Hoisting and Hauling Chains

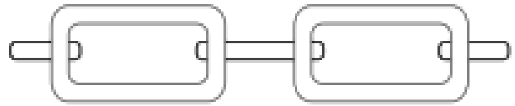
These chains are used for hoisting and hauling purposes and operate at a maximum velocity of 0.25 m / s. The hoisting and hauling chains are of the following two types:

1. Chain with oval links. The links of this type of chain are of oval shape, as shown in Fig. The sprockets which are used for this type of chain have receptacles to receive the links. Such types of chains are used only at low speeds such as in chain hoists and in anchors for marine works.

2. Chain with square links. The links of this type of chain are of square shape, as shown in Fig. Such type of chains is used in hoists, cranes, dredges. The manufacturing cost of this type of chain is less than that of chain with oval links, but in these chains, the kinking occurs easily on overloading.



(a) Chain with oval links.



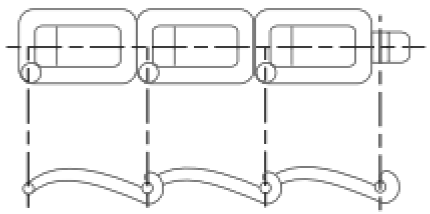
(b) Chain with square links.

Conveyor Chains

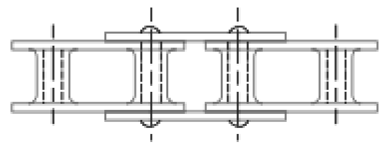
These chains are used for elevating and conveying the materials continuously at a speed upto 2 m / s. The conveyor chains are of the following two types:

1. Detachable or hook joint type chain, as shown in Fig. (a), and

2. Closed joint type chain, as shown in Fig. (b).



(a) Detachable or hook joint type chain.



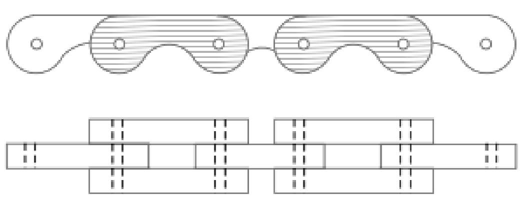
(b) Closed joint type chain.

The conveyor chains are usually made of malleable cast iron. These chains do not have smooth running qualities. The conveyor chains run at slow speeds of about 0.8 to 3 m / s.

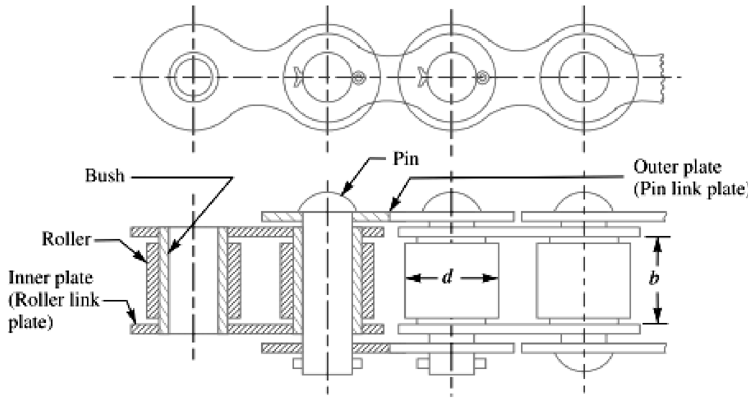
Power Transmitting Chains

These chains are used for transmission of power, when the distance between the centers of shafts is short. These chains have provision for efficient lubrication. The power transmitting chains are of the following three types.

1. Block or bush chain. A block or bush chain is shown in Fig. This type of chain was used in the early stages of development in the power transmission. It produces noise when approaching or leaving the teeth of the sprocket because of rubbing between the teeth and the links. Such type of chains is used to some extent as conveyor chain at small speed.

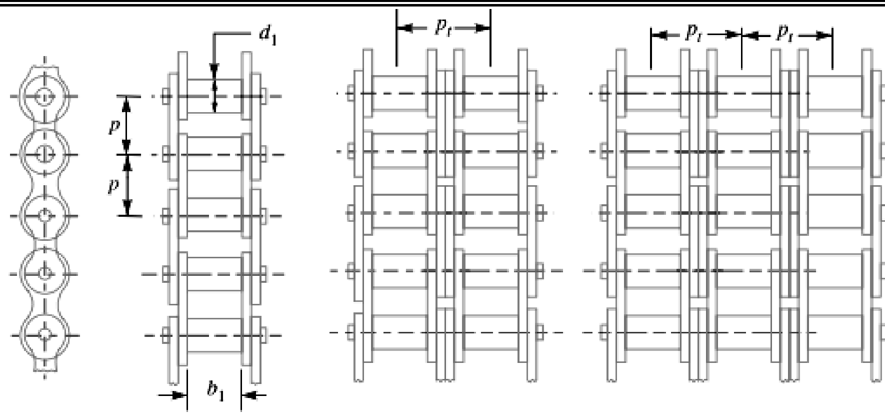


2. Bush roller chain. A bush roller chain as shown in Fig. consists of outer plates or pin link plates, inner plates or roller link plates, pins, bushes and rollers. A pin passes through the bush which is secured in the holes of the roller between the two sides of the chain. The rollers are free to rotate on the bush which protects the sprocket wheel teeth against wear. The pins, bushes and rollers are made of alloy steel.

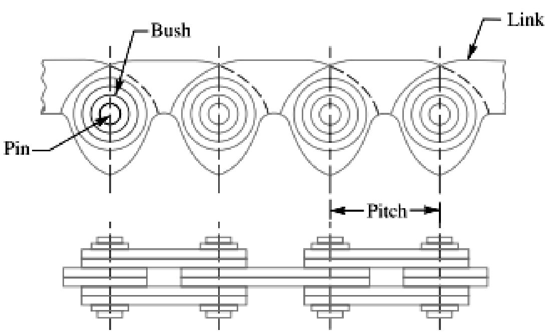


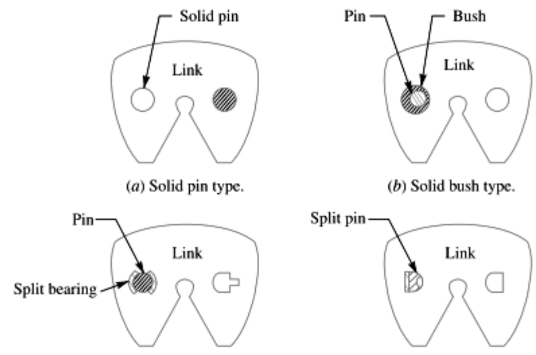
A bush roller chain is extremely strong and simple in construction. It gives good service under severe conditions. There is a little noise with this chain which is due to impact of the rollers on the sprocket wheel teeth. This chain may be used where there is a little lubrication. When one of these chains elongates slightly due to wear and stretching of the parts, then the extended chain is of greater pitch than the pitch of the sprocket wheel teeth. The rollers then fit unequally into the cavities of the wheel. The result is that the total load falls on one teeth or on a few teeth. The stretching of the parts increase wear of the surfaces of the roller and of the sprocket wheel teeth. The roller chains are standardized and manufactured on the basis of pitch. These chains are available in single-row or multi-row roller chains such as simple, duplex or triplex strands, as shown in Fig.

3. Silent chain. A silent chain (also known as inverted tooth chain) is shown in Fig. It is designed to eliminate the evil effects caused by stretching and to produce noiseless running. When the chain stretches and the pitch of the chain increases, the links ride on the teeth of the sprocket wheel at a slightly increased radius. This automatically corrects the small change in the pitch. There is no relative sliding between the teeth of the inverted tooth chain and the sprocket wheel teeth. When properly lubricated, this chain gives durable service and runs very smoothly and quietly. The various types of joints used in a silent chain are shown in Fig.



Simplex chain. Duplex chain. Triplex chain.





(a) Solid pin type. (b) Solid bush type. (c) Split bush type. (d) Rocket pin type.

DESIGN PROCEDURE:

Step 1: Selection of transmission ratio (i)

It is selected from the table given in data book pg.no. 7.74

Step 2: Selection of number of teeth on the driver sprocket

It is selected from the data book pg.no. 7.74 based on the transmission ratio.

Step 3: Determination of number of teeth on the driver sprocket

$$z_2 = i \times z_1$$

The recommended value is around 100 to 120 from data book pg.no. 7.74.

Step 4: Selection of standard pitch (p)

By knowing the initial center distance (a), determine the range of chain pitch by using the relation

$$a = (30 - 50)p$$

From the pitch range obtained, select the suitable standard pitch from the table given in data book pg.no. 7.74.

Step 5: Selection of the chain:

The chain type and chain number is selected from the table given in data book pg.no. 7.71 to 7.73 by using the selected pitch. Initially assume the chain to be simplex or duplex.

Step 6: Calculation of total load on the driving side of the chain (P_T)

$$P_T = P_t + P_c + P_s$$

(i) to find tangential force (P_t)

$$P_t = \frac{1020\ N}{v}$$

Where , N is the transmitted power in kW and

$$v = chain\ velocity\ in\ m/s = \frac{z_1 \times p \times N_1}{60 \times 1000}\ or\ \frac{z_2 \times p \times N_2}{60 \times 1000}$$

(ii) To find centrifugal tension (P_c)

$$P_c = m\,v^2$$

Where, m is the mass of the chain per meter from data book pg.no. 7.71 to 7.73

(iii) to find the tension due to sagging

$$P_s = k\ \ w\ \ a$$

Where,

k = coefficient of sagging from table in data book pg.no. 7.78

w = weight of the chain/meter = m . g.

a = center distance, m

Step 7: Calculation of service factor:

$$Service\ factor, k_s = \ k_1.k_2.k_3.k_4.k_5.k_6$$

Where, k_1 is the load factor (data book pg.no. 7.76), k_2 = factor of distance regulation (pg.no. 7.76)

k_3 – factor for center distance of sprockets (pg.no. 7.76),

k_4 – factor for position of the sprockets (pg.no. 7.77),

k_5 - lubrication factor (pg.no. 7.77),

k_6 - Rating factor (pg.no. 7.77)

Step 8: Calculation of design load:

$$Design\ load = P_T \times k_s$$

Step 9: Calculation of working factor of safety (FS_w)

$$FS_w = \frac{Q}{P_T \times k_s}$$

Step 10: Check for factor of safety

The working FOS is compared with the recommended FOS from the table given in data book pg.no.7.77

If it is greater than the recommended value design is safe otherwise change the chain type from simplex to duplex or duplex to triplex or increase the pitch.

Step 11: Cheack for bearing stress in the roller

$$Bearing\ stress, \sigma = \frac{P_t \times k_s}{A}$$

Where, A is the bearing area given in table from data book pg.no. 7.71 to 7.73

If the calculated bearing stress is less than the allowable stress taken from data book pg.no. 7.77, then design is safe.

Step 12: Calculation of actual length of the chain (L)

Find the number of links in the chain, l_p using relation given in data book pg.no. 7.75.

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

Where,

$$a_p = \frac{a_o}{p} = \frac{Initial\ center\ distance}{pitch}$$

The actual length,

$$L = l_p \times p$$

Step 13: Calculation of exact center distance:

It is calculated by using the relations given in data book pg.no. 7.75

$$Exact\ center\ distance, a = \frac{e + \sqrt{e^2 - 8M}}{4} \times p$$

Where,

$$e = l_p - \left(\frac{z_1 + z_2}{2}\right)\ and\ M = \left(\frac{z_2 - z_1}{2\pi}\right)^2 = constant$$

Step 14: Calculation of pitch circle diameters of the sprockets:

$$pcd\ of\ smaller\ sprocket, d_1 = \frac{p}{\sin\left(\frac{180}{z_1}\right)}$$

$$pcd\ of\ larger\ sprocket, d_1 = \frac{p}{\sin\left(\frac{180}{z_2}\right)}$$

The outer diameter of smaller and larger sprocket is given by

$$d_{o1} = d_1 + 0.8\ d_r \quad and\ d_{o2} = d_2 + 0.8\ d_r$$

Where, d_r is the diameter of the roller taken from table given in pg.no. 7.71 to 7.73

Problem:

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 lubrication. Rating is continuous with 3 shifts per day. Design the complete chain drive.

Solution:

Step 1: Selection of transmission ratio (i)

It is given , i = 2.4

$$Speed\ of\ the\ driven, N_2 = \frac{N_1}{i} = \frac{1440}{2.4} = 600\ rpm$$

Step 2: Selection of number of teeth on the driver sprocket

It is selected from the data book pg.no. 7.74 based on the transmission ratio.

$$z_1 = 27\ teeth$$

Step 3: Determination of number of teeth on the driver sprocket

$$z_2 = i \times z_1 = 2.4 \times 27 = 64.8 \approx 65\ teeth$$

The recommended value is around 100 to 120 from data book pg.no. 7.74.

Step 4: Selection of standard pitch (p)

By knowing the initial center distance (a), determine the range of chain pitch by using the relation

$$a = (30 - 50)p$$

$$maxi\ pitch, P_{maxi} = \frac{a}{30} = \frac{500}{30} = 16.6\ mm$$

$$mini\ pitch, P_{mini} = \frac{a}{50} = \frac{500}{50} = 10\ mm$$

From the pitch range obtained, select the suitable standard pitch from the table given in data book pg.no. 7.74.

$$p = 15.875\ mm$$

Step 5: Selection of the chain:

The chain type and chain number is selected from the table given in data book pg.no. 7.71 to 7.73 by using the selected pitch. Initially assume the chain to be simplex.

$$\boldsymbol{Chain\ slected = \ 10A - 1\ /R50}$$

Step 6: Calculation of total load on the driving side of the chain (P_T)

$$P_T = P_t + \ P_c + P_s$$

(i) to find tangential force (P_t)

$$P_t = \frac{1020\ N}{v} = \frac{1020 \times 4.5}{10.287} = 446.19\ N$$

Where , N is the transmitted power in kW and

$$v = chain\ velocity\ in\ \ m/s = \frac{z_1 \times \ p \times N_1}{60 \times 1000} = \frac{27 \times \ 15.875 \times 1440}{60 \times 1000} = 10.287\ m/s$$

(ii) To find centrifugal tension (P_c)

$$P_c = m\ v^2 = 1.01 \times 10.287^2 \ = 106.88\ N$$

Where, m is the mass of the chain per meter from data book pg.no. 7.71 to 7.73

(iii) to find the tension due to sagging

$$P_s = k\ \ \ w\ \ a\ \ = 6 \times 1.01 \times 9.81 \times 0.5 = 29.72\ N$$

Where,

k = coefficient of sagging from table in data book pg.no. 7.78 = 6 for horizontal drive

w = weight of the chain/meter = m . g.

a = center distance, m

$$Total\ load, P_T = 466.19 + 106.88 + 29.72 = 582.79\ N$$

Step 7: Calculation of service factor:

$$Service\ factor, k_s = \ \ k_1.\ k_2.k_3.k_4.k_5.k_6$$

Where, k_1 is the load factor (data book pg.no. 7.76) = 1.25

k_2 = factor of distance regulation (pg.no. 7.76) = 1

k_3 – factor for center distance of sprockets (pg.no. 7.76) = 1

k_4 – factor for position of the sprockets (pg.no. 7.77) = 1

k_5 - lubrication factor (pg.no. 7.77) = 0.8

k_6 - Rating factor (pg.no. 7.77) = 1.5

$$Service\ factor, k_s = \ 1.25 \times 1 \times 1 \times 1 \times \ 0.8 \times 1.5 = 1.5$$

Step 8: Calculation of design load:

$$Design\ load = P_T \times k_s = 582.79 \times 1.5 = 874.19\ N$$

Step 9: Calculation of working factor of safety (FS_w)

$$FS_w = \frac{Q}{P_T \times k_s} = \frac{22200}{874.19} = 25.39$$

Step 10: Check for factor of safety

The working FOS is compared with the recommended FOS from the table given in data book pg.no.7.77. The FS_w is 25.39, where as the recommended value is 13.2, which is greater than the recommended value, hence design is safe.

Step 11: Cheack for bearing stress in the roller

$$Bearing\ stress, \sigma = \frac{P_t \times k_s}{A} = \frac{446.19 \times 1.5}{70} = 9.56\ N/mm^2$$

Where, A is the bearing area given in table from data book pg.no. 7.71 to 7.73, A = 70mm^2

The calculated bearing stress is less than the allowable stress of 18.5 N/mm^2 taken from data book pg.no. 7.77, hence the design is safe.

Step 12: Calculation of actual length of the chain (L)

Find the number of links in the chain, l_p using relation given in data book pg.no. 7.75.

$$l_p = 2\ a_p + \left[\frac{z_1 + \ z_2}{2}\right] + \frac{\left(\frac{z_2 - z_1}{2\pi}\right)^2}{a_p} = 110.153 \approx 112$$

Where,

$$a_p = \frac{a_o}{p} = \frac{Initial\ center\ distance}{pitch} = \frac{500}{15.875} = 31.496$$

The actual length,

$$L = l_p \times p = 1778\ mm$$

Step 13: Calculation of exact center distance:

It is calculated by using the relations given in data book pg.no. 7.75

$$Exact\ center\ distance, a = \frac{e + \sqrt{e^2 - 8M}}{4} \times p = 514.92\ mm$$

Where,

$$e = l_p - \left(\frac{z_1 + z_2}{2}\right) = 66\ \ and\ M = \left(\frac{z_2 - z_1}{2\pi}\right)^2 = 36.57 = constant$$

Step 14: Calculation of pitch circle diameters of the sprockets:

$$pcd\ of\ smaller\ sprocket, d_1 = \frac{p}{\sin\left(\frac{180}{z_1}\right)} = 136.74\ mm$$

$$pcd\ of\ larger\ sprocket, d_2 = \frac{p}{\sin\left(\frac{180}{z_2}\right)} = 328.58\ mm$$

The outer diameter of smaller and larger sprocket is given by

$$d_{o1} = d_1 + 0.8\ d_r = 144.87\ mm\ \ and\ d_{o2} = d_2 + 0.8\ d_r = 336.71\ mm$$

Where, d_r = 10.16 mm is the diameter of the roller taken from table given in pg.no. 7.71 to 7.73

MULTIPLE CHOICE QUESTIONS

	QUESTION	OPTION 1	OPTION 2	OPTION 3	OPTION 4	ANSWER
1	Pitch circle diameter is equal to the product of………..	Flat pivot bearing	Flat collar bearing	Conical pivot bearing	Truncated conical pivot bearing	Flat collar bearing
2	In a disc clutch, if there are n_1 number of discs on the driving shaft and n_2 number of discs on the driven shaft, then the number of pairs of contact surfaces will be	n_1+n_2	n_1+n_2+1	n_1+n_2-1	n_1+n_2-2	n_1+n_2-1
3	The frictional torque transmitted by a cone clutch is same as that of	Flat pivot bearing	Flat collar bearing	Conical pivot bearing	Truncated conical pivot bearing	Truncated conical pivot bearing
4	Claw clutch is	Positive drive because there is no slip in power transmission	Mostly preferred in automobiles because of smooth engagement	A friction drive because it consists of 2 pairs of friction surfaces	Seldom used in machine tools because it transmit power in one direction only	Positive drive because there is no slip in power transmission
5	The cone clutches	Have become obsolete because of small cone angles	Have become obsolete because of small cone angles	Are used widely because it is a positive drive	Has two pairs of friction discs because it is a multiple disk drive	Have become obsolete because of small cone angles
6	A cam profile is the	Profile of traced by a cam follower	Actual working contour of the cam	The profile of the path to be traced by the mechanism	Total surface area of the cam	Actual working contour of the cam
7	Cams are classified on the basis of	Line of the motion of the follower with respect to the axis of cam	Surface in contact between the cam and follower	Type of motion of the follower on the cam	All of these	All of these
8	The pitch point on the cam is	Any point on the pitch curve	The point at a distance equal to pitch circle radius from center	A point at a distance equal to pitch circle radius from center	None of these	The point at a distance equal to pitch circle radius from center
9	The cam profile and the cam pitch are same for	Roller follower	Flat faced follower	Knife edged follower	All of these	Knife edged follower

10	The reference point on the follower for laying cam profile is	Pitch point	Cam point	Trace point	Roller point	Trace point
11	The minimum radius circle drawn tangent to cam profile is	Prime circle	Pitch circle	Base circle	Pitch curve	Base circle
12	The ordinates and abscissa of a Flat faced follower displacement diagram are	Velocity and acceleration	SHM and velocity	Stroke and cam angle	Cam angle and velocity	Stroke and cam angle
13	The point on cam profile with maximum pressure angle is	Trace point	Pitch point	Pressure point	Cam center	Pitch point
14	If the speed of engine is moderate, the cam follower should move with	Uniform acceleration and retardation	Single harmonic motion	Uniform velocity	Cycloidal motion	Single harmonic motion
15	According to principal of kinematic inversion	Followers is stationary and cam rotates	Cam is stationary and follower rotates	Both cam and follower are moving	Both cam and follower are stationary	Cam is stationary and follower rotates
16	If is the distance between circular blank and centre of nose, then decelerations of flats follower when in catered with apex of the more of a circular are can be given by	ω x MN cos ɵ	ω x N	ω^2 x MN	ω x MN^2	ω^2 x MN
17	For the following moving with instant velocity acceleration in the followers at the time when is starts lifting is	Infinity	0	Small	Velocity	Infinity
18	The cam follower generally used in air craft engine is	Roller follower	Flat faced follower	Knife edged follower	Spherical faced followers	Flat faced follower
19	For low and moderate speed engines, the cam follower should move with	Uniform acceleration and retardation	Single harmonic motion	Uniform velocity	Cycloid motion	Cycloid motion
20	The displacement of a flat-faced follower when it has contact with the flank of the circular arc cam is given by	R(1-cos ɵ)	(R-r_1) (1-cos ɵ)	R(1-sin ɵ)	(R-r_1) (1-sin ɵ)	(R-r_1) (1-cos ɵ)
21	The retardation of a flat-faced follower when it has contact with the flank of the	ω^2 x OQ	ω^2 x OQ sin ɵ	ω^2 x OQ cos ɵ	ω^2 x OQ tan ɵ	ω^2 x OQ

	circular arc cam is given by					
22	Smallest circle drawn to the cam profile from the cam centre is known as	Prime circle	Base circle	Pitch circle	Pitch curve	Base circle
23	The circle, with centre as the centre of cam axis and radius such that it passes through the pitch point, is known as	Prime circle	Base circle	Pitch circle	Pitch curve	Pitch circle
24	On which circle or curve the size of the cam depends?	Prime circle	Base circle	Pitch circle	Pitch curve	Base circle
25	The pressure angle of a cam is the angle at any point on the pitch curve	Made by line of motion of the follower with horizontal axis	Made by line of motion of the follower with vertical axis	Between the normal to that point on the curve and the line of motion of follower at that instant	Between the tangent to that point on the curve and the line of motion of follower at that instant	Between the normal to that point on the curve and the line of motion of follower at that instant
26	Choose the wrong statement	The reference point for a knife edge follower is the edge of the knife	The reference point for a roller follower is the centre of the roller	The pitch curve and the cam profile are the for a knife edge follower	The pressure angle is the angle between the directions of the follower motion and a tangent to the pitch curve.	The pressure angle is the angle between the directions of the follower motion and a tangent to the pitch curve.
27	Which factor affects the life of the cam?	Base circle	Pressure angle	Hub size	All of these	Pressure angle
28	The term $\omega^2 x\ d^3y/dq^3$ in a cam follower motion, represents	Acceleration of follower	Velocity of follower	Jerk	Displacement	Acceleration of follower
29	The dynamic friction is the friction experienced by a body, when the body	Is in motion	Is at rest	Just begins to slide over the surface of the other body	None of these	Is in motion
30	Which of the following statements regarding laws governing the friction between dry surfaces are correct?	The frictional force is dependent on the materials of the contact surfaces	The frictional force is directly proportional to the normal force	The frictional force is independent of the area of contact	All of these	All of these
31	In a screw jack the load cup	Is made as integral part	is made separate from the spindle	Enhancers the capacity	Prevents the toppling of	Is made separate from

		because it reduces the friction	because it will prevent the rotation of load being lifted	of screw jack became it improves the mechanical advantages	load because it rotates along with the screw rod	the spindle because it will prevent the rotation of load being lifted
32	In a screw jack, the effort required to lift the load W is given by	P = W tan (α - φ)	P = W tan (α + φ)	P = W tan (φ - α)	P = W cos (α + φ)	P = W tan (α + φ)
33	(S1) The static friction is independent of the area of contact, between the two surface	S1 is right	S2 is right	Both S1 & S2 are right	Both S1 & S2 are wrong	Both S1 & S2 are right
34	In a screw jack, the effort required to lower the load W is given by	P = W tan (α - φ)	P = W tan (α + φ)	P = W tan (φ - α)	P = W cos (α + φ)	P = W tan (φ - α)
35	Frictional torque for square thread at mean radius r while raising load W is given by	T = W.r tan (φ - α)	T = W.r tan (φ + α)	T = W.r tan α	T = W.r tan φ	T = W.r tan (φ + α)
36	The frictional torque transmitted in a conical pivot bearing, considering uniform wear, is	1/2 µ *W R cosec α*	2/3 µ *W R cosec α*	3/4 µ *W R cosec α*	µ *W R cosec α*	1/2 µ *W R cosec α*
37	The frictional torque transmitted by a disc or plat clutch is same as that of	Flat pivot bearing	Flat collar bearing	Conical pivot bearing	Truncated conical pivot bearing	Flat collar bearing
38	Brake is a device used for bringing a moving body	To rest	To retard the motion	To keep it in a state of rest against the external forces	All of these	All of these
39	Material used for brake lining should not have	High heat dissipation capacity	Low heat resistance	High strength	High coefficient of friction	Low heat resistance
40	In a Geneva wheel mechanism, the number of slots in the wheel is 4. For one full revolution of the wheel, the driver disc has to make	16 revolutions	¼ revolutions	4 revolutions	8 revolutions	4 revolutions
41	A Geneva wheel has 6 slots. Driving crank radius is 100 mm. the distance between centres is	200 mm	100 mm	50 mm	141.44 mm	200 mm

42	Properties of a good brake lining material are	High heat dissipation capacity	High coefficient of friction	High strength	All of these	All of these
43	Reduce the impact load in Geneva wheel…	Increase the number of slots	Reduces the number of slots	Increase the pin diameter	Increase the pin diameter	Increase the number of slots
44	Dwell period (in caser of cam and follower) is the tim	That cam rotates	During which the follower moves from its lowest position to highest posit	That cam not rotates.	None of these	None of these
45	The dynamic friction is the friction experienced by a body, when the body	Is rolling	Is at rest	Just begins to slide over the surface of the other body	None of these	None of these
46	To determine the geometric dimensions of a Geneva wheel, the most important parameter to be known are	Diameter of the Geneva wheel and its shaft.	Slot length and width	Diameter of the driving disc and its shafts	Number of slots and distance between centers.	Number of slots and distance between centers.
47	The pawl of the ratchet wheel mechanism is subjected to	Direct load	Bending moment	Twisting moment	Direct load & Bending moment	Direct load & Bending moment
48	The factor affects the life of the cam is	Base circle	Cam angle	Hub size	None of these	None of these

UNIT II
DESIGN OF SPUR AND HELICAL GEARS

Gear Terminology – Speed ratios and number of teeth–Force analysis – Tooth stresses – Dynamic effects – Fatigue strength – Factor of safety – Gear materials – Module and Face width–power rating calculations based on strength and wear considerations – Parallel axis Helical Gears – Pressure angle in the normal and transverse plane– Equivalent number of teeth–forces and stresses – Estimating the size of the helical gears.

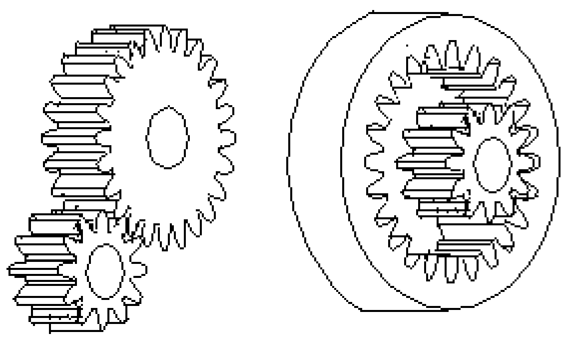
Gears are machine elements that transmit motion by means of successively engaging teeth. The gear teeth act like small levers.

Gears may be classified according to the relative position of the axes of revolution. The axes may be

1. parallel,
2. intersecting,
3. Neither parallel nor intersecting.

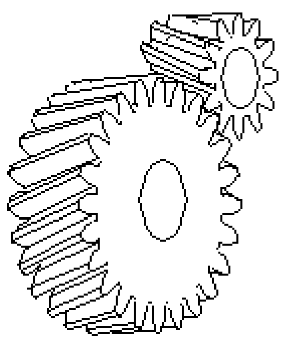
Gears for connecting parallel shafts

1. *Spur gears*

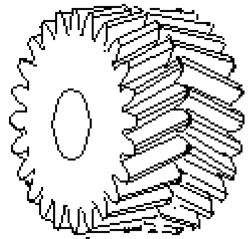


The left pair of gears makes **external contact**, and the right pair of gears makes **internal contact**

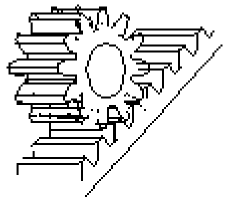
2. *Parallel helical gears*



3. *Herringbone gears* (or double-helical gears)

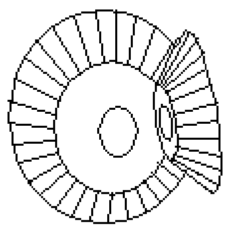


4. *Rack* and *pinion* (The rack is like a gear whose axis is at infinity.)



Gears for connecting intersecting shafts

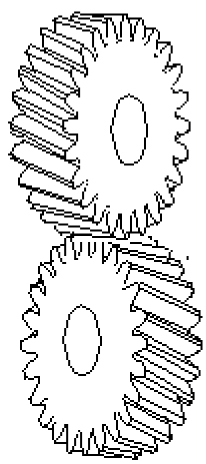
1. *Straight bevel gears*



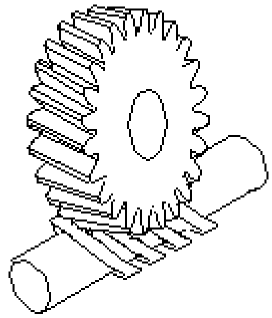
2. *Spiral bevel gears*

Neither parallel nor intersecting shafts

1. *Crossed-helical gears*



2. *Hypoid gears*
3. *Worm and worm gear*

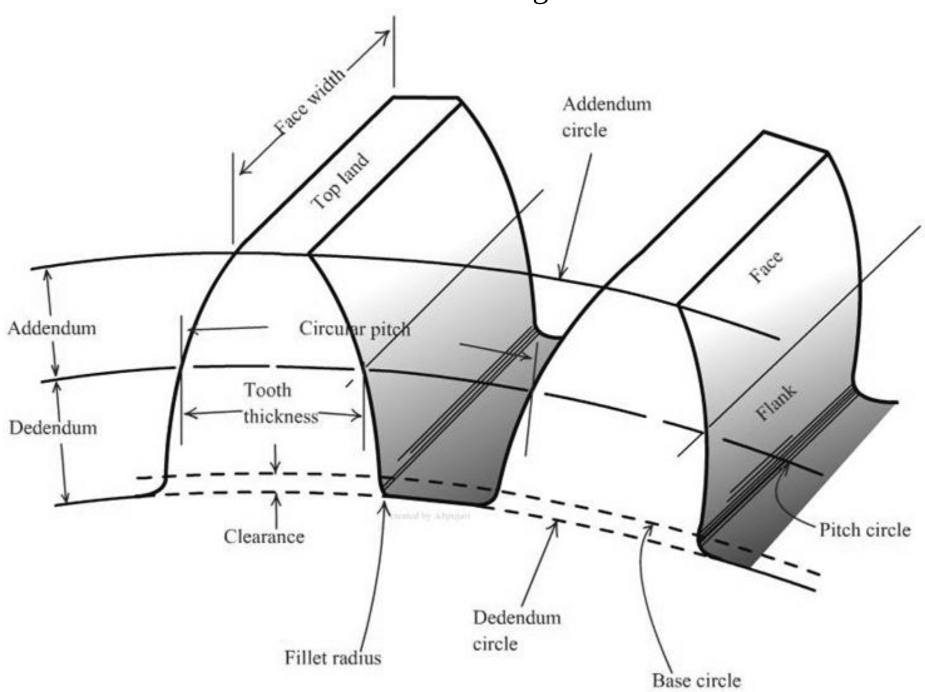


The **fundamental law of gear-tooth action** may now also be stated as follow (for gears with fixed center distance). The common normal to the tooth profiles at the point of contact must always pass through a fixed point (the pitch point) on the line of centers (to get a constant velocity ration).

Terminology for Spur Gears

In the following section, we define many of the terms used in the analysis of spur gears. Some of the terminology has been defined previously but we include them here for completeness.

- **Pitch surface**: The surface of the imaginary rolling cylinder (cone, etc.) that the toothed gear may be considered to replace.
- **Pitch circle**: A right section of the pitch surface.
- **Addendum circle**: A circle bounding the ends of the teeth, in a right section of the gear.
- **Root (or dedendum) circle**: The circle bounding the spaces between the teeth, in a right section of the gear.
- **Addendum**: The radial distance between the pitch circle and the addendum circle.
- **Dedendum**: The radial distance between the pitch circle and the root circle.
- **Clearance**: The difference between the dedendum of one gear and the addendum of the mating gear.



- **Face of a tooth**: That part of the tooth surface lying outside the pitch surface.
- **Flank of a tooth**: The part of the tooth surface lying inside the pitch surface.

- **Circular thickness** (also called the **tooth thickness**): The thickness of the tooth measured on the pitch circle. It is the length of an arc and not the length of a straight line.
- **Tooth space**: The distance between adjacent teeth measured on the pitch circle.
- **Backlash**: The difference between the circle thickness of one gear and the tooth space of the mating gear.
- **Circular pitch** p_c: The width of a tooth and a space, measured on the pitch circle.

$$p_c = \frac{\pi D}{z}$$

- **Diametral pitch** P_d: The number of teeth of a gear per inch of its pitch diameter. A toothed gear must have an integral number of teeth. The *circular pitch*, therefore, equals the pitch circumference divided by the number of teeth. The *diametral pitch* is, by definition, the number of teeth divided by the *pitch diameter*. That is,

$$p_d = \frac{z}{D} = \frac{\pi}{p_c}$$

  Where,
  p_c = circular pitch, p_d = diametral pitch, z = number of teeth, D = pitch diameter
  That is, the product of the diametral pitch and the circular pitch equals π.
- **Module** m: Pitch diameter divided by number of teeth. The pitch diameter is usually specified in inches or millimeters; in the former case the module is the inverse of diametral pitch.

$$m = \frac{D}{z}$$

- **Fillet:** The small radius that connects the profile of a tooth to the root circle.
- **Pinion**: The smaller of any pair of mating gears. The larger of the pair is called simply the gear.
- **Velocity ratio**: The ratio of the number of revolutions of the driving (or input) gear to the number of revolutions of the driven (or output) gear, in a unit of time.

$$i = \frac{z_2}{z_1} = \frac{N_1}{N_2}$$

  Where, N_1 – speed of the pinion (small gear), N_2 – sped of the gear (big gear)

- **Pitch point**: The point of tangency of the pitch circles of a pair of mating gears.
- **Common tangent**: The line tangent to the pitch circle at the pitch point.
- **Line of action**: A line normal to a pair of mating tooth profiles at their point of contact.
- **Path of contact**: The path traced by the contact point of a pair of tooth profiles.
- **Pressure angle**, α: The angle between the common normal at the point of tooth contact and the common tangent to the pitch circles. It is also the angle between the line of action and the common tangent.
- **Base circle**: An imaginary circle used in involute gearing to generate the involutes that form the tooth profiles.

Gear profiles:

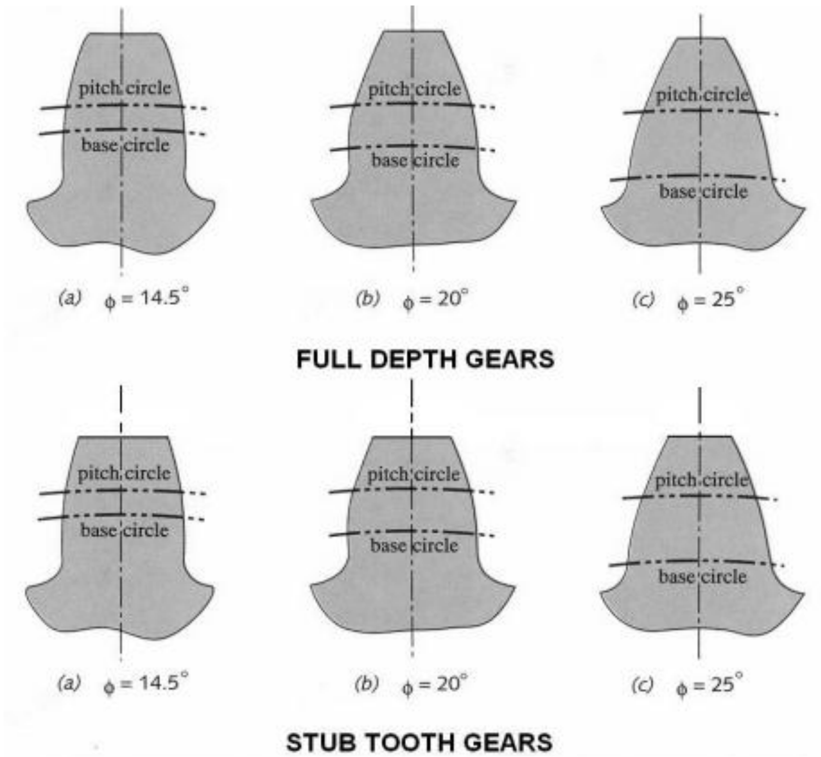
1. Involute: The involute gear profile is the most commonly used system for gearing today, with cycloid still used for some specialties such as clocks. In an involute gear, the profiles of the teeth are involutes of a circle. (The involute of a circle is the spiraling curve traced by the end of an imaginary taut string unwinding itself from that stationary circle called the base circle.)

2. Cycloid: The cycloidal gear profile is a form of toothed gear used in mechanical clocks, rather than the involute gear form used for most other gears. The gear tooth profile is based on the epicycloid and hypocycliod

curves, which are the curves generated by a circle rolling around the outside and inside of another circle, respectively.

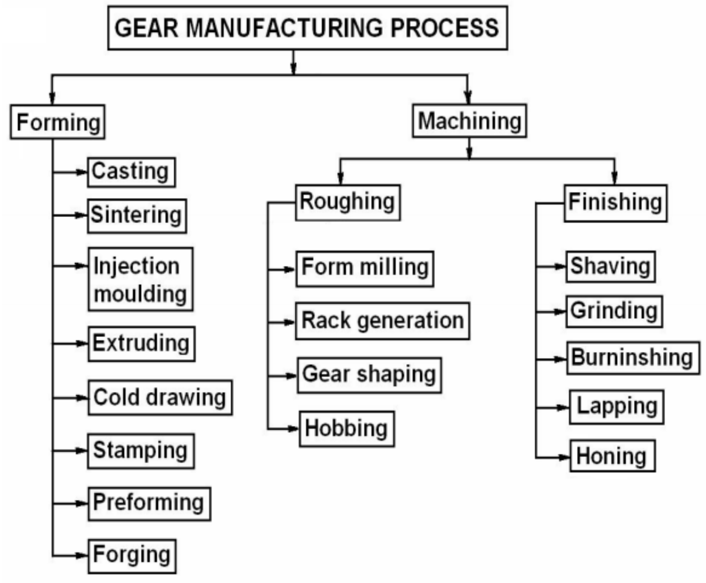
Standard Tooth Systems for Spur Gears

To reduce the varieties of gears to a manageable numbers, standards are evolved. Standard makes it easy for design, production, quality assurance, replacement etc. Three commonly used pressure angles are 14.5°, 20° and 25° pressure angle systems as shown in Fig. In this, one can have full depth gears or stronger stub tooth gears. In Standard tooth system for metric gears, addendum: a =1m, dedendum: b= 1.25m where as the for the stub tooth gears, addendum a = 0.8m and dedendum: b= 1.0m. The shorter tooth makes it stronger and its load carrying capacity increases. It also helps in avoiding interference in certain cases.



Gear Manufacturing

Gear manufacturing can be divided into two categories namely forming and machining as shown in flow chart in Fig. Forming consists of direct casting, molding, drawing, or extrusion of tooth forms in molten, powdered, or heat softened materials and machining involves roughing and finishing operations.



Design of Spur gear based on Beam Strength using Lewis and Buckingham equations:

Design Procedure:

Step 1: Selection of material

From PSGDB pg. no. 1.40, the suitable material for gear and pinion are selected if not given in the problem.

Step 2: To find z_1 and z_2

If number of teeth in the pinion is not given assume them to be, i.e, $z_1 \geq 17$, say 18 teeth

The number of teeth in the gear is given by, gear ratio or velocity ratio or transmission ratio, i

$$i = \frac{z_2}{z_1} = \frac{N_1}{N_2}$$

Where, N_1 – speed of the pinion (small gear), N_2 – sped of the gear (big gear)

Step 3: To find the tangential load on the gear tooth, F_t

$$F_t = \frac{P}{v} \times k_o, \qquad N$$

Where, P – power transmitted,

v – pitch line velocity, m/s;

$$v = \frac{\pi\, d\, N}{60} = \frac{\pi\, m\, z\, N}{60 \times 1000}\ m/s$$

k_o – shock factor

Type of load	Shock factor, k_o
Steady	1.0
Light shock	1.25
Medium shock	1.5
Heavy Shock	2.0

Step 4: Calculation of initial dynamic load, F_d

$$F_d = \frac{F_t}{c_v}, \qquad N$$

Where, c_v – velocity factor (from PSGDB pg.no. 8.54)

In general always take that to be,

$$c_v = \frac{6}{6+v}, with\ v\ assumed\ as\ 10\ to\ 15\ m/s$$

Step 5: Calculation of beam strength, F_s

According to Lewis beam strength equation,

$$F_s = \pi\, m\, b\, [\sigma_b]\, y$$

Where, b – face width in mm, (assume it to be 10 m); $[\sigma_b]$ – allowable static stress of gear material, N/mm^2, if not given may be taken as $[\sigma_b] = \sigma_u / 3$; y – Form factor, taken from PSGDB pg. no. 8.50 based on the pressure angle.

Step 6: To find module, m

By taking the condition,

$$F_s \geq F_d$$

Find the module, m and select the standard module from the PSGDB pg.no. 8.2.

Step 7: To find b, d and v

Face width, b = 10 m

Pitch circle diameter of the gear, d = m z

Pitch line velocity, v

$$v = \frac{\pi\, d\, N}{60 \times 1000}\ m/s$$

Step 8: Revised beam strength, F_s

$$F_s = \pi\, m\, b\, [\sigma_b]\, y, \qquad N$$

Step 9: To find accurate dynamic load, F_d

It is calculated using the Buckingham's equation,

$$F_d = F_t + F_I, \qquad N$$

Where,

Tangential load, F_t

$$F_t = \frac{P}{v}, \qquad N$$

Incremental load, F_I

$$F_I = \frac{21\ v\ (b\ c + F_t\ )}{21\ v + \sqrt{b\ c + F_t}}, \qquad N$$

Where,

c – Deformation factor taken from PSGDB pg. no. 8.53.

Step 10: Check for beam strength

If $F_s \geq F_d$, design is safe, otherwise increase the face width and module or change the deformation factor value to the precision gears type to fulfill the condition.

Step 11: To find maximum wear load, F_w

$$F_w = d \times b \times Q \times K_w$$

Where, Q – ratio factor

$$Q = \frac{2\ i}{i + 1}$$

And K_w – load stress factor based on the gear materials.

Step 12: check for wear strength

If $F_w \geq F_d$, design is safe

Step 13: To find the basic dimensions of the pinion and gear:

From PSGDB pg. no. 8.22, the formulas to find the basic dimensions are given

- Center Distance, a

$$a = \frac{m\ (z_1 + z_2)}{2}, \qquad mm$$

- Height Factor, f_o = 1 for full depth teeth
- Bottom clearance, c

$$c = 0.25\ m\ for\ FD\ teeth$$

- Tooth depth, h

$$h = 2.25\ m\ for\ FD\ teeth$$

- Pitch circle diameter, d

$$for\ pinion, d_1 = m\ z_1$$

$$for\ gear, d_2 = m\ z_2$$

- Tip diameter, d_a

$$for\ pinion, d_{a1} = (z_1 + 2f_o)\, m$$

$$for\ gear, d_{a2} = (z_2 + 2f_o)\, m$$

- Root diameter, d_f

$$for\ pinion, d_{f1} = (z_1 - 2f_o)\, m - 2c$$

$$for\ gear, d_{f2} = (z_2 - 2f_o)\, m - 2c$$

Note:

- ✓ ***If pinion and gear are made of same material then design pinion alone.***
- ✓ ***If pinion and gear are made of different material then material having low static allowable static stress is to be designed.***

PROBLEMS:

1. Design a spur gear drive required to transmit 15 kW at pinion speed of 1400 rpm to a low speed shaft rotating at 500 rpm. The teeth are 20° full depth involute with 25 teeth on the pinion. Both the pinion and gear are made of C.I with a maximum safe static stress of 56 N/mm².

Solution:

Step 1: Selection of material

Given that, the pinion and gear are made of C.I with a maximum safe static stress of 56 N/mm².

Step 2: To find z_1 and z_2

The number of teeth in the pinion is assumed to be

$$z_1 = 18\ teeth$$

Then from the speed ratio,

$$i = \frac{z_2}{z_1} = \frac{N_1}{N_2} = \frac{1400}{500} = 2.8$$

$$z_2 = i \times z_1 = 2.8 \times 18 = 50.4 \approx 50\ teeth$$

Step 3: To find the tangential load on the gear tooth, F_t

$$F_t = \frac{P}{v} \times k_o = \frac{15 \times 10^3}{1.32\ m} \times 1.25 = \frac{14.2 \times 10^3}{m}, \qquad N$$

Where, P – power transmitted, v – pitch line velocity, m/s;

$$v = \frac{\pi\ d\ N}{60} = \frac{\pi\ m\ z_1\ N_1}{60 \times 1000} = \frac{\pi \times m \times 18 \times 1400}{60 \times 1000} = 1.32\ m, \qquad {}^{m}/_{s}$$

The K_o = 1.25 for medium shock conditions

Step 4: Calculation of initial dynamic load, F_d

$$F_d = \frac{F_t}{c_v} = \frac{14.2 \times 10^3}{m \times 0.333} = \frac{42.7 \times 10^3}{m}, \qquad N$$

Where, c_v – velocity factor (from PSGDB pg. no. 8.54)

In general always take that to be, v is assumed to be 12 m/s

$$c_v = \frac{6}{6+v} = \frac{6}{6+12} = 0.333$$

Step 5: Calculation of beam strength, F_s

According to Lewis beam strength equation,

$$F_s = \pi \times m \times b \times \ [\sigma_b] \times y = \pi \times m \times 10m \times 56 \times 0.1033 = 181.73\, m^2$$

Where, b – face width in mm, (assume it to be 10 m); $[\sigma_b]$ – allowable static stress of gear material, N/mm^2

y – Form factor, taken from PSGDB pg. no. 8.50 based on the pressure angle.

$$y = 0.154 - \frac{0.912}{z_1} = 0.154 - \frac{0.912}{18} = 0.1033$$

Step 6: To find module, m

By taking the condition,

$$F_s \geq F_d$$

$$181.73\, m^2 = \frac{42.7 \times 10^3}{m}$$

$$m = 6.17\ mm$$

Find the module, m and select the standard module from the PSGDB pg.no. 8.2.

$$m = 8\ mm$$

Step 7: To find b, d and v

Face width, b = 10 m = 80 mm

Pitch circle diameter of the gear,

$$d_1 = m \times z_1 = 8 \times 18 = 144\ mm$$

Pitch line velocity, v

$$v = 1.32\, m = 1.32 \times 8 = 10.56\ m/s$$

Step 8: Revised beam strength, F_s

$$F_s = \pi\, m\, b\, [\sigma_b]\, y = \ 181.73\, m^2 = \ 181.73 \times \ 8^2 \ = 11.63 \times 10^3, \qquad N$$

Step 9: To find accurate dynamic load, F_d

It is calculated using the Buckingham's equation,

$$F_d = F_t + F_I = \ \ 1.42 \times 10^3 + 7.143 \times 10^3 \ = 8.56 \times 10^3, \qquad N$$

Where,

Tangential load, F_t

$$F_t = \frac{P}{v} = \frac{15 \times 10^3}{10.56} = 1.42 \times 10^3, \quad N$$

Incremental load, F_I

$$F_I = \frac{21\, v\, (b\, c + F_t)}{21\, v + \sqrt{b\, c + F_t}} = \frac{21 \times 10.56\, (80 \times 112.67 + 1.42 \times 10^3)}{21 \times 10.56 + \sqrt{80 \times 112.67 + 1.42 \times 10^3}}$$

$$F_t = 7.143 \times 10^3, \quad N$$

Where,

c – Deformation factor taken from PSGDB pg. no. 8.53.

$$c = 5930\, e \text{ for CI and CI combination,}$$

$$\text{The expected error, 'e' for module of } 8\, mm \text{ and precision gears is } 0.019$$

$$c = 5930 \times 0.019 = 112.67$$

Step 10: Check for beam strength

$$\text{The beam strength}, F_s > \text{Dynamics load}, F_d$$

Hence design is safe

Step 11: To find maximum wear load, F_w

$$F_w = d_1 \times b \times Q \times K_w = 144 \times 80 \times 1.5 \times 1.42 = 24.53 \times 10^3, \quad N$$

Where, Q – ratio factor

$$Q = \frac{2\, i}{i + 1} = \frac{2 \times 2.8}{2.8 + 1} = 1.5$$

K_w – load stress factor based on the gear materials = 1.42 for CI and CI combination

Step 12: check for wear strength

$$F_w > F_s$$

Hence Design is safe

Step 13: To find the basic dimensions of the pinion and gear:

From PSGDB pg. no. 8.22, the formulas to find the basic dimensions are given

- Center Distance, a

  $$a = \frac{m\,(z_1 + z_2)}{2} = \frac{8 \times (18 + 50)}{2} = 272\, mm$$

- Height Factor, f_o = 1 for full depth teeth
- Bottom clearance, c

  $$c = 0.25\, m = 0.25 \times 8 = 2\, mm$$

- Tooth depth, h

  $$h = 2.25\, m = 2.25 \times 8 = 18\, mm$$

- Pitch circle diameter, d

  $for\ pinion, d_1 = m\, z_1 = 144\ mm$

  $for\ gear, d_2 = m\, z_2 = 400\ mm$

- Tip diameter, da

  $for\ pinion, d_{a1} = (z_1 + 2f_o)\, m = 160\ mm$

  $for\ gear, d_{a2} = (z_2 + 2f_o)\, m = 416\ mm$

- Root diameter, df

  $for\ pinion, d_{f1} = (z_1 - 2f_o)\, m - 2c = 124\ mm$

  $for\ gear, d_{f2} = (z_2 - 2f_o)\, m - 2c = 380\ mm$

2. Design a spur gear drive required to transmit 45 kW at pinion speed of 800 rpm. The velocity ratio 3.5:1. the teeth are 20^o full depths involute with 18 teeth on the pinion. Both the pinion and gear are made of steel with a maximum safe static stress of $180 N/mm^2$.Assume medium shock conditions.

Solution:

Step 1: Selection of material

Given that, the pinion and gear are made of steel with a maximum safe static stress of $180 N/mm^2$.

Step 2: To find z_1 and z_2

The number of teeth in the pinion is assumed to be

$$z_1 = 18\ teeth$$

Then from the speed ratio,

$$i = \frac{z_2}{z_1} = \frac{N_1}{N_2} = 3.5\ (given)$$

$$z_2 = i \times z_1 = 3.5 \times 18 = 63\ teeth$$

Step 3: To find the tangential load on the gear tooth, F_t

$$F_t = \frac{P}{v} \times k_o = \frac{45 \times 10^3}{0.754\ m} \times 1.25 = \frac{74.6 \times 10^3}{m}, \qquad N$$

Where, P – power transmitted, v – pitch line velocity, m/s;

$$v = \frac{\pi\, d\, N}{60} = \frac{\pi\, m\, z_1\, N_1}{60 \times 1000} = \frac{\pi \times m \times 18 \times 800}{60 \times 1000} = 0.754\ m, \qquad {}^{m}/_{s}$$

The K_o = 1.25 for medium shock conditions

Step 4: Calculation of initial dynamic load, F_d

$$F_d = \frac{F_t}{c_v} = \frac{74.6 \times 10^3}{m \times 0.333} = \frac{223.81 \times 10^3}{m}, \qquad N$$

Where, c_v – velocity factor (from PSGDB pg. no. 8.54)

In general always take that to be, v is assumed to be 12 m/s

$$c_v = \frac{6}{6+v} = \frac{6}{6+12} = 0.333$$

Step 5: Calculation of beam strength, F_s

According to Lewis beam strength equation,

$$F_s = \pi \times m \times b \times [\sigma_b] \times y = \pi \times m \times 10m \times 180 \times 0.1033 = 584.15\, m^2$$

Where, b – face width in mm, (assume it to be 10 m); $[\sigma_b]$ – allowable static stress of gear material, N/mm^2

y – Form factor, taken from PSGDB pg. no. 8.50 based on the pressure angle.

$$y = 0.154 - \frac{0.912}{z_1} = 0.154 - \frac{0.912}{18} = 0.1033$$

Step 6: To find module, m

By taking the condition,

$$F_s \geq F_d$$

$$584.15\, m^2 \geq \frac{223.8 \times 10^3}{m}$$

$$m \geq 7.26\, mm$$

Find the module, m and select the standard module from the PSGDB pg.no. 8.2.

$$m = 8\, mm$$

Step 7: To find b, d and v

Face width, b = 10 m = 80 mm

Pitch circle diameter of the gear,

$$d_1 = m \times z_1 = 8 \times 18 = 144\, mm$$

Pitch line velocity, v

$$v = 1.32\, m = 1.32 \times 8 = 6.03\, m/s$$

Step 8: Revised beam strength, F_s

$$F_s = \pi\, m\, b\, [\sigma_b]\, y = 181.73\, m^2 = 181.73 \times 8^2 = 37.4 \times 10^3, \quad N$$

Step 9: To find accurate dynamic load, F_d

It is calculated using the Buckingham's equation,

$$F_d = F_t + F_I = 7.46 \times 10^3 + 25.46 \times 10^3 = 32.92 \times 10^3, \quad N$$

Where,

Tangential load, F_t

$$F_t = \frac{P}{v} = \frac{45 \times 10^3}{6.03} = 7.46 \times 10^3, \quad N$$

Incremental load, F_I

$$F_I = \frac{21\ v\ (b\ c + F_t)}{21\ v + \sqrt{b\ c + F_t}} = \frac{21 \times 6.03\ (80 \times 225.34 + 7.46 \times 10^3)}{21 \times 6.03 + \sqrt{80 \times 225.34 + 7.46 \times 10^3}}$$

$$F_t = 25.46 \times 10^3, \quad N$$

Where,

c – Deformation factor taken from PSGDB pg. no. 8.53.

$$c = 11860\ e\ for\ steel\ and\ steel\ combination,$$

$$The\ expected\ error, 'e'\ for\ module\ of\ 8\ mm\ and\ precision\ gears\ is\ 0.019$$

$$c = 11860 \times 0.019 = 225.34$$

Step 10: Check for beam strength

$$The\ beam\ strength, F_s > Dynamics\ load, F_d$$

Hence design is safe

Step 11: To find maximum wear load, F_w

$$F_w = d_1 \times b \times Q \times K_w = 144 \times 80 \times 1.56 \times 2.553 = 45.7 \times 10^3, \quad N$$

Where, Q – ratio factor

$$Q = \frac{2\ i}{i + 1} = \frac{2 \times 3.5}{3.5 + 1} = 1.56$$

K_w – load stress factor based on the gear materials = 2.553 for CI and CI combination

Step 12: check for wear strength

$$F_w > F_s$$

Hence Design is safe

Step 13: To find the basic dimensions of the pinion and gear:

From PSGDB pg. no. 8.22, the formulas to find the basic dimensions are given

- Center Distance, a

$$a = \frac{m\ (z_1 + z_2)}{2} = \frac{8 \times (18 + 63)}{2} = 324\ mm$$

- Height Factor, f_o = 1 for full depth teeth
- Bottom clearance, c

$$c = 0.25\ m = 0.25 \times 8 = 2\ mm$$

- Tooth depth, h

$$h = 2.25\ m = 2.25 \times 8 = 18\ mm$$

- Pitch circle diameter, d

  $for\ pinion, d_1 = m\, z_1 = 144\ mm$

  $for\ gear, d_2 = m\, z_2 = 504\ mm$

- Tip diameter, da

  $for\ pinion, d_{a1} = (z_1 + 2f_o)\, m = 160\ mm$

  $for\ gear, d_{a2} = (z_2 + 2f_o)\, m = 520\ mm$

- Root diameter, df

  $for\ pinion, d_{f1} = (z_1 - 2f_o)\, m - 2c = 124\ mm$

  $for\ gear, d_{f2} = (z_2 - 2f_o)\, m - 2c = 484\ mm$

3. A Bakelite pinion driving a cast iron gear. The pinion rotating at 700 rpm transmits 5 kW to a gear. The velocity ratio is 3, the teeth are 20^o FD, and the load is smooth. Design the spur gear drive. Take the allowable static stress for Bakelite as 40 N/mm^2.

Given Data:

Solution:

Step 1: Selection of material

Step 2: To find z_1 and z_2

Step 3: To find the tangential load on the gear tooth, F_t

$$F_t = \frac{P}{v} \times k_o =$$

Where, P – power transmitted, v – pitch line velocity, m/s;

$$v = \frac{\pi\, d\, N}{60} = \frac{\pi\, m\, z\, N}{60 \times 1000} =$$

Step 4: Calculation of initial dynamic load, F_d

$$F_d = \frac{F_t}{c_v} =$$

Where, c_v – velocity factor (from PSGDB pg. no. 8.54)

In general always take that to be,

$$c_v = \frac{6}{6 + v} =$$

Step 5: Calculation of beam strength, F_s

According to Lewis beam strength equation,

$$F_s = \pi \times m \times b \times [\sigma_b] \times y =$$

Where, b – face width in mm, (assume it to be 10 m); $[\sigma_b]$ – allowable static stress of gear material, N/mm^2

y – Form factor, taken from PSGDB pg. no. 8.50 based on the pressure angle.

Step 6: To find module, m

By taking the condition,

$$F_s \geq F_d$$

Find the module, m and select the standard module from the PSGDB pg.no. 8.2.

Step 7: To find b, d and v

Face width, b = 10 m

Pitch circle diameter of the gear, d = m z

Pitch line velocity, v

$$v = \frac{\pi\ d\ N}{60 \times 1000} =$$

Step 8: Revised beam strength, F_s

$$F_s = \pi\ m\ b\ [\sigma_b]\ y =$$

Step 9: To find accurate dynamic load, F_d

It is calculated using the Buckingham's equation,

$$F_d = F_t + F_I =$$

Where,

Tangential load, F_t

$$F_t = \frac{P}{v} =$$

Incremental load, F_I

$$F_I = \frac{21\ v\ (b\ c + F_t\ )}{21\ v + \sqrt{b\ c + F_t}} =$$

Where,

c – Deformation factor taken from PSGDB pg. no. 8.53.

Step 10: Check for beam strength

Step 11: To find maximum wear load, F_w

$$F_w = d \times b \times Q \times K_w =$$

Where, Q – ratio factor

$$Q = \frac{2\ i}{i + 1} =$$

K_w – load stress factor based on the gear materials.

Step 12: check for wear strength

Step 13: To find the basic dimensions of the pinion and gear:

From PSGDB pg. no. 8.22, the formulas to find the basic dimensions are given

- Center Distance, a

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

- Height Factor, $f_o = 1$ for full depth teeth
- Bottom clearance, c

  $c = 0.25\, m =$
- Tooth depth, h

  $h = 2.25\, m =$
- Pitch circle diameter, d

  $for\ pinion, d_1 = m\, z_1 =$

  $for\ gear, d_2 = m\, z_2 =$
- Tip diameter, da

  $for\ pinion, d_{a1} = (z_1 + 2f_o)\, m =$

  $for\ gear, d_{a2} = (z_2 + 2f_o)\, m =$
- Root diameter, df

  $for\ pinion, d_{f1} = (z_1 - 2f_o)\, m - 2c =$

  $for\ gear, d_{f2} = (z_2 - 2f_o)\, m - 2c =$

4. A compressor running at 300 rpm is driven by a 15 kW, 1200 rpm motor through a 14½ ^{0} full depth Spur gears, the centre distance is 375 mm. The motor pinion is to be of C30 forged steel hardened and tempered, and the driven gear is to be of cast iron. Assuming medium shock condition, design the gear drive.

Given Data:

Solution:

Step 1: Selection of material

Step 2: To find z_1 and z_2

Step 4: To find module, m

We know that, Center Distance, a

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

Find the module, m and select the standard module from the PSGDB pg.no. 8.2.

Step 5: To find b, d and v

Face width, b = 10 m =

Pitch circle diameter of the gear, d = m z =

Pitch line velocity, v

$$v = \frac{\pi \, d \, N}{60 \times 1000} =$$

Step 6: Beam strength, F_s

Using Lewis Beam strength equation

$$F_s = \pi \, m \, b \, [\sigma_b] \, y =$$

Step 7: To find accurate dynamic load, F_d

It is calculated using the Buckingham's equation,

$$F_d = F_t + F_I =$$

Where,

Tangential load, F_t

$$F_t = \frac{P}{v} =$$

Incremental load, F_I

$$F_I = \frac{21 \, v \, (b \, c + F_t)}{21 \, v + \sqrt{b \, c + F_t}} =$$

Where,

c – Deformation factor taken from PSGDB pg. no. 8.53.

Step 8: Check for beam strength

Step 9: To find maximum wear load, F_w

$$F_w = d \times b \times Q \times K_w =$$

Where, Q – ratio factor

$$Q = \frac{2\,i}{i+1} =$$

K_w – load stress factor based on the gear materials.

Step 10: check for wear strength

Step 11: To find the basic dimensions of the pinion and gear:

From PSGDB pg. no. 8.22, the formulas to find the basic dimensions are given

- Center Distance, a

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

- Height Factor, f_o = 1 for full depth teeth
- Bottom clearance, c

  $c = 0.25\, m =$

- Tooth depth, h

  $h = 2.25\, m =$

- Pitch circle diameter, d

  $for\ pinion, d_1 = m\, z_1 =$

  $for\ gear, d_2 = m\, z_2 =$

- Tip diameter, da

  $for\ pinion, d_{a1} = (z_1 + 2f_o)\, m =$

  $for\ gear, d_{a2} = (z_2 + 2f_o)\, m =$

- Root diameter, df

  $for\ pinion, d_{f1} = (z_1 - 2f_o)\, m - 2c =$

  $for\ gear, d_{f2} = (z_2 - 2f_o)\, m - 2c =$

5. 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 is 720 N/mm^2 and yield strength is 360 N/ mm^2.

Given Data:

Solution:

Step 1: Selection of material

Step 2: To find z_1 and z_2

Step 3: To find the tangential load on the gear tooth, F_t

$$F_t = \frac{P}{v} \times k_o =$$

Where, P – power transmitted, v – pitch line velocity, m/s;

$$v = \frac{\pi\, d\, N}{60} = \frac{\pi\, m\, z\, N}{60 \times 1000} =$$

Step 4: Calculation of initial dynamic load, F_d

$$F_d = \frac{F_t}{c_v} =$$

Where, c_v – velocity factor (from PSGDB pg. no. 8.54)

In general always take that to be,

$$c_v = \frac{6}{6+v} =$$

Step 5: Calculation of beam strength, F_s

According to Lewis beam strength equation,

$$F_s = \pi \times m \times b \times [\sigma_b] \times y =$$

Where, b – face width in mm, (assume it to be 10 m); $[\sigma_b]$ – allowable static stress of gear material, N/mm^2

y – Form factor, taken from PSGDB pg. no. 8.50 based on the pressure angle.

Step 6: To find module, m

By taking the condition,

$$F_s \geq F_d$$

Find the module, m and select the standard module from the PSGDB pg.no. 8.2.

Step 7: To find b, d and v

Face width, b = 10 m

Pitch circle diameter of the gear, d = m z

Pitch line velocity, v

$$v = \frac{\pi\ d\ N}{60 \times 1000} =$$

Step 8: Revised beam strength, F_s

$$F_s = \pi\, m\, b\, [\sigma_b]\, y =$$

Step 9: To find accurate dynamic load, F_d

It is calculated using the Buckingham's equation,

$$F_d = F_t + F_I =$$

Where,

Tangential load, F_t

$$F_t = \frac{P}{v} =$$

Incremental load, F_I

$$F_I = \frac{21\ v\ (b\ c + F_t\ )}{21\ v + \sqrt{b\ c + F_t}} =$$

Where,

c – Deformation factor taken from PSGDB pg. no. 8.53.

Step 10: Check for beam strength

Step 11: To find maximum wear load, F_w

$$F_w = d \times b \times Q \times K_w =$$

Where, Q – ratio factor

$$Q = \frac{2\,i}{i+1} =$$

K_w – load stress factor based on the gear materials.

Step 12: check for wear strength

Step 13: To find the basic dimensions of the pinion and gear:

From PSGDB pg. no. 8.22, the formulas to find the basic dimensions are given

- Center Distance, a

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

- Height Factor, f_o = 1 for full depth teeth
- Bottom clearance, c

  $c = 0.25\,m =$

- Tooth depth, h

  $h = 2.25\,m =$

- Pitch circle diameter, d

  $for\ pinion, d_1 = m\,z_1 =$

  $for\ gear, d_2 = m\,z_2 =$

- Tip diameter, da

  $for\ pinion, d_{a1} = (z_1 + 2f_o)\,m =$

  $for\ gear, d_{a2} = (z_2 + 2f_o)\,m =$

- Root diameter, df

  $for\ pinion, d_{f1} = (z_1 - 2f_o)\,m - 2c =$

$$for\ gear, d_{f2} = (z_2 - 2f_o)\, m - 2c =$$

Design of Spur gear based on Gear Life:

Design Procedure:

Step 1: To find the Gear ratio, i

$$i = \frac{z_2}{z_1} = \frac{N_1}{N_2}$$

Step 2: Material Selection

From PSGDB pg.no 1.40, based on the gear ratio, the material can be selected for the pinion and gear. Always assume the surface hardness to be greater than 350.

Step 3: To find the gear life in number of cycles, N

$$N = Number\ of\ hours \times 60 \times speed$$

Step 4: To find initial design torque, [Mt]

$$[M_t] = M_t \times k \times k_d, \qquad N.m$$

Where, M_t – Torque; k – Load concentration factor, k_d – Dynamic load factor (Initially assume k x k_d = 1.3)

$$M_t = \frac{P \times 60}{2\,\pi\,N}, \qquad N.m$$

Step 5: To find Equivalent Young's Modulus, Eeq

It is selected from PSGDB pg. no. 8.14 based on the material of the pinion and gear

Step 6: To find Design bending stress, [σb]

From PSGDB pg. no. 8.18

$$[\sigma_b] = \frac{1.4\, k_{bl}}{n\, k_\sigma}\, \sigma_{-1}\ \ N/mm^2$$

Where,

k_{bl} – life factor, (PSGDB pg. no. 8.20)

n – Factor of safety, (PSGDB pg. no. 8.19)

k_σ – Stress concentration factor for fillet, (PSGDB pg. no. 8.19)

σ_{-1} – Endurance limit, (PSGDB pg. no. 8.19)

Step 7: To find the Design contact stress, [σc]

From PSGDB pg. no. 8.16

$$[\sigma_c] = C_R \times HRC \times k_{cl}\ or\ C_B \times HB \times k_{cl}, \qquad N/mm^2$$

C_B or C_R – Coefficient based on material and heat treatment, (PSGDB pg. no. 8.16)

HRC – Rockwell hardness number, (PSGDB pg. no. 8.16)

HB – Brinell hardness number, (PSGDB pg. no. 8.16)

K_{cl} – life factor for surface strength, (PSGDB pg. no. 8.17)

Step 8: To find the center distance, a

From PSGDB pg.no. 8.13

$$a \geq (i+1)\sqrt{\left(\frac{0.74}{[\sigma_c]}\right)^2 \times \frac{E_{eq}[M_t]}{i\,\psi}}$$

Where, ψ = b /d ratio and assume it as 0.3 always

Step 9: To find z_1 and z_2

If number of teeth in the pinion is not given assume them to be, i.e, $z_1 \geq 17$, say 18 teeth

The number of teeth in the gear is given by, gear ratio or velocity ratio or transmission ratio, i

$$i = \frac{z_2}{z_1} = \frac{N_1}{N_2}$$

Step 10: To find module, m

From PSGDB pg. no. 8.22,

$$m = \frac{2\,a}{(z_1 + z_2)}, \quad mm$$

Then select standard module from PSGDB pg. no. 8.2

Step 11: To find revised center distance, a_{rev}

$$a_{rev} = \frac{m\,(z_1 + z_2)}{2}, \qquad mm$$

Step 12: Calculation of b, d, v

Face width, b = 10 m

Pitch circle diameter of the gear, d = m z

Pitch line velocity, v

$$v = \frac{\pi\, d\, N}{60 \times 1000} \quad m/s$$

$$\psi_p = \frac{b}{d}$$

Step 13: Selection of quality of gear

From PSGDB pg. no. 8.3, the gear quality is selected

Step 14: To find revised Design torque

$$[M_t]_{rev} = M_t \times k \times k_d, \qquad N.m$$

Where,

M_t – Torque;

k – Load concentration factor, (PSGDB pg. no. 8.15)

k_d – Dynamic load factor, (PSGDB pg. no. 8.16)

Step 15: Check for bending stress, σ_b

From PSGDB pg. no. 8.13

$$\sigma_b = \frac{i+1}{a\ m\ b\ y} \times [M_t]_{rev}$$

The design is safe only if, $\sigma_b < [\sigma_b]$

Step 16: Check for wear strength, σ_c

From PSGDB pg. no. 8.13

$$\sigma_c = 0.74\ \frac{i+1}{a_{rev}} \sqrt{\frac{i+1}{i\ b}\ E_{eq}[M_t]_{rev}}$$

The design is safe only if, $\sigma_c < [\sigma_c]$

Step 17: To find the basic dimensions of the pinion and gear:

From PSGDB pg. no. 8.22, the formulas to find the basic dimensions are given

- Center Distance, a

$$a = \frac{m\ (z_1 + z_2)}{2}, \qquad mm$$

- Height Factor, $f_o = 1$ for full depth teeth
- Bottom clearance, c

$$c = 0.25\ m\ for\ FD\ teeth$$

- Tooth depth, h

$$h = 2.25\ m\ for\ FD\ teeth$$

- Pitch circle diameter, d

$$for\ pinion, d_1 = m\ z_1$$

$$for\ gear, d_2 = m\ z_2$$

  - Tip diameter, d_a

$$for\ pinion, d_{a1} = (z_1 + 2f_o)\ m$$

$$for\ gear, d_{a2} = (z_2 + 2f_o)\ m$$

  - Root diameter, d_f

$$for\ pinion, d_{f1} = (z_1 - 2f_o)\ m - 2c$$

$$for\ gear, d_{f2} = (z_2 - 2f_o)\ m - 2c$$

Note:

- ***If pinion and gear are made of same material then design pinion alone.***
- ***If pinion and gear are made of different material then material having low static allowable static stress is to be designed.***

PROBLEM:

In a spur gear drive for a stone crusher, the gears are made of C40 steel. The pinion is transmitting 30 kW at 1200 rpm. The gear ratio is 3. Gear is to work for 8 hours per day for 6 days in a week for 3 years. Design the gear drive.

Given Data:

Material – C40 steel, P = 30 kW = 30 x 10^3 N.m/s, N_1 = 1200 rpm, i = 3, 8 hours per day for 6 days in a week for 3 years = 8 x 6 x 52 x 3 = 7488 hours

Step 1: To find the Gear ratio, i

$$i = \frac{z_2}{z_1} = \frac{N_1}{N_2} = 3$$

Step 2: Material Selection

Both pinion and gear are made of same materials C40, pinion is to be designed

Assume the surface hardness to be greater than 350.

Step 3: To find the gear life in number of cycles, N

$$N = Number\ of\ hours \times 60 \times speed = 7488 \times 60 \times 1200 = \mathbf{53.9 \times 10^7\ cycles}$$

Step 4: To find initial design torque, [Mt]

$$[M_t] = M_t \times k \times k_d = 238.73 \times 1.3 = \mathbf{310.34\ N.m}$$

Where, M_t – Torque; k – Load concentration factor, k_d – Dynamic load factor (Initially assume k x k_d = 1.3)

$$M_t = \frac{P \times 60}{2\,\pi\, N} = \frac{30 \times 10^3 \times 60}{2\,\pi \times 1200} = \mathbf{238.73\ N.m}$$

Step 5: To find Equivalent Young's Modulus, Eeq

From PSGDB pg. no. 8.14 for C40 steel,

$$E_{eq} = 2.15 \times 10^5\ N/mm^2$$

Step 6: To find Design bending stress, [σ_b]

From PSGDB pg. no. 8.18

$$[\sigma_b] = \frac{1.4\, k_{bl}}{n\, k_\sigma}\, \sigma_{-1}\ N/mm^2$$

Where,

k_{bl} – life factor, (PSGDB pg. no. 8.20) = 0.7

n – Factor of safety, (PSGDB pg. no. 8.19) = 2

k_σ – Stress concentration factor for fillet, (PSGDB pg. no. 8.19) = 1.5

σ_{-1} – Endurance limit, (PSGDB pg. no. 8.19) = 0.35 σ_u + 120

Step 7: To find the Design contact stress, [σc]

From PSGDB pg. no. 8.16

$$[\sigma_c] = C_R \times HRC \times k_{cl} \; or \; C_B \times HB \times k_{cl}, \qquad N/mm^2$$

C_B or C_R – Coefficient based on material and heat treatment, (PSGDB pg. no. 8.16)

HRC – Rockwell hardness number, (PSGDB pg. no. 8.16)

HB – Brinell hardness number, (PSGDB pg. no. 8.16)

K_{cl} – life factor for surface strength, (PSGDB pg. no. 8.17)

Step 8: To find the center distance, a

From PSGDB pg.no. 8.13

$$a \geq (i+1)\sqrt{\left(\frac{0.74}{[\sigma_c]}\right)^2 \times \frac{E_{eq}[M_t]}{i\,\psi}}$$

Where, ψ = b /d ratio and assume it as 0.3 always

Step 9: To find z_1 and z_2

If number of teeth in the pinion is not given assume them to be, i.e, $z_1 \geq 17$, say 18 teeth

The number of teeth in the gear is given by, gear ratio or velocity ratio or transmission ratio, i

$$i = \frac{z_2}{z_1} = \frac{N_1}{N_2}$$

Step 10: To find module, m

From PSGDB pg. no. 8.22,

$$m = \frac{2\,a}{(z_1 + z_2)}, \quad mm$$

Then select standard module from PSGDB pg. no. 8.2

Step 11: To find revised center distance, a_{rev}

$$a_{rev} = \frac{m\,(z_1 + z_2)}{2}, \qquad mm$$

Step 12: Calculation of b, d, v

Face width, b = 10 m

Pitch circle diameter of the gear, d = m z

Pitch line velocity, v

$$v = \frac{\pi\, d\, N}{60 \times 1000} \quad m/s$$

$$\psi_p = \frac{b}{d}$$

Step 13: Selection of quality of gear

From PSGDB pg. no. 8.3, the gear quality is selected

Step 14: To find revised Design torque

$$[M_t]_{rev} = M_t \times k \times k_d, \qquad N.m$$

Where,

M_t – Torque;

k – Load concentration factor, (PSGDB pg. no. 8.15)

k_d – Dynamic load factor, (PSGDB pg. no. 8.16)

Step 15: Check for bending stress, σ_b

From PSGDB pg. no. 8.13

$$\sigma_b = \frac{i+1}{a\ m\ b\ y} \times [M_t]_{rev}$$

The design is safe only if, $\sigma_b < [\sigma_b]$

Step 16: Check for wear strength, σ_c

From PSGDB pg. no. 8.13

$$\sigma_c = 0.74\ \frac{i+1}{a_{rev}} \sqrt{\frac{i+1}{i\ b}\ E_{eq}[M_t]_{rev}}$$

The design is safe only if, $\sigma_c < [\sigma_c]$

Step 17: To find the basic dimensions of the pinion and gear:

From PSGDB pg. no. 8.22, the formulas to find the basic dimensions are given

- Center Distance, a

$$a = \frac{m\ (z_1 + z_2)}{2}, \qquad mm$$

- Height Factor, f_o = 1 for full depth teeth
- Bottom clearance, c

$$c = 0.25\ m\ for\ FD\ teeth$$

- Tooth depth, h

$$h = 2.25\ m\ for\ FD\ teeth$$

- Pitch circle diameter, d

$$for\ pinion, d_1 = m\ z_1$$
$$for\ gear, d_2 = m\ z_2$$

- Tip diameter, d_a

$$for\ pinion, d_{a1} = (z_1 + 2f_o)\ m$$

$$for\ gear, d_{a2} = (z_2 + 2f_o)\, m$$

- Root diameter, d_f

$$for\ pinion, d_{f1} = (z_1 - 2f_o)\, m - 2c$$

$$for\ gear, d_{f2} = (z_2 - 2f_o)\, m - 2c$$

Helical gear

It is most expensive method of power transmission with heavy load to one shaft to another shaft than spur gear. A simple configuration modified in the helical gear as compared to ordinary spur gear. Helical gear has helix form around the teeth or gear. That is most common gear used in modern industry for automobiles, turbine and highest speed application. It configures the teeth of two wheel are of opposite direction. The helixes may right handed on one wheel and left handed on other wheel.

Advantages of helical gear in modern industries:

There are three main reasons to say helical gears are better than spur gear. 1) Noise 2) load carrying capacity 3) manufacturing method

Noise:

The less noise produced during the operation as compared to spur gear of equivalent in quality because the total contact ratio will be increased. When the engagement of teeth at any time more than one pair of teeth (up to 10 pair of teeth). The load is transmitted for one shaft to another shaft for gradually and uniformly as successive teeth come into engagement. And also, this helical gear operates smoothly.

Load Carrying Capacity of Helical Gear

The total length of line of contact will be increased. When the spur gear having the line of contact parallel to axis of rotation and the total length of line contact is equal to face width. For helical gear, the total length of line contact is diagonal across face of teeth. So, in this case, the line of contact is more than face width in helical gear, which is lower unit loading and increase load carrying capacity.

Manufacturing methods

The Limited number of standard cutters are used to cut the variety of helical gear. That is simply by varying helix angle.

Disadvantages of helical gear teeth

Teeth are inclined to axis of the rotation, so gears are subjected to axial thrust load. The thrust load can be eliminated by using herringbone gear (double helical gear).

Different types of helical gears

Parallel axis helical gear

- The gears are operated in two parallel shafts
- The magnitude of helical angle same for pinion and gear
- They have opposite of helix angle

Crossed helical gear (spiral gear)

- That operate on two non parallel shaft
- Gears having same or opposite hand of helix.

Kinematics of helical gear and Nomenclature

They have view at three ways of teeth on rack.

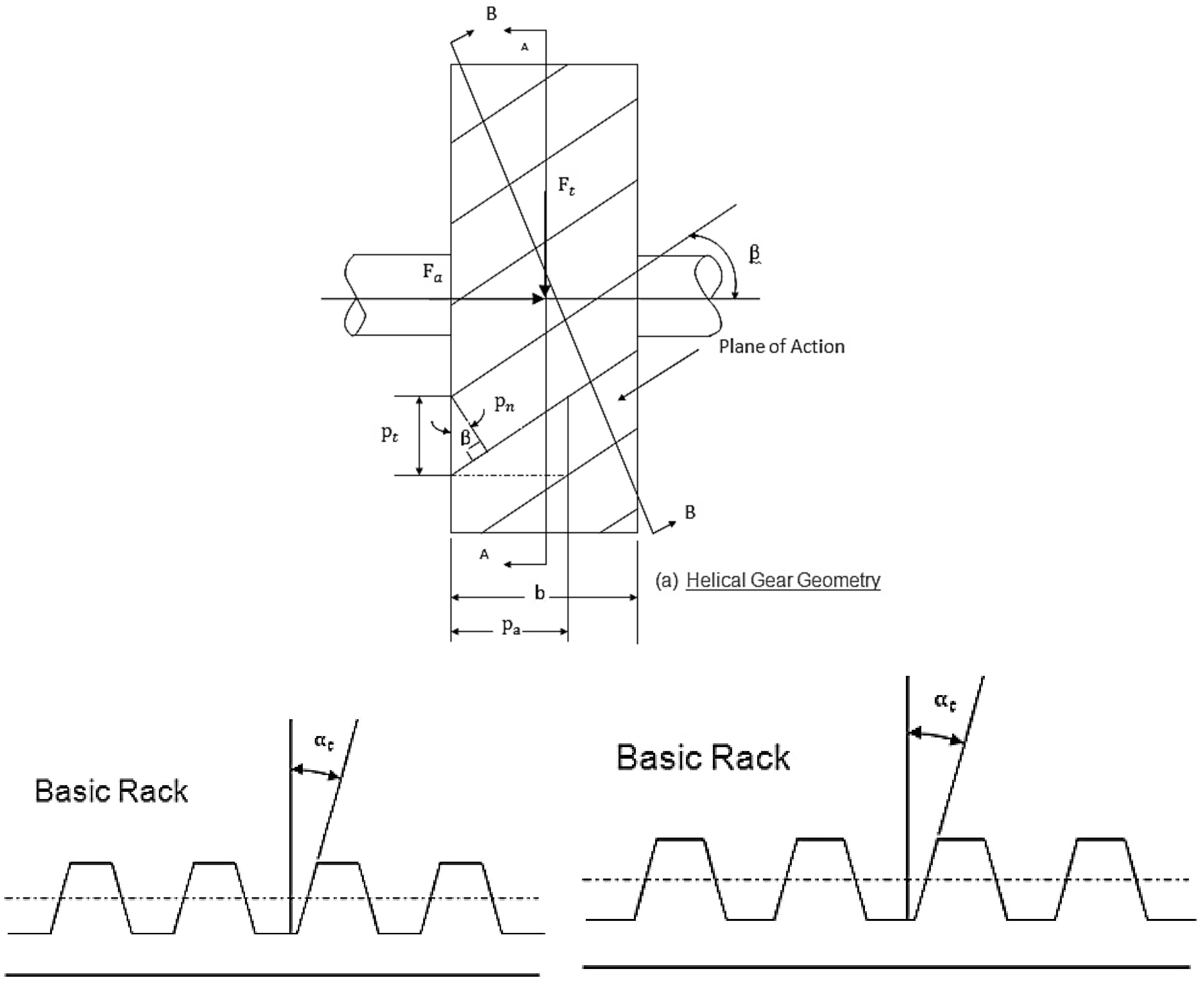


Fig showing section A-A Transverse plane and B-B – Normal direction

Transverse direction

The pair of helical gear view in direction of motion of the rack (section A-A as shown in fig), this orientation is called transverse direction. For this transverse plane(the plane perpendicular to action of the gear), all helical gear geometry is identical to that of spur gear.

Normal direction

If the teeth are consider for direction that aligned with the slant of teeth (section B-B as shown in fig), which orientation is called normal direction.

Axial direction

The teeth are viewed in direction on the axis of the rack, whichorientation is called actual direction.

β = Helix angle

P_t= Transverse circular pitch

P_n= Normal circular pitch

P_a= Axial pitch

P_d= Diametral pitch

$\propto_1$ and $\propto_2$ = Transverse and normal pressure angle

M_t and M_n= Transverse and normal module

Z_1 and Z_2 = Number of teeth on pinion and gear

d_1 and d_2= Pitch circle diameter for pinion and gear respectively

N_1 and N_2= Speed of pinion and gear

a = Standard distance between gear and pinion

Helix angle or spiral angle

It is angle between tooth axis and plane containing the wheel axis. It is known as helix angle. The Helix angle varies from 15° to 25°.

Lead

The distance measured for each teeth per revolution along the axis of parallel to the axis.

Transverse circular pitch

The distance measured by corresponding point on the adjacent teeth in plane perpendicular to the axis of the shaft is known as transverse circular pitch.

$$p_t = \frac{\pi\ d}{z} = \frac{p_n}{\cos\beta}$$

Normal circular pitch

The distance measured by corresponding point on the adjacent teeth in plane perpendicular to helix is known as normal circular pitch.

$$p_n = p_t \times \cos\beta = \pi\ m_n$$

Axial pitch

The distance measured by corresponding point on the adjacent teeth in parallel plane to the axis shaft is known as axial pitch.

$$p_a = \frac{p_t}{\tan\beta} = \frac{p_n}{\sin\beta} = \frac{\pi\ m_n}{\sin\beta}$$

Normal diametral pitch

It is reciprocal of normal module is known as normal diametral pitch.

$$p_d = \frac{1}{m_n} = \frac{\pi}{p_n}$$

Transverse pressure angle(α_t)

The pressure angle measured in transverse plane along the plane (A-A) is known as transverse pressure angle.

Normal pressure angle (α_n)

The pressure angle measured in normal plane along with the plane (B-B) is known as normal pressure angle.

$$\cos\beta = \frac{\tan\alpha_n}{\tan\alpha_t}$$

Pitch circle diameter of pinion and gear

Pitch circle diameter of pinion

$$d_1 = \frac{m_n z_1}{\cos\beta}$$

Pitch circle diameter of gear

$$d_2 = \frac{m_n z_2}{\cos\beta}$$

Centre distance(a)

The centre to Centre distance between gear is called Centre distance.

$$a = \frac{m_n(z_1 + z_2)}{2\cos\beta}$$

Speed ratio(i)

The ratio between speeds of driving gear to driven gear or the ratio between number of teeth of driven gear to driving gear.

$$i = \frac{N_1}{N_2} = \frac{z_2}{z_1}$$

Formative or equivalent number of teeth for helical gear:

The formative or equivalent number of teeth for a helical gear may be defined as the number of teeth that can be generated on the surface of a cylinder having a radius equal to the radius of curvature at a point at the tip of the minor axis of an ellipse obtained by taking a section of the gear in the normal plane. Mathematically, formative or equivalent number of teeth in a helical gear

$$z_v = \frac{z}{\cos^3\beta}$$

z= Actual number of teeth on a helical gear and β = helix angle.

Design of Helical gear based on Beam Strength using Lewis and Buckingham equations:

Design Procedure:

Step 1: Selection of material

From PSGDB pg. no. 1.40, the suitable material for gear and pinion are selected if not given in the problem.

Step 2: To find z_1 and z_2

If number of teeth in the pinion is not given assume them to be, i.e, $z_1 \geq 17$, say 18 teeth

The number of teeth in the gear is given by, gear ratio or velocity ratio or transmission ratio, i

$$i = \frac{z_2}{z_1} = \frac{N_1}{N_2}$$

Where, N_1 – speed of the pinion (small gear), N_2 – sped of the gear (big gear)

Step 3: To find the tangential load on the gear tooth, F_t

$$F_t = \frac{P}{v} \times k_o, \qquad N$$

Where, P – power transmitted,

v – pitch line velocity, m/s;

$$v = \frac{\pi\, d\, N}{60} = \frac{\pi\, m_n\, z\, N}{60 \times 1000}\ m/s$$

k_o – shock factor

Type of load	Shock factor, k_o
Steady	1.0
Light shock	1.25
Medium shock	1.5
Heavy Shock	2.0

Step 4: Calculation of initial dynamic load, F_d

$$F_d = \frac{F_t}{c_v}, \qquad N$$

Where, c_v – velocity factor (from PSGDB pg.no. 8.54)

In general always take that to be,

$$c_v = \frac{6}{6+v}, with\ v\ assumed\ as\ 15\ m/s$$

Step 5: Calculation of beam strength, F_s

According to Lewis beam strength equation,

$$F_s = \pi\, m_n\, b\, [\sigma_b]\, y'$$

Where, b – face width in mm, (assume it to be 10 m); $[\sigma_b]$ – allowable static stress of gear material, N/mm^2, if not given may be taken as $[\sigma_b] = \sigma_u / 3$; y' – Form factor based on virtual number of teeth (z_v), taken from PSGDB pg. no. 8.50 based on the pressure angle.

Step 6: To find normal module, m_n

By taking the condition,

$$F_s \geq F_d$$

Find the normal module, m_n and select the standard module from the PSGDB pg.no. 8.2.

Step 7: To find b, d and v

Face width, $b = 10\ m_n$

Pitch circle diameter of the gear,

$$d = \frac{m_n\, z}{\cos\beta}$$

Pitch line velocity, v

$$v = \frac{\pi\, d\, N}{60 \times 1000}\ m/s$$

Step 8: Revised beam strength, F_s

$$F_s = \pi\, m_n\, b\, [\sigma_b]\, y', \qquad N$$

Step 9: To find accurate dynamic load, F_d

It is calculated using the Buckingham's equation,

$$F_d = F_t + F_I, \qquad N$$

Where,

Tangential load, F_t

$$F_t = \frac{P}{v}, \qquad N$$

Incremental load, F_I

$$F_I = \frac{21\, v\, (b\, c \cos^2\beta + F_t) \cos\beta}{21\, v + \sqrt{b\, c \cos^2\beta + F_t}}, \qquad N$$

Where,

c – Deformation factor taken from PSGDB pg. no. 8.53.

Step 10: Check for beam strength

If $F_s \geq F_d$, design is safe, otherwise increase the face width and module or change the deformation factor value to the precision gears type to fulfill the condition.

Step 11: To find maximum wear load, F_w

$$F_w = \frac{d \times b \times Q \times K_w}{\cos^2 \beta}$$

Where, Q – ratio factor

$$Q = \frac{2\,i}{i + 1}$$

And K_w – load stress factor based on the gear materials.

Step 12: check for wear strength

If $F_w \geq F_d$, design is safe

Step 13: To find the basic dimensions of the pinion and gear:

From PSGDB pg. no. 8.22, the formulas to find the basic dimensions are given

- Center Distance, a

$$a = \frac{m_n(z_1 + z_2)}{2\cos\beta}, \qquad mm$$

- Height Factor, $f_o = 1$ for full depth teeth
- Bottom clearance, c

$$c = 0.25\, m_n\, for\ FD\ teeth$$

- Tooth depth, h

$$h = 2.25\, m_n\, for\ FD\ teeth$$

- Pitch circle diameter, d

$$for\ pinion, d_1 = \frac{m_n z_1}{\cos\beta}$$

$$for\ gear, d_2 = \frac{m_n z_2}{\cos\beta}$$

- Tip diameter, d_a

$$for\ pinion, d_{a1} = \left(\frac{z_1}{\cos\beta} + 2f_o\right) m_n$$

$$for\ gear, d_{a2} = \left(\frac{z_2}{\cos\beta} + 2f_o\right) m_n$$

- Root diameter, d_f

$$for\ pinion, d_{f1} = \left(\frac{z_1}{\cos\beta} - 2f_o\right) m_n - 2c$$

$$for\ gear, d_{f2} = \left(\frac{z_2}{\cos\beta} - 2f_o\right) m_n - 2c$$

- Virtual number of teeth:

$$for\ pinion, z_{v1} = \frac{z_1}{\cos^2\beta}$$

$$for\ gear, z_{v2} = \frac{z_2}{\cos^2\beta}$$

Problem:

1. Design a pair of helical gears for the following data.

Power	=	7.5 kW
Speed of pinion	=	1400 rpm
Speed reduction ratio	=	2
Helix angle	=	10^o
Pressure angle	=	20^o

The gear and pinion are made of forged steel with allowable static stress of 112 N/mm^2.

Given Data:

P = 7.5 kW = 7.5 x 10^3 W, N_1 = 1400 rpm, i = 2, $\beta = 10°$, $\alpha = 20°$ FD, $[\sigma_b]$ = 112 N/mm^2

Solution:

Step 1: Selection of material

Step 2: To find z_1 and z_2

$$z_1 = 20\ to\ be\ assumed$$

$$z_2 = i \times z_1 =$$

Step 3: To find the tangential load on the gear tooth, F_t

$$F_t = \frac{P}{v} \times k_o =$$

Where, P – power transmitted, v – pitch line velocity, m/s; k_o = 1.5 assumed as medium shock

$$v = \frac{\pi\ d_1 N_1}{60} =$$

Step 4: Calculation of initial dynamic load, F_d

$$F_d = \frac{F_t}{c_v} =$$

Where, c_v – velocity factor (from PSGDB pg. no. 8.54)

In general, always take that to be, by assuming v = 12 m/s

$$c_v = \frac{6}{6+v} = \frac{6}{12+6} = 0.333$$

Step 5: Calculation of beam strength, F_s

According to Lewis beam strength equation,

$$F_s = \pi \times m_n \times b \times [\sigma_b] \times y' =$$

Where, b – face width in mm, (assume it to be 10 m_n); $[\sigma_b]$ – allowable static stress of gear material, N/mm^2

y' – Form factor, taken from PSGDB pg. no. 8.50 based on the pressure angle along with the number of teeth to be virtual number of teeth or equivalent number of teeth.

$$z_{v1} = \frac{z_1}{\cos^3 \beta} =$$

$$y = 0.154 - \frac{0.912}{z_{v1}} =$$

Step 6: To find module, m

By taking the condition,

$$F_s \geq F_d$$

Find the module, m and select the standard module from the PSGDB pg.no. 8.2. using choice 2

$$m_n =$$

Step 7: To find b, d and v

Face width, b = 10 m_n =

Pitch circle diameter of the gear,

$$d_1 = \frac{m_n \times z_1}{\cos \beta} =$$

Pitch line velocity, v

$$v = \frac{\pi\, d_1 N_1}{60 \times 1000} =$$

Step 8: Revised beam strength, F_s

$$F_s = \pi \, m_n \, b \, [\sigma_b] y' =$$

Step 9: To find accurate dynamic load, F_d

It is calculated using the Buckingham's equation,

$$F_d = F_t + F_I =$$

Where,

Tangential load, F_t

$$F_t = \frac{P}{v} =$$

Incremental load, F_I

$$F_I = \frac{21\, v\, (b\, c \cos^2 \beta + F_t).\cos \beta}{21\, v + \sqrt{b\, c \cos^2 \beta + F_t}}$$

$$F_I =$$

Where,

c – Deformation factor taken from PSGDB pg. no. 8.53.

$$c = 11860\, e =$$

Step 10: Check for beam strength

$$F_s \geq F_d$$

Condition is satisfied, hence design is safe

Step 11: To find maximum wear load, F_w

$$F_w = d_1 \times b \times Q \times K_w =$$

Where, Q – ratio factor

$$Q = \frac{2\, i}{i + 1} =$$

K_w – load stress factor based on the gear materials. (Assume for forged steel it is 2.4 N/mm^2)

Step 12: check for wear strength

$$F_w \geq F_d$$

Condition is satisfied, hence design is safe

Step 13: To find the basic dimensions of the pinion and gear:

From PSGDB pg. no. 8.22, the formulas to find the basic dimensions are given

- Center Distance, a

$$a = \frac{m_n(z_1 + z_2)}{2 \times \cos\beta} =$$

- Height Factor, f_o = 1 for full depth teeth
- Bottom clearance, c

$$c = 0.25\, m_n =$$

- Tooth depth, h

$$h = 2.25\, m_n =$$

- Pitch circle diameter, d

$$for\ pinion, d_1 = \frac{m_n z_1}{\cos\beta} =$$

$$for\ gear, d_2 = \frac{m_n z_2}{\cos\beta} =$$

- Tip diameter, da

$$for\ pinion, d_{a1} = \left(\frac{z_1}{\cos\beta} + 2f_o\right) m_n =$$

$$for\ gear, d_{a2} = \left(\frac{z_2}{\cos\beta} + 2f_o\right) m_n =$$

- Root diameter, df

$$for\ pinion, d_{f1} = \left(\frac{z_1}{\cos\beta} - 2f_o\right) m_n - 2c =$$

$$forgear, d_{f2} = \left(\frac{z_2}{\cos\beta} - 2f_o\right) m_n - 2c =$$

2. 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 and pressure angles equal to 25° and 20° respectively. The number of teeth on the pinion may be taken as 30. Both the gear and pinion are made of C45 steel. The allowable static stress is 180 N/mm^2, the surface endurance limit is 800 N/mm^2, and Young's modulus of material is 2 x 10^5 N/mm^2.

Given Data:

P = 30 kW = 30 x 10^3 W, N_1 = 2000 rpm, i = 4, β = 25°, α = 20° FD, [σ_b] = 180 N/mm^2, σ_{-1} = 800 N/mm^2, E = 2 x 10^5 N/mm^2

Solution:

Solution:

Step 1: Selection of material

Step 2: To find z_1 and z_2

$$z_1 = 20\ to\ be\ assumed$$

$$z_2 = i \times z_1 =$$

Step 3: To find the tangential load on the gear tooth, F_t

$$F_t = \frac{P}{v} \times k_o =$$

Where, P – power transmitted, v – pitch line velocity, m/s; k_o = 1.5 assumed as medium shock

$$v = \frac{\pi \, d_1 N_1}{60} =$$

Step 4: Calculation of initial dynamic load, F_d

$$F_d = \frac{F_t}{c_v} =$$

Where, c_v – velocity factor (from PSGDB pg. no. 8.54)

In general, always take that to be, by assuming v = 12 m/s

$$c_v = \frac{6}{6+v} = \frac{6}{12+6} = 0.333$$

Step 5: Calculation of beam strength, F_s

According to Lewis beam strength equation,

$$F_s = \pi \times m_n \times b \times \, [\sigma_b] \times y' =$$

Where, b – face width in mm, (assume it to be 10 m_n); $[\sigma_b]$ – allowable static stress of gear material, N/mm^2

y' – Form factor, taken from PSGDB pg. no. 8.50 based on the pressure angle along with the number of teeth to be virtual number of teeth or equivalent number of teeth.

$$z_{v1} = \frac{z_1}{\cos^3 \beta} =$$

$$y = 0.154 - \frac{0.912}{z_{v1}} =$$

Step 6: To find module, m

By taking the condition,

$$F_s \geq F_d$$

Find the module, m and select the standard module from the PSGDB pg.no. 8.2. using choice 2

$$m_n =$$

Step 7: To find b, d and v

Face width, b = 10 m_n =

Pitch circle diameter of the gear,

$$d_1 = \frac{m_n \times z_1}{\cos\beta} =$$

Pitch line velocity, v

$$v = \frac{\pi\, d_1 N_1}{60 \times 1000} =$$

Step 8: Revised beam strength, F_s

$$F_s = \pi\, m_n\, b\, [\sigma_b] y' =$$

Step 9: To find accurate dynamic load, F_d

It is calculated using the Buckingham's equation,

$$F_d = F_t + F_I =$$

Where,

Tangential load, F_t

$$F_t = \frac{P}{v} =$$

Incremental load, F_I

$$F_I = \frac{21\, v\, (b\, c \cos^2\beta + F_t).\cos\beta}{21\, v + \sqrt{b\, c \cos^2\beta + F_t}}$$

$$F_I =$$

Where,

c – Deformation factor taken from PSGDB pg. no. 8.53.

$$c = 11860\ e =$$

Step 10: Check for beam strength

$$F_s \geq F_d$$

Condition is satisfied, hence design is safe

Step 11: To find maximum wear load, F_w

$$F_w = d_1 \times b \times Q \times K_w =$$

Where, Q – ratio factor

$$Q = \frac{2\,i}{i+1} =$$

K_w – load stress factor based on the gear materials. (Assume for forged steel it is 2.4 N/mm^2)

Step 12: check for wear strength

$$F_w \geq F_d$$

Condition is satisfied, hence design is safe

Step 13: To find the basic dimensions of the pinion and gear:

From PSGDB pg. no. 8.22, the formulas to find the basic dimensions are given

- Center Distance, a

  $$a = \frac{m_n(z_1 + z_2)}{2 \times \cos\beta} =$$

- Height Factor, f_o = 1 for full depth teeth
- Bottom clearance, c

  $$c = 0.25\ m_n =$$

- Tooth depth, h

  $$h = 2.25\ m_n =$$

- Pitch circle diameter, d

  $$for\ pinion, d_1 = \frac{m_n z_1}{\cos\beta} =$$

  $$for\ gear, d_2 = \frac{m_n z_2}{\cos\beta} =$$

- Tip diameter, da

  $$for\ pinion, d_{a1} = \left(\frac{z_1}{\cos\beta} + 2f_o\right) m_n =$$

  $$for\ gear, d_{a2} = \left(\frac{z_2}{\cos\beta} + 2f_o\right) m_n =$$

- Root diameter, df

$$for\ pinion, d_{f1} = \left(\frac{z_1}{\cos\beta} - 2f_o\right)m_n - 2c =$$

$$forgear, d_{f2} = \left(\frac{z_2}{\cos\beta} - 2f_o\right)m_n - 2c =$$

3. A pair of helical gears subjected to moderate shock loading is to transmit 37.5kW at 1750 r.p.m. of the pinion. The speed reduction ratio is 4.25 and the helix angle is 15^0. The service is continuous and the teeth are 20^0 FD in the normal plane. Design the gear drive. Assume the pinion and gear is made of cast steel with allowable static stress of 80 N/mm^2.

Given Data:

P = 37.5 kW = 37.5 x 10^3 W, N_1 = 1750 rpm, i = 4.25, β = 15°, α = 20° FD, $[\sigma_b]$ = 80 N/mm^2

Solution:

Step 1: Selection of material

Both pinion and gear are made same material, we should design pinion alone.

Step 2: To find z_1 and z_2

$$z_1 = 20\ to\ be\ assumed$$

$$z_2 = i \times z_1 = 4.25 \times 20 = 85\ teeth$$

Step 3: To find the tangential load on the gear tooth, F_t

$$F_t = \frac{P}{v} \times k_o = \frac{37.5 \times 10^3}{1.897\ m_n} \times 1.5 = \frac{29.65 \times 10^3}{m_n}, \qquad N$$

Where, P – power transmitted, v – pitch line velocity, m/s; k_o = 1.5 assumed as medium shock

$$v = \frac{\pi\ d_1 N_1}{60} = \frac{\pi\ m_n z_1 N_1}{\cos\beta \times 60 \times 1000} = \frac{\pi\ m_n \times 20 \times 1750}{\cos 15° \times 60 \times 1000} = 1.897\ m_n \quad m/s$$

Step 4: Calculation of initial dynamic load, F_d

$$F_d = \frac{F_t}{c_v} = \frac{29.65 \times 10^3}{m_n \times 0.333} = \frac{89.04 \times 10^3}{m_n}, \qquad N$$

Where, c_v – velocity factor (from PSGDB pg. no. 8.54)

In general, always take that to be, by assuming v = 12 m/s

$$c_v = \frac{6}{6 + v} = \frac{6}{12 + 6} = 0.333$$

Step 5: Calculation of beam strength, F_s

According to Lewis beam strength equation,

$$F_s = \pi \times m_n \times b \times [\sigma_b] \times y' = \pi \times m_n \times 10\ m_n \times 80 \times 0.113 = 284\ m_n^2, \qquad N$$

Where, b – face width in mm, (assume it to be 10 m_n); $[\sigma_b]$ – allowable static stress of gear material, N/mm^2

y' – Form factor, taken from PSGDB pg. no. 8.50 based on the pressure angle along with the number of teeth to be virtual number of teeth or equivalent number of teeth.

$$z_{v1} = \frac{z_1}{\cos^3\beta} = \frac{20}{\cos^3 15°} = 22.19 \approx 22\ teeth$$

$$y = 0.154 - \frac{0.912}{z_{v1}} = 0.154 - \frac{0.912}{22} = 0.113$$

Step 6: To find module, m

By taking the condition,

$$F_s \geq F_d$$

$$284 m_n^2 \geq \frac{89.04 \times 10^3}{m_n}$$

$$m_n^3 \geq \frac{89.04 \times 10^3}{284}$$

$$m_n^3 \geq 313.48$$

$$m_n \geq (313.48)^{1/3}$$

$$m_n \geq 6.79\ mm$$

Find the module, m and select the standard module from the PSGDB pg.no. 8.2. using choice 1

$$m_n = 8\ mm$$

Step 7: To find b, d and v

Face width, b = 10 m_n = 10 x 8 = 80 mm

Pitch circle diameter of the gear,

$$d_1 = \frac{m_n \times z_1}{\cos\beta} = \frac{8 \times 20}{\cos 15} = 165.6\ mm \approx 166\ mm$$

Pitch line velocity, v

$$v = \frac{\pi\ d_1 N_1}{60 \times 1000} = \frac{\pi \times 166 \times 1750}{60 \times 1000} = 15.2\ m/s$$

Step 8: Revised beam strength, F_s

$$F_s = \pi\ m_n\ b\ [\sigma_b] y' = \pi \times 8 \times 80 \times 80 \times 0.113 = 28.4 \times 10^3,\ \ N$$

Step 9: To find accurate dynamic load, F_d

It is calculated using the Buckingham's equation,

$$F_d = F_t + F_I = 2.5 \times 10^3 + 18.65 \times 10^3 = 21.25 \times 10^3,\ \ N$$

Where,

Tangential load, F_t

$$F_t = \frac{P}{v} = \frac{37.5 \times 10^3}{15.2} = 2.5 \times 10^3,\ \ N$$

Incremental load, F_I

$$F_I = \frac{21\ v\ (b\ c \cos^2\beta + F_t).\cos\beta}{21\ v + \sqrt{b\ c \cos^2\beta + F_t}}$$

$$F_I = \frac{21 \times\ 15.2 \times 10^3 (80 \times\ 225.34 \times \cos^2 15° + 2.5 \times 10^3).\cos 15°}{21 \times 15.2 \times 10^3 + \sqrt{80 \times\ 225.34 \times \cos^2 15° + 2.5 \times 10^3}} = 18.65 \times 10^3, \qquad N$$

Where,

c – Deformation factor taken from PSGDB pg. no. 8.53.

$$c = 11860\ e = 11860 \times 0.019 = 225.34$$

Step 10: Check for beam strength

$$F_s \geq F_d$$

Condition is satisfied, hence design is safe

Step 11: To find maximum wear load, F_w

$$F_w = d_1 \times b \times Q \times K_w = 166 \times 80 \times 1.61 \times 2.1 = 29.9 \times 10^3, \ \ N$$

Where, Q – ratio factor

$$Q = \frac{2\ i}{i+1} = \frac{2 \times 4}{4+1} = 1.61$$

K_w – load stress factor based on the gear materials. (Assumed to be 1.4 N/mm^2)

Step 12: check for wear strength

$$F_w \geq F_d$$

Condition is satisfied, hence design is safe

Step 13: To find the basic dimensions of the pinion and gear:

From PSGDB pg. no. 8.22, the formulas to find the basic dimensions are given

- Center Distance, a

$$a = \frac{m_n(z_1 + z_2)}{2 \times \cos\beta} = \frac{8\ (20 + 85)}{2 \times \cos 15°} = 434.8\ mm$$

- Height Factor, f_o = 1 for full depth teeth
- Bottom clearance, c

$$c = 0.25\ m_n = 0.25 \times 8 = 2\ mm$$

- Tooth depth, h

$$h = 2.25\ m_n = 2.25 \times 8 = 18\ mm$$

- Pitch circle diameter, d

$$for\ pinion, d_1 = \frac{m_n z_1}{\cos\beta} = 166\ mm$$

$$for\ gear, d_2 = \frac{m_n z_2}{\cos\beta} = \frac{8 \times 85}{\cos 15°} = 703.9 \approx 704\ mm$$

- Tip diameter, da

$$for\ pinion, d_{a1} = \left(\frac{z_1}{\cos\beta} + 2f_o\right) m_n = \left(\frac{20}{\cos 15°} + 2\right) \times 8 = 181.64\ mm$$

$$for\ gear, d_{a2} = \left(\frac{z_2}{\cos\beta} + 2f_o\right) m_n = \left(\frac{85}{\cos 15°} + 2\right) \times 8 = 720\ mm$$

- Root diameter, df

$$for\ pinion, d_{f1} = \left(\frac{z_1}{\cos\beta} - 2f_o\right) m_n - 2c = \left(\frac{20}{\cos 15°} - 2\right) 8 - 2 \times 2 = 145.64\ mm$$

$$forgear, d_{f2} = \left(\frac{z_2}{\cos\beta} - 2f_o\right) m_n - 2c = \left(\frac{85}{\cos 15°} - 2\right) 8 - 2 \times 2 = 684\ mm$$

4. For intermittent duty of an elevator, two cylindrical gears must transmit 12.5 kW at a pinion speed of 1200 rpm. Design the gear pair for the following specifications: Gear ratio 3.5, pressure angle 20°, involute full depth, helix angle 15°. The pinion and gear are made of steel with allowable static stress of 120 N/mm^2.

Given Data:

P = 12.5 kW = 12.5 x 10^3 W, N_1 = 1200 rpm, i = 3.5, β = 15°, α = 20° FD, [σ_b] = 120 N/mm^2

Solution:

Step 1: Selection of material

Both pinion and gear are made same material, we should design pinion alone.

Step 2: To find z_1 and z_2

$$z_1 = 20\ to\ be\ assumed$$

$$z_2 = i \times z_1 = 3.5 \times 20 = 70\ teeth$$

Step 3: To find the tangential load on the gear tooth, F_t

$$F_t = \frac{P}{v} \times k_o = \frac{12.5 \times 10^3}{1.3\, m_n} \times 1.5 = \frac{14.42 \times 10^3}{m_n}, \qquad N$$

Where, P – power transmitted, v – pitch line velocity, m/s; k_o = 1.5 assumed as medium shock

$$v = \frac{\pi\, d_1 N_1}{60} = \frac{\pi\, m_n z_1 N_1}{\cos\beta \times 60 \times 1000} = \frac{\pi\, m_n \times 20 \times 1200}{\cos 15° \times 60 \times 1000} = 1.3\, m_n\ \ m/s$$

Step 4: Calculation of initial dynamic load, F_d

$$F_d = \frac{F_t}{c_v} = \frac{14.42 \times 10^3}{m_n \times 0.333} = \frac{43.3 \times 10^3}{m_n}, \qquad N$$

Where, c_v – velocity factor (from PSGDB pg. no. 8.54)

In general, always take that to be, by assuming v = 12 m/s

$$c_v = \frac{6}{6 + v} = \frac{6}{12 + 6} = 0.333$$

Step 5: Calculation of beam strength, F_s

According to Lewis beam strength equation,

$$F_s = \pi \times m_n \times b \times [\sigma_b] \times y' = \pi \times m_n \times 10\, m_n \times 120 \times 0.113 = 426\, m_n^2, \quad N$$

Where, b – face width in mm, (assume it to be 10 m_n); $[\sigma_b]$ – allowable static stress of gear material, N/mm^2

y' – Form factor, taken from PSGDB pg. no. 8.50 based on the pressure angle along with the number of teeth to be virtual number of teeth or equivalent number of teeth.

$$z_{v1} = \frac{z_1}{\cos^3 \beta} = \frac{20}{\cos^3 15°} = 22.19 \approx 22\, teeth$$

$$y = 0.154 - \frac{0.912}{z_{v1}} = 0.154 - \frac{0.912}{22} = 0.113$$

Step 6: To find module, m

By taking the condition,

$$F_s \geq F_d$$

$$426 m_n^2 \geq \frac{43.3 \times 10^3}{m_n}$$

$$m_n^3 \geq \frac{43.3 \times 10^3}{426}$$

$$m_n^3 \geq 101.64$$

$$m_n \geq (101.64)^{1/3}$$

$$m_n \geq 4.7\, mm$$

Find the module, m and select the standard module from the PSGDB pg.no. 8.2. using choice 1

$$m_n = 5\, mm$$

Step 7: To find b, d and v

Face width, b = 10 m_n = 10 x 5 = 50 mm

Pitch circle diameter of the gear,

$$d_1 = \frac{m_n \times z_1}{\cos \beta} = \frac{5 \times 20}{\cos 15} = 103.5\, mm \approx 104\, mm$$

Pitch line velocity, v

$$v = \frac{\pi\, d_1 N_1}{60 \times 1000} = \frac{\pi \times 104 \times 1200}{60 \times 1000} = 6.54\, m/s$$

Step 8: Revised beam strength, F_s

$$F_s = \pi\, m_n\, b\, [\sigma_b] y' = \pi \times 5 \times 50 \times 120 \times 0.113 = 10.65 \times 10^3, \quad N$$

Step 9: To find accurate dynamic load, F_d

It is calculated using the Buckingham's equation,

$$F_d = F_t + F_I = 1.91 \times 10^3 + 8.52 \times 10^3 = 10.43 \times 10^3, \ N$$

Where,

Tangential load, F_t

$$F_t = \frac{P}{v} = \frac{12.5 \times 10^3}{6.54} = 1.91 \times 10^3, \ N$$

Incremental load, F_I

$$F_I = \frac{21\ v\ (b\ c \cos^2 \beta + F_t). \cos \beta}{21\ v + \sqrt{b\ c \cos^2 \beta + F_t}}$$

$$F_I = \frac{21 \times 6.54 \times 10^3 (50 \times 148.25 \times \cos^2 15° + 1.91 \times 10^3). \cos 15°}{21 \times 6.54 \times 10^3 + \sqrt{50 \times 148.25 \times \cos^2 15° + 1.91 \times 10^3}} = 8.52 \times 10^3, \qquad N$$

Where,

c – Deformation factor taken from PSGDB pg. no. 8.53.

$$c = 11860\ e = 11860 \times 0.0125 = 148.25$$

Step 10: Check for beam strength

$$F_s \geq F_d$$

Condition is satisfied, hence design is safe

Step 11: To find maximum wear load, F_w

$$F_w = d_1 \times b \times Q \times K_w = 104 \times 50 \times 1.6 \times 1.4 = 11.7 \times 10^3, \ N$$

Where, Q – ratio factor

$$Q = \frac{2\ i}{i + 1} = \frac{2 \times 3.5}{3.5 + 1} = 1.6$$

K_w – load stress factor based on the gear materials. (Assumed to be 1.4 N/mm^2)

Step 12: check for wear strength

$$F_w \geq F_d$$

Condition is satisfied, hence design is safe

Step 13: To find the basic dimensions of the pinion and gear:

From PSGDB pg. no. 8.22, the formulas to find the basic dimensions are given

- Center Distance, a

$$a = \frac{m_n (z_1 + z_2)}{2 \times \cos \beta} = \frac{5\ (20 + 70)}{2 \times \cos 15°} = 434.8\ mm$$

- Height Factor, f_o = 1 for full depth teeth
- Bottom clearance, c

$$c = 0.25\ m_n = 0.25 \times 5 = 1.25\ mm$$

- Tooth depth, h

  $$h = 2.25\ m_n = 2.25 \times 5 = 11.25\ mm$$

- Pitch circle diameter, d

  $$for\ pinion, d_1 = \frac{m_n z_1}{\cos\beta} = 104\ mm$$

  $$for\ gear, d_2 = \frac{m_n z_2}{\cos\beta} = \frac{5 \times 70}{\cos 15°} = 362.34\ mm$$

- Tip diameter, da

  $$for\ pinion, d_{a1} = \left(\frac{z_1}{\cos\beta} + 2f_o\right) m_n = \left(\frac{20}{\cos 15°} + 2\right) \times 5 = 113.5\ mm$$

  $$for\ gear, d_{a2} = \left(\frac{z_2}{\cos\beta} + 2f_o\right) m_n = \left(\frac{70}{\cos 15°} + 2\right) \times 5 = 372.34\ mm$$

- Root diameter, df

  $$for\ pinion, d_{f1} = \left(\frac{z_1}{\cos\beta} - 2f_o\right) m_n - 2c = \left(\frac{20}{\cos 15°} - 2\right) 5 - 2 \times 1.25 = 91.03\ mm$$

  $$forgear, d_{f2} = \left(\frac{z_2}{\cos\beta} - 2f_o\right) m_n - 2c = \left(\frac{70}{\cos 15°} - 2\right) 5 - 2 \times 1.25 = 349.85\ mm$$

5. Design a helical gear to transmit 15 kW at 1400 rpm to the following specification: speed reduction ratio 3, pressure angle 20°, helix angle 15°, gear and pinion are made of C40 steel with allowable static stress of 170 N/mm^2, the endurance limit of 780 N/mm^2.

Given Data:

P = 15 kW = 15 x 10^3 W, N_1 = 1400 rpm, i = 3, β = 15°, α = 20° FD, [σ_b] = 170 N/mm^2

Solution:

Step 1: Selection of material

Both pinion and gear are made same material, we should design pinion alone.

Step 2: To find z_1 and z_2

$$z_1 = 20\ to\ be\ assumed$$

$$z_2 = i \times z_1 = 3 \times 20 = 60\ teeth$$

Step 3: To find the tangential load on the gear tooth, F_t

$$F_t = \frac{P}{v} \times k_o = \frac{15 \times 10^3}{1.52\ m_n} \times 1.5 = \frac{14.8 \times 10^3}{m_n}, \qquad N$$

Where, P – power transmitted, v – pitch line velocity, m/s; k_o = 1.5 assumed as medium shock

$$v = \frac{\pi\ d_1 N_1}{60} = \frac{\pi\ m_n z_1 N_1}{\cos\beta \times 60 \times 1000} = \frac{\pi\ m_n \times 20 \times 1400}{\cos 15° \times 60 \times 1000} = 1.52\ m_n \quad m/s$$

Step 4: Calculation of initial dynamic load, F_d

$$F_d = \frac{F_t}{c_v} = \frac{14.8 \times 10^3}{m_n \times 0.333} = \frac{44.4 \times 10^3}{m_n}, \qquad N$$

Where, c_v – velocity factor (from PSGDB pg. no. 8.54)

In general, always take that to be, by assuming v = 12 m/s

$$c_v = \frac{6}{6+v} = \frac{6}{12+6} = 0.333$$

Step 5: Calculation of beam strength, F_s

According to Lewis beam strength equation,

$$F_s = \pi \times m_n \times b \times [\sigma_b] \times y' = \pi \times m_n \times 10\, m_n \times 170 \times 0.113 = 603.5\, m_n^2, \qquad N$$

Where, b – face width in mm, (assume it to be 10 m_n); $[\sigma_b]$ – allowable static stress of gear material, N/mm^2

y' – Form factor, taken from PSGDB pg. no. 8.50 based on the pressure angle along with the number of teeth to be virtual number of teeth or equivalent number of teeth.

$$z_{v1} = \frac{z_1}{\cos^3 \beta} = \frac{20}{\cos^3 15°} = 22.19 \approx 22\ teeth$$

$$y = 0.154 - \frac{0.912}{z_{v1}} = 0.154 - \frac{0.912}{22} = 0.113$$

Step 6: To find module, m

By taking the condition,

$$F_s \geq F_d$$

$$603.5 m_n^2 \geq \frac{44.4 \times 10^3}{m_n}$$

$$m_n^3 \geq \frac{44.4 \times 10^3}{603.5}$$

$$m_n^3 \geq 73.57$$

$$m_n \geq (73.57)^{1/3}$$

$$m_n \geq 4.19\ mm$$

Find the module, m and select the standard module from the PSGDB pg.no. 8.2. using choice 1

$$m_n = 5\ mm$$

Step 7: To find b, d and v

Face width, b = 10 m_n = 10 x 5 = 50 mm

Pitch circle diameter of the gear,

$$d_1 = \frac{m_n \times z_1}{\cos \beta} = \frac{5 \times 20}{\cos 15} = 103.5\ mm \approx 104\ mm$$

Pitch line velocity, v

$$v = \frac{\pi\, d_1 N_1}{60 \times 1000} = \frac{\pi \times 104 \times 1400}{60 \times 1000} = 7.62\ m/s$$

Step 8: Revised beam strength, F_s

$$F_s = \pi\, m_n\, b\, [\sigma_b] y' = \pi \times 5 \times 50 \times 170 \times 0.113 = 15.1 \times 10^3,\quad N$$

Step 9: To find accurate dynamic load, F_d

It is calculated using the Buckingham's equation,

$$F_d = F_t + F_I = \ 1.97 \times 10^3 + 8.58 \times 10^3 = \ 10.55 \times 10^3,\ \ N$$

Where,

Tangential load, F_t

$$F_t = \frac{P}{v} = \frac{15 \times 10^3}{7.62} = 1.97 \times 10^3,\ \ N$$

Incremental load, F_I

$$F_I = \frac{21\ v\ (b\ c \cos^2\beta + F_t).\cos\beta}{21\ v + \sqrt{b\ c \cos^2\beta + F_t}}$$

$$F_I = \frac{21 \times\ 7.62 \times 10^3 (50 \times\ 148.25 \times \cos^2 15° + 1.97 \times 10^3).\cos 15°}{21 \times 7.62 \times 10^3 + \sqrt{50 \times\ 148.25 \times \cos^2 15° + 1.97 \times 10^3}} = 8.58 \times 10^3, \qquad N$$

Where,

c – Deformation factor taken from PSGDB pg. no. 8.53.

$$c = 11860\ e = 11860\ \times 0.0125 = 148.25$$

Step 10: Check for beam strength

$$F_s \geq F_d$$

Condition is satisfied, hence design is safe

Step 11: To find maximum wear load, F_w

$$F_w = d_1 \times b \times Q \times K_w = 104 \times 50 \times 1.5 \times 1.6 = 12.5 \times 10^3,\ \ N$$

Where, Q – ratio factor

$$Q = \frac{2\ i}{i + 1} = \frac{2 \times 3}{3 + 1} = 1.5$$

K_w – load stress factor based on the gear materials. (Assumed to be 1.6 N/mm^2)

Step 12: check for wear strength

$$F_w \geq F_d$$

Condition is satisfied, hence design is safe

Step 13: To find the basic dimensions of the pinion and gear:

From PSGDB pg. no. 8.22, the formulas to find the basic dimensions are given

- Center Distance, a

$$a = \frac{m_n(z_1 + z_2)}{2 \times \cos\beta} = \frac{5\ (20 + 60)}{2 \times \cos 15°} = 434.8\ mm$$

- Height Factor, f_o = 1 for full depth teeth
- Bottom clearance, c

$$c = 0.25\ m_n = 0.25 \times 5 = 1.25\ mm$$

- Tooth depth, h

$$h = 2.25\ m_n = 2.25 \times 5 = 11.25\ mm$$

- Pitch circle diameter, d

$$for\ pinion, d_1 = \frac{m_n z_1}{\cos\beta} = 104\ mm$$

$$for\ gear, d_2 = \frac{m_n z_2}{\cos\beta} = \frac{5 \times 60}{\cos 15°} = 310.6\ mm$$

- Tip diameter, da

$$for\ pinion, d_{a1} = \left(\frac{z_1}{\cos\beta} + 2f_o\right) m_n = \left(\frac{20}{\cos 15°} + 2\right) \times 5 = 113.5\ mm$$

$$for\ gear, d_{a2} = \left(\frac{z_2}{\cos\beta} + 2f_o\right) m_n = \left(\frac{60}{\cos 15°} + 2\right) \times 5 = 320.6\ mm$$

- Root diameter, df

$$for\ pinion, d_{f1} = \left(\frac{z_1}{\cos\beta} - 2f_o\right) m_n - 2c = \left(\frac{20}{\cos 15°} - 2\right)\ 5 - 2 \times 1.25 = 91.03\ mm$$

$$forgear, d_{f2} = \left(\frac{z_2}{\cos\beta} - 2f_o\right) m_n - 2c = \left(\frac{60}{\cos 15°} - 2\right)\ 5 - 2 \times 1.25 = 298.1\ mm$$

MULTIPLE CHOICE QUESTIONS AND ANSWERS

S.NO	QUESTION	OPTION1	OPTION2	OPTION3	OPTION4	ANSWER
1	Involute profile is preferred to cycloidal because…………………	The profile is easy to cut	Only one curve is required to be cut	The rack has straight line profile and hence can be accurately cut	None of these	The rack has straight line profile and hence can be accurately cut
2	The condition of correct gearing is stated as……………	Pitch line velocities of teeth should be the same	Radius of curvature of the two profiles is same	Common normal to the pitch surface should cut the line joining the centers at a fixed point	None of these	Common normal to the pitch surface should cut the line joining the centers at a fixed point
3	Spur gears are used for…………	Connecting skew shafts	Connecting intersecting shafts	Transmitting power from one shaft to another shaft	Connecting two parallel shafts to transmit power	Connecting two parallel shafts to transmit power
4	A rack is gear of ………...	Definite pitch	Infinite module	Infinite diameter	Infinite number of teeth	Infinite diameter
5	Module of the gear is ………...	P.c.d/no.of teeth	No.of teeth/p.c.d	P.c.d X no.of teeth	1/P.c.d	P.c.d/no.of teeth
6	Interference can be avoided in involute gear with 20 degree pressure angle by…..	Cutting involute profile accurately	Using as small number of teeth as possible	Using more than 20 teeth	Using more than 8 teeth	Using more than 20 teeth
7	Which is incorrect relationship of gears?	P X diameter pitch=π	m = p.c.d/ no.of teeth	Dedendum = 1.157m	Addendum= 2.157m	Addendum = 2.157m
8	Stub tooth is …………	Provided on racks only	A tooth of standard profile	Longer than standard tooth	Shorter than standard tooth	Shorter than standard tooth
9	Preliminary design of gears using Lewis equation based on………	Shear stress	Contact stresses	Bending stress	Wear	Bending stress
10	If Fo, Fw, Fd represent beam strength, wear strength and dynamic load the condition for safe design of gear is………...	Fo> Fd >Fw	Fo≥ Fd ≥Fw	Fd> Fo> Fw	Fd≥ Fo≥ Fw	Fd> Fo> Fw

11	Wear strength of gear can be improved by ……………..	Increasing B.H.N	Increasing endurance limit	Increasing S_y	Increasing compressive strength	Increasing B.H.N
12	Velocity factor is used to take care of………..	Effect of high velocity	Possibility of fatigue failure	Possibility of high wear	Pitting	Possibility of fatigue failure
13	Material combination factor is used for finding………..	Beam strength	Dynamical load	Wear strength	Heat capacity	wear strength
14	Out of the pinion and gear design should be made of the gear for which…….	Lewis factor y is smaller	Bending stresses σb> is smaller	σy is smaller	σy is bigger	σy is smaller
15	Dynamic tooth load depends on …………..	Pitch line velocity	Misalignment of shafts	Inaccuracy in tooth profile ‘	Pressure angle	inaccuracy in tooth profile ‘
16	The expression used for the wear strength of the gear is……………	Fu= Dg *b * Ku*Q	Fu= Dg *b * Ku*Q	Fu= F	None of these	Fu= Dg *b * Ku*Q
17	Spur gears are used for gear ratios upto……….	6	2	10	20	6
18	For 50mm diameter gear of involute 20 degree teeth of the interference will occur is module is ………………..	1.5mm	2.0mm	3.0mm	4.0mm	4.0mm
19	If the dynamic tooth load is not within the limit it is advisable for making design suffer to……………	Reduce the module	Reduce the face width	Reduce the error	Reduce the hardness	Reduce the error
20	If two pairs of spur gears are used for speed reductions from 1800 r.p.m to 200r.p.m the speed of the 2^{nd} pinion for compact gear box should be………………	900r.p.m	600r.p.m	450r.p.m	800 r.p.m	600r.p.m
21	With the point of view of wear strength 20degree pressure angle is…………..	Superior to 14 ½ degree	Inferior to 14 ½ degree	As good as 14 ½ degree	None of the above	superior to 14 ½ degree
22	In an involute gear, the base circle must be	At root circle	Above root circle	Under root circle	Above pitch circle	At root circle
23	For the spur gear, the product of the circular pitch and diametral pitch is equal to	Unity	1/π	π	Module	π
24	By which type of teeth variation in centre distance within limit does not affect the velocity ratio of the mating gears	Cycloidal	Involute	Hypoid	None of these	Involute
25	Lewis equation in gears is used to find the	Tensile stress	Fatigue stress	Contact stress	Bending stress	Bending stress
26	Low pressure angle gears have	Stronger teerh	Weaker teeth	No effect on strength	None of the above	Weaker teeth
27	Interference is inherently absent in the following type of gears	Involute	Cycloidal	Stub	Hypocycloidal	Cycloidal
28	In helical gears, the right-hand helix gears mesh with	Left-hand helix	Right-hand helix	Left-hand helix & Right-hand helix	Zero helix	Left-hand helix

29	If both pinion and gear are made of same material, then the load transmission capacity is decided by…………………	The gear	The pinion	The gear & The pinion	None of these	The pinion
30	The surface hardness of the gear material is helpful in	Static mode of design	Dynamic mode of design	Wear mode of design	all of these	Wear mode of design
31	The initial contact in a helical gear is…………	A point	A line	A surface	Unpredictabl e	A point
32	The pressure angle recommended by BIS for gear is………	14.5 degree	20 degree	25 degree	30 degree	20 degree
33	Diameter quotient is defined as …………	Axial module/ reference diameter	Pitch dia / module	Module / pitch dia	Pitch / pitch dia	Pitch dia / module
34	In the miter bevel gear set……………	The axis of gears is at 90 degree	The gears are of the same size	The axis of gears is at 90 degree & The gears are of the same size	The axis of gears are at more than 90 degree	The axis of gears is at 90 degree & The gears are of the same size
35	The size of gear is specified by……….	Addendum circle diameter	Pitch circle diameter	Dedendum circle diameter	Base circle diameter	Pitch circle diameter
36	In spur gears the circle from which the involute profile is generated is called	Pitch circle	Clearance circle	Base circle	Addendum circle	Base circle
37	Larger pressure angle in comparison to smaller pressure angle make the gear	Weaker	Stronger	Weaker & Stronger	None of these	Stronger
38	Stub tooth form is ………. than the full depth tooth form	Weaker	Stronger	Weaker & Stronger	None of these	Stronger
39	In involute gears the pressure angle is…………	Dependent on the size of the teeth	Dependent on the size of the gear	Always constant	Always variable	Always constant
40	In which type of teeth is variation in center distance, within certain limits, does not affect he velocity ratio of mating gears………	Cycloidal	Involutes	Hypoid	All of these	Involutes
41	Interference is inherently absent in the following types of gears…….	Involutes	Stub	Cycloidal	Epicycloids	Cycloidal
42	If both the pinion and gear are made of the same material, then the power transmitting capacity is decided by…………	Gear	Pinion	Pinion or gear	Both pinion and gear	Pinion
43	To prevent tooth breakage, beam strength should be………. than dynamic load	More	Less	Equal	None of these	More
44	For the possible wear to be permitted in a gear drive…………	Wear load should be greater than the dynamic load	Wear load should be less than the dynamic load	Wear load should be greater than the beam strength	Wear load should be greater than tangential tooth	Wear load should be greater than the dynamic load

45	The role of hunting tooth in gear drive is that it……	Increases efficiency of transmission	Distributes wear uniformly	Reduce vibration	Permit stable operation	Distributes wear uniformly
46	The transverse section of a helical gear is identical to……….	Bevel gear	Spur gear	Worm gear	None of these	Spur gear
47	In parallel helical gears the right hand helix will mesh with………...	Right hand helix	Left hand helix	Both of the above	None of these	Left hand helix
48	Zero axial thrust is experienced in …………..	Helical gears	Bevel gears	Spiral gears	Herring bone gears	Herring bone gears
49	Helical gears are used for ………...	External meshing only	Either external or internal meshing	Internal meshing only	None of these	External meshing only
50	An imaginary circle, by pure rolling action gives the same motion as the actual gear, is known as	Pitch circle	Clearance circle	Addendum circle	Dedendum circle	Pitch circle
51	Surface of the tooth above the pitch surface, is known as	Addendum	Dedendum	Flank	Face	Addendum
52	Radial distance from the pitch circle to the top of a tooth, is known as	Addendum	Dedendum	Flank	Face	Addendum
53	Radial distance from the pitch circle to the bottom of a tooth, is known as	Addendum	Dedendum	Flank	Face	Dedendum
54	Surface of the tooth below the pitch circle, is known as	Addendum	Dedendum	Flank	Face	Dedendum
55	Radial distance from the addendum to the clearance circle, is known as	Clearance	Working depth	Flank	Module	Clearance
56	Size of the gear generally specified by	Pressure angle	Pitch circle diameter	Circular pitch	Clearance angle	Pressure angle
57	Thickness of gear tooth is measured along the	Pitch circle	Root circle	Circular pitch	Dedendum circle	Pitch circle
58	Ratio of pitch circle diameter in millimeter to the number of teeth, is known as	Diametral pitch	Circular pitch	Module	Clearance	Diametral pitch

UNIT III

DESIGN OF BEVEL AND WORM GEARS

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.

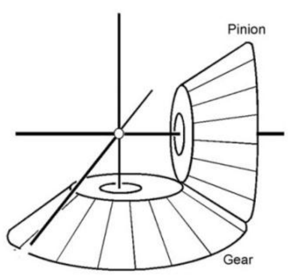
Bevel gears are used to transmit power between two intersecting shafts. There are two types of bevel gears - Straight and spiral. In case of straight bevel gears, the teeth are straight, which converge into a common apex. In case of spiral bevel gears, the teeth are curved. Straight bevel gears are easy to design and manufacture. These gears produce noise at high speed conditions. Spiral bevel gears are difficult to design and manufacture. These gears facilitate quiet operations, even at high speeds. In general, the angle between the axes of intersecting shafts is 90°. i.e., the axes of the shafts are at right angles.

TYPES OF BEVEL GEARS:

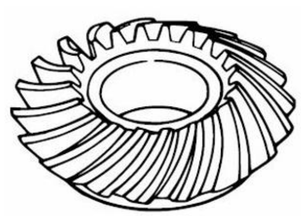
(i) Classification based on teeth shape:

Straight Bevel Gears

The gear are placed on parallel to line generating the pitch code, in this type of gear called as straight bevel gears.

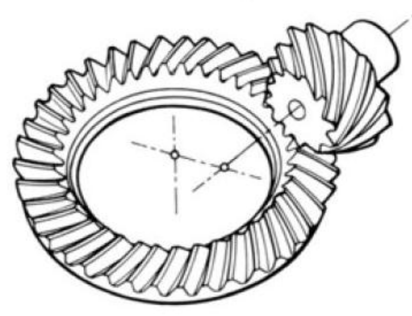


Straight Bevel Gears



Spiral Bevel Gears

Zerol Bevel Gears



Hypoid Bevel Gear

The teeth are straight or radial to the point of intersection of the shaft axes and varying cross section throughout the length. There are used to connect the shaft at right angle, which is run at low speed.

Spiral bevel gears:

The bevel gear teeth are inclined at an angle to face of bevel, there are known as spiral bevel gears. Helical type of gear used in bevel gears, Because of Which is used to reduce the noise during the operation. so provide smooth operation and quick than straight teeth bevel gears, also used to gradual load application and low impact stresses.

The axial thrust is exited in spiral bevel gears, so it requires strong bearing and supporting assembly. These types of bevel gears are used for drive to the differential shaft of an automobiles.

Zerol bevel gear:

The spiral bevel gear having curved teeth, but it should be having zero-degree spiral angle is known as zerol bevel gear. The action of teeth and trust are same as straight bevel gear. The zero-bevel gear are quicker in action than straight bevel gear, because the teeth are curved form.

Hypoid bevel gear or hypoid gear:

It is similar to spiral bevel gears, but smaller difference from spiral gear is axis pinion is offset from axis of gear and the surface of pitch are hyperboloids rather than cone of gear.

The hypoid gear used to most of application involving large speed reduction ratios. They operate in mostly for smoothly and quietly than spiral bevel gear.

ii) Classification based on pitch angle

Crown gear:

The bevel gears having 90-degree pitch angle and plane for its pitch surface is called as Crown gear.

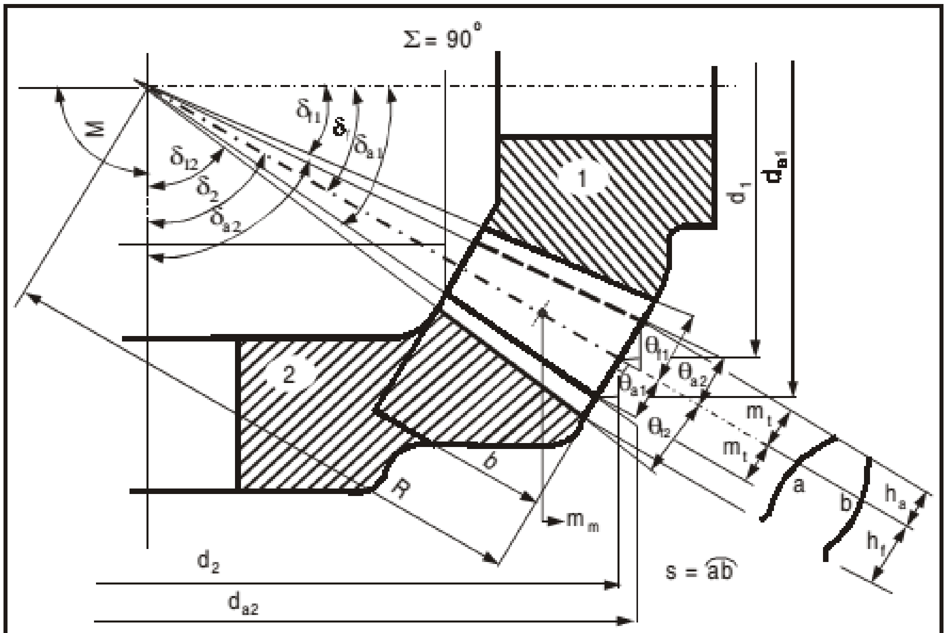
Internal bevel gear:

when the gear having pitch angle of bevel exceeds in 90 degree. This is known as internal bevel gears. Which type of gear manufacturing method is difficult and internal bevel gear are rarely used.

Mitre Gear:

When the messing of two bevel gears having shaft angle of 90 degree and the number of teeth is almost same. This is called as mitre gear. It also having speed ratio is 1. Each two gears have 45 degrees pitch angle.

BEVEL GEAR NOMENCLATURE



Nomenclature of standard bevel Gears

1. Pitch cone: It is an imaginary cone that rolls without slipping on a pitch surface of another gear.

2. Cone Centre: It is the point where the axes of two mating gears intersect each other. It may be also defined as the apex of the pitch cone.

3. Pitch Angle: It is the angle made by the pitch line of a gear with the gear axes. It is denoted by (δ)

4. Cone Distance: It is the length of the pitch cone element. It is also known as pitch cone radius and is denoted by (R).

$$R = 0.5\, m_t \sqrt{z_1^2 + z_2^2}$$

5. Addendum Angle θ(*a*)**:** It is the angle subtended by the addendum of the tooth at the cone centre.

6. Dedendum Angle θ(*f*)**:** It is the angle subtended by the dedendum of the tooth at the cone centre.

7. Tip Angle (Face Angle): It is the angle subtended by the root of the tooth at the cone centre. It is denoted by δ(*a*)

8. Root Angle: It is the angle subtended by the root of the tooth at the cone centre. It is denoted by δ(*f*)

9. Back Cone: It is an imaginary cone, perpendicular to the pitch cone at the end of the tooth. It is also known as normal cone.

10. Back cone distance: It is the length of the back cone. It is also known as back cone radius.

11. Backing: It is the distance of the pitch point from the back of the boss, parallel to the axis of the gear.

12. Mounting height: It is the distance of the back of the boss from the cone centre.

13. Pitch diameter: It is the diameter of the largest pitch circle.

14. Outside (or) addendum cone diameter: It is the maximum diameter of the teeth of the gear. It is equal to the diameter of the blank from which the gear can be cut.

Outside Diameter = Pitch diameter + 2 × $h_a \cos\delta$

15. Inside (or) dedendum cone diameter: Inside diameter = Pitch diameter – $2h_f \times \cos\delta$

Where h_a – Addendum, h_f – Dedendum

FORMATIVE (OR) EQUIVALENT (OR) VIRTUALNUMBER OF TEETH FOR BEVEL GEARS

Bevel gears are replaced by equivalent spur gears, to simplify the design calculation and analysis. An Imaginary spur gear considered in a plane perpendicular to the tooth at the largest end, is called as virtual spur gear. The pitch cone radius 'R' of the bevel gear is equal to the pitch circle radius of the virtual spur gear.

The virtual number of teeth

$$z_{v1} = \frac{z_1}{\cos\delta_1}; \quad z_{v2} = \frac{z_2}{\cos\delta_2}$$

Design of Bevel gear based on Beam Strength using Lewis and Buckingham equations:

Design Procedure:

Step 1: Selection of material

From PSGDB pg. no. 1.40, the suitable material for gear and pinion are selected if not given in the problem.

Step 2: To find z_1 and z_2

If number of teeth in the pinion is not given assume them to be, i.e, $z_1 \geq 17$, say 18 teeth

The number of teeth in the gear is given by, gear ratio or velocity ratio or transmission ratio, i

$$i = \frac{z_2}{z_1} = \frac{N_1}{N_2}$$

Where, N_1 – speed of the pinion (small gear), N_2 – sped of the gear (big gear)

Step 3: To find the pitch angles and virtual number of teeth

Pitch angle for right angle bevel gears are given by

$$\tan\delta_2 = i \quad and \quad \delta_1 = 90° - \delta_2$$

The virtual number of teeth

$$z_{v1} = \frac{z_1}{\cos\delta_1}; \quad z_{v2} = \frac{z_2}{\cos\delta_2}$$

Step 4: To find the tangential load on the gear tooth, F_t

$$F_t = \frac{P}{v} \times k_o, \qquad N$$

Where, P – power transmitted,

v – pitch line velocity, m/s;

$$v = \frac{\pi\, d\, N}{60} = \frac{\pi\, m_t\, z\, N}{60 \times 1000}\ m/s$$

k_o – shock factor

Type of load	Shock factor, k_o
Steady	1.0
Light shock	1.25
Medium shock	1.5
Heavy Shock	2.0

Step 5: Calculation of initial dynamic load, F_d

$$F_d = \frac{F_t}{c_v}, \qquad N$$

Where, c_v – velocity factor (from PSGDB pg.no. 8.54)

In general always take that to be,

$$c_v = \frac{5.6}{5.6 + v}, with\ v\ assumed\ as\ 5\ m/s$$

Step 6: Calculation of beam strength, F_s

According to Lewis beam strength equation,

$$F_s = \pi\, m_t\, b\, [\sigma_b] y' \left(\frac{R - b}{R}\right)$$

Where, b – face width in mm, (assume it to be 10 m_t); $[\sigma_b]$ – allowable static stress of gear material, N/mm^2, if not given may be taken as $[\sigma_b] = \sigma_u / 3$; y' – Form factor based on virtual number of teeth (z_v), taken from PSGDB pg. no. 8.50 based on the pressure angle.

Cone distance,

$$R = 0.5\, m_t \sqrt{z_1^2 + z_2^2}$$

Step 7: To find transverse module, m_t

By taking the condition,

$$F_s \geq F_d$$

Find the transverse module, m_t and select the standard module from the PSGDB pg.no. 8.2.

Step 8: To find b, d and v

Face width, b = 10 m_t

Pitch circle diameter of the gear,

$$d = m_t z$$

Pitch line velocity, v

$$v = \frac{\pi\, d\, N}{60 \times 1000}\ m/s$$

Step 9: Revised beam strength, F_s

$$F_s = \pi\, m_t\, b\, [\sigma_b]\, y' \left(\frac{R-b}{R}\right), \qquad N$$

Step 10: To find accurate dynamic load, F_d

It is calculated using the Buckingham's equation,

$$F_d = F_t + F_I, \qquad N$$

Where,

Tangential load, F_t

$$F_t = \frac{P}{v}, \qquad N$$

Incremental load, F_I

$$F_I = \frac{21\, v\, (b\, c + F_t)}{21\, v + \sqrt{b\, c + F_t}}, \qquad N$$

Where,

c – Deformation factor taken from PSGDB pg. no. 8.53.

Step 11: Check for beam strength

If $F_s \geq F_d$, design is safe, otherwise increase the face width and module or change the deformation factor value to the precision gears type to fulfill the condition.

Step 12: To find maximum wear load, F_w

$$F_w = \frac{d \times b \times Q' \times K_w}{\cos\delta_1}$$

Where, Q – ratio factor

$$Q' = \frac{2\, z_{v2}}{z_{v1} + z_{v2}}$$

And K_w – load stress factor based on the gear materials.

Step 13: check for wear strength

If $F_w \geq F_d$, design is safe

Step 14: To find the basic dimensions of the pinion and gear:

From PSGDB pg. no. 8.22, the formulas to find the basic dimensions are given

- Height Factor, $f_o = 1$ for full depth teeth
- Tip diameter, d_a

$$for\ pinion, d_{a1} = m_t(z_1 + 2\cos\delta_1)$$

$$for\ gear, d_{a2} = m_t(z_2 + 2\cos\delta_2)$$

- Clearance, c

$$c = 0.2$$

- Addendum angle

$$\theta_{a1} = \theta_{a2} = \tan^{-1}\left(\frac{m_t \times f_o}{R}\right)$$

- Dedendum angle

$$\theta_{f1} = \theta_{f2} = \tan^{-1}\left(\frac{m_t \times (f_o + c)}{R}\right)$$

- Tip angle

$$\delta_{a1} = \delta_1 + \theta_{a1}$$

$$\delta_{a2} = \delta_2 + \theta_{a2}$$

- Root angle

$$\delta_{f1} = \delta_1 - \theta_{f1}$$

$$\delta_{f2} = \delta_2 - \theta_{f2}$$

PROBLEMS

1. Design a pair of CI straight bevel gears with the transmission ratio of 2.5. The input is from a 3 kW electric motor running at 960 rpm. Select suitable grade of material for the gears and determine the dimensions.

   Given datas:

   Solution:

   Step 1: Selection of material

   Step 2: To find z_1 and z_2

   If number of teeth in the pinion is not given assume them to be, i.e, $z_1 \geq 17$, say 18 teeth

   The number of teeth in the gear is given by, gear ratio or velocity ratio or transmission ratio, i

$$i = \frac{z_2}{z_1} = \frac{N_1}{N_2}$$

Step 3: To find the pitch angles and virtual number of teeth

Pitch angle for right angle bevel gears are given by

$$\tan\delta_2 = i \quad and \quad \delta_1 = 90° - \delta_2$$

The virtual number of teeth

$$z_{v1} = \frac{z_1}{\cos\delta_1}; \quad z_{v2} = \frac{z_2}{\cos\delta_2}$$

Step 4: To find the tangential load on the gear tooth, F_t

$$F_t = \frac{P}{v} \times k_o, \qquad N$$

$$v = \frac{\pi\, d\, N}{60} = \frac{\pi\, m_t\, z\, N}{60 \times 1000} =$$

Step 5: Calculation of initial dynamic load, F_d

$$F_d = \frac{F_t}{c_v} =$$

Where, c_v – velocity factor (from PSGDB pg.no. 8.54)

In general always take that to be,

$$c_v = \frac{5.6}{5.6 + v}, with\ v\ assumed\ as\ 5\ m/s$$

$$c_v = \frac{5.6}{5.6 + v} =$$

Step 6: Calculation of beam strength, F_s

According to Lewis beam strength equation,

$$F_s = \pi\, m_t\, b\, [\sigma_b] y' \left(\frac{R - b}{R}\right) =$$

Where, b – face width in mm, (assume it to be 10 m_t); $[\sigma_b]$ – allowable static stress of gear material, N/mm^2, if not given may be taken as $[\sigma_b] = \sigma_u / 3$; y' – Form factor based on virtual number of teeth (z_v), taken from PSGDB pg. no. 8.50 based on the pressure angle.

Cone distance,

$$R = 0.5\, m_t \sqrt{z_1^2 + z_2^2} =$$

Step 7: To find transverse module, m_t

By taking the condition,

$$F_s \geq F_d$$

Find the transverse module, m_t and select the standard module from the PSGDB pg.no. 8.2.

Step 8: To find b, d and v

Face width, $b = 10\ m_t$

Pitch circle diameter of the gear,

$$d = m_t z =$$

Pitch line velocity, v

$$v = \frac{\pi\ d\ N}{60 \times 1000} =$$

Step 9: Revised beam strength, F_s

$$F_s = \pi\ m_t\ b\ [\sigma_b]\ y' \left(\frac{R - b}{R}\right) =$$

Step 10: To find accurate dynamic load, F_d

It is calculated using the Buckingham's equation,

$$F_d = F_t + F_I =$$

Where,

Tangential load, F_t

$$F_t = \frac{P}{v} =$$

Incremental load, F_I

$$F_I = \frac{21\,v\,(b\,c + F_t)}{21\,v + \sqrt{b\,c + F_t}} =$$

Where,

c – Deformation factor taken from PSGDB pg. no. 8.53.

Step 11: Check for beam strength

If $F_s \geq F_d$, design is safe, otherwise increase the face width and module or change the deformation factor value to the precision gears type to fulfill the condition.

Step 12: To find maximum wear load, F_w

$$F_w = \frac{d \times b \times Q' \times K_w}{\cos \delta_1} =$$

Where, Q – ratio factor

$$Q' = \frac{2\,z_{v2}}{z_{v1} + z_{v2}} =$$

And K_w – load stress factor based on the gear materials.

Step 13: check for wear strength

If $F_w \geq F_d$, design is safe

Step 14: To find the basic dimensions of the pinion and gear:

From PSGDB pg. no. 8.22, the formulas to find the basic dimensions are given

- Height Factor, $f_o = 1$ for full depth teeth

- Tip diameter, d_a

$$for\ pinion, d_{a1} = m_t(z_1 + 2\cos\delta_1) =$$

$$for\ gear, d_{a2} = m_t(z_2 + 2\cos\delta_2) =$$

- Clearance, c

$$c = 0.2$$

- Addendum angle

$$\theta_{a1} = \theta_{a2} = \tan^{-1}\left(\frac{m_t \times f_o}{R}\right) =$$

- Dedendum angle

$$\theta_{f1} = \theta_{f2} = \tan^{-1}\left(\frac{m_t \times (f_o + c)}{R}\right) =$$

- Tip angle

$$\delta_{a1} = \delta_1 + \theta_{a1} =$$

$$\delta_{a2} = \delta_2 + \theta_{a2} =$$

- Root angle

$$\delta_{f1} = \delta_1 - \theta_{f1} =$$

$$\delta_{f2} = \delta_2 - \theta_{f2} =$$

2. A pair of 20° full depth involute teeth bevel gears connects two shafts at right angles having a velocity ratio of 3:1. The gear is made of cast steel with permissible stress of 10 N/mm². The pinion transmits 37.5 KW at 750 rpm. Determine (1) Module and face width (2) pitch diameters.

Given datas:

Solution:

Step 1: Selection of material

Step 2: To find z_1 and z_2

If number of teeth in the pinion is not given assume them to be, i.e, $z_1 \geq 17$, say 18 teeth

The number of teeth in the gear is given by, gear ratio or velocity ratio or transmission ratio, i

$$i = \frac{z_2}{z_1} = \frac{N_1}{N_2}$$

Step 3: To find the pitch angles and virtual number of teeth

Pitch angle for right angle bevel gears are given by

$$\tan\delta_2 = i \quad and \quad \delta_1 = 90^\circ - \delta_2$$

The virtual number of teeth

$$z_{v1} = \frac{z_1}{\cos\delta_1}; \quad z_{v2} = \frac{z_2}{\cos\delta_2}$$

Step 4: To find the tangential load on the gear tooth, F_t

$$F_t = \frac{P}{v} \times k_o, \qquad N$$

$$v = \frac{\pi\, d\, N}{60} = \frac{\pi\, m_t\, z\, N}{60 \times 1000} =$$

Step 5: Calculation of initial dynamic load, F_d

$$F_d = \frac{F_t}{c_v} =$$

Where, c_v – velocity factor (from PSGDB pg.no. 8.54)

In general always take that to be,

$$c_v = \frac{5.6}{5.6 + v}, with\ v\ assumed\ as\ 5\ m/s$$

$$c_v = \frac{5.6}{5.6 + v} =$$

Step 6: Calculation of beam strength, F_s

According to Lewis beam strength equation,

$$F_s = \pi\, m_t\, b\, [\sigma_b] y' \left(\frac{R - b}{R}\right) =$$

Where, b – face width in mm, (assume it to be 10 m_t); $[\sigma_b]$ – allowable static stress of gear material, N/mm^2, if not given may be taken as $[\sigma_b] = \sigma_u / 3$; y' – Form factor based on virtual number of teeth (z_v), taken from PSGDB pg. no. 8.50 based on the pressure angle.

Cone distance,

$$R = 0.5\, m_t \sqrt{z_1^2 + z_2^2} =$$

Step 7: To find transverse module, m_t

By taking the condition,

$$F_s \geq F_d$$

Find the transverse module, m_t and select the standard module from the PSGDB pg.no. 8.2.

Step 8: To find b, d and v

Face width, $b = 10\ m_t$

Pitch circle diameter of the gear,

$$d = m_t z =$$

Pitch line velocity, v

$$v = \frac{\pi\ d\ N}{60 \times 1000} =$$

Step 9: Revised beam strength, F_s

$$F_s = \pi\ m_t\ b\ [\sigma_b]\ y' \left(\frac{R-b}{R}\right) =$$

Step 10: To find accurate dynamic load, F_d

It is calculated using the Buckingham's equation,

$$F_d = F_t + F_I =$$

Where,

Tangential load, F_t

$$F_t = \frac{P}{v} =$$

Incremental load, F_I

$$F_I = \frac{21\, v\, (b\, c + F_t)}{21\, v + \sqrt{b\, c + F_t}} =$$

Where,

c – Deformation factor taken from PSGDB pg. no. 8.53.

Step 11: Check for beam strength

If $F_s \geq F_d$, design is safe, otherwise increase the face width and module or change the deformation factor value to the precision gears type to fulfill the condition.

Step 12: To find maximum wear load, F_w

$$F_w = \frac{d \times b \times Q' \times K_w}{\cos \delta_1} =$$

Where, Q – ratio factor

$$Q' = \frac{2\, z_{v2}}{z_{v1} + z_{v2}} =$$

And K_w – load stress factor based on the gear materials.

Step 13: check for wear strength

If $F_w \geq F_d$, design is safe

Step 14: To find the basic dimensions of the pinion and gear:

From PSGDB pg. no. 8.22, the formulas to find the basic dimensions are given

- Height Factor, $f_o = 1$ for full depth teeth

- Tip diameter, d_a

$$for\ pinion, d_{a1} = m_t(z_1 + 2\cos\delta_1) =$$

$$for\ gear, d_{a2} = m_t(z_2 + 2\cos\delta_2) =$$

- Clearance, c

$$c = 0.2$$

- Addendum angle

$$\theta_{a1} = \theta_{a2} = \tan^{-1}\left(\frac{m_t \times f_o}{R}\right) =$$

- Dedendum angle

$$\theta_{f1} = \theta_{f2} = \tan^{-1}\left(\frac{m_t \times (f_o + c)}{R}\right) =$$

- Tip angle

$$\delta_{a1} = \delta_1 + \theta_{a1} =$$

$$\delta_{a2} = \delta_2 + \theta_{a2} =$$

- Root angle

$$\delta_{f1} = \delta_1 - \theta_{f1} =$$

$$\delta_{f2} = \delta_2 - \theta_{f2} =$$

3. Design a bevel gear drive to transmit 3.5 kW with the following specifications: speed ratio is 4; driving shaft speed = 200 r.p.m.; drive is non-reversible; the teeth are 20° full depths involute; material for pinion is steel; material for wheel is cast iron.

Given datas:

Solution:

Step 1: Selection of material

Step 2: To find z_1 and z_2

If number of teeth in the pinion is not given assume them to be, i.e, $z_1 \geq 17$, say 18 teeth

The number of teeth in the gear is given by, gear ratio or velocity ratio or transmission ratio, i

$$i = \frac{z_2}{z_1} = \frac{N_1}{N_2}$$

Step 3: To find the pitch angles and virtual number of teeth

Pitch angle for right angle bevel gears are given by

$$\tan\delta_2 = i \quad and \quad \delta_1 = 90° - \delta_2$$

The virtual number of teeth

$$z_{v1} = \frac{z_1}{\cos\delta_1}; \quad z_{v2} = \frac{z_2}{\cos\delta_2}$$

Step 4: To find the tangential load on the gear tooth, F_t

$$F_t = \frac{P}{v} \times k_o, \qquad N$$

$$v = \frac{\pi\ d\ N}{60} = \frac{\pi\ m_t\ z\ N}{60 \times 1000} =$$

Step 5: Calculation of initial dynamic load, F_d

$$F_d = \frac{F_t}{c_v} =$$

Where, c_v – velocity factor (from PSGDB pg.no. 8.54)

In general always take that to be,

$$c_v = \frac{5.6}{5.6 + v}, with\ v\ assumed\ as\ 5\ m/s$$

$$c_v = \frac{5.6}{5.6 + v} =$$

Step 6: Calculation of beam strength, F_s

According to Lewis beam strength equation,

$$F_s = \pi\, m_t\, b\, [\sigma_b] y' \left(\frac{R - b}{R}\right) =$$

Where, b – face width in mm, (assume it to be 10 m_t); $[\sigma_b]$ – allowable static stress of gear material, N/mm^2, if not given may be taken as $[\sigma_b] = \sigma_u / 3$; y' – Form factor based on virtual number of teeth (z_v), taken from PSGDB pg. no. 8.50 based on the pressure angle.

Cone distance,

$$R = 0.5\, m_t \sqrt{z_1^2 + z_2^2} =$$

Step 7: To find transverse module, m_t

By taking the condition,

$$F_s \geq F_d$$

Find the transverse module, m_t and select the standard module from the PSGDB pg.no. 8.2.

Step 8: To find b, d and v

Face width, b = 10 m_t

Pitch circle diameter of the gear,

$$d = m_t z =$$

Pitch line velocity, v

$$v = \frac{\pi\, d\, N}{60 \times 1000} =$$

Step 9: Revised beam strength, F_s

$$F_s = \pi\, m_t\, b\, [\sigma_b]\, y' \left(\frac{R - b}{R}\right) =$$

Step 10: To find accurate dynamic load, F_d

It is calculated using the Buckingham's equation,

$$F_d = F_t + F_I =$$

Where,

Tangential load, F_t

$$F_t = \frac{P}{v} =$$

Incremental load, F_I

$$F_I = \frac{21\, v\, (b\, c + F_t)}{21\, v + \sqrt{b\, c + F_t}} =$$

Where,

c – Deformation factor taken from PSGDB pg. no. 8.53.

Step 11: Check for beam strength

If $F_s \geq F_d$, design is safe, otherwise increase the face width and module or change the deformation factor value to the precision gears type to fulfill the condition.

Step 12: To find maximum wear load, F_w

$$F_w = \frac{d \times b \times Q' \times K_w}{\cos \delta_1} =$$

Where, Q – ratio factor

$$Q' = \frac{2\, z_{v2}}{z_{v1} + z_{v2}} =$$

And K_w – load stress factor based on the gear materials.

Step 13: check for wear strength

If $F_w \geq F_d$, design is safe

Step 14: To find the basic dimensions of the pinion and gear:

From PSGDB pg. no. 8.22, the formulas to find the basic dimensions are given

- Height Factor, $f_o = 1$ for full depth teeth

- Tip diameter, d_a

$$for\ pinion, d_{a1} = m_t(z_1 + 2\cos\delta_1) =$$

$$for\ gear, d_{a2} = m_t(z_2 + 2\cos\delta_2) =$$

- Clearance, c

$$c = 0.2$$

- Addendum angle

$$\theta_{a1} = \theta_{a2} = \tan^{-1}\left(\frac{m_t \times f_o}{R}\right) =$$

- Dedendum angle

$$\theta_{f1} = \theta_{f2} = \tan^{-1}\left(\frac{m_t \times (f_o + c)}{R}\right) =$$

- Tip angle

$$\delta_{a1} = \delta_1 + \theta_{a1} =$$

$$\delta_{a2} = \delta_2 + \theta_{a2} =$$

- Root angle

$$\delta_{f1} = \delta_1 - \theta_{f1} =$$

$$\delta_{f2} = \delta_2 - \theta_{f2} =$$

4. Design a pair of bevel gear to transmit 10 kW at a pinion speed of 1440 rpm. Required transmission ratio is 4. Material for gears is 15ni2cr1mo15 steel. The Tooth profiles of the gears are of 20° composite form.

Given datas:

Solution:

Step 1: Selection of material

Step 2: To find z_1 and z_2

If number of teeth in the pinion is not given assume them to be, i.e, $z_1 \geq 17$, say 18 teeth

The number of teeth in the gear is given by, gear ratio or velocity ratio or transmission ratio, i

$$i = \frac{z_2}{z_1} = \frac{N_1}{N_2}$$

Step 3: To find the pitch angles and virtual number of teeth

Pitch angle for right angle bevel gears are given by

$$\tan\delta_2 = i \quad and \quad \delta_1 = 90° - \delta_2$$

The virtual number of teeth

$$z_{v1} = \frac{z_1}{\cos\delta_1}; \quad z_{v2} = \frac{z_2}{\cos\delta_2}$$

Step 4: To find the tangential load on the gear tooth, F_t

$$F_t = \frac{P}{v} \times k_o, \qquad N$$

$$v = \frac{\pi\, d\, N}{60} = \frac{\pi\, m_t\, z\, N}{60 \times 1000} =$$

Step 5: Calculation of initial dynamic load, F_d

$$F_d = \frac{F_t}{c_v} =$$

Where, c_v – velocity factor (from PSGDB pg.no. 8.54)

In general always take that to be,

$$c_v = \frac{5.6}{5.6 + v}, with\ v\ assumed\ as\ 5\ m/s$$

$$c_v = \frac{5.6}{5.6 + v} =$$

Step 6: Calculation of beam strength, F_s

According to Lewis beam strength equation,

$$F_s = \pi\, m_t\, b\, [\sigma_b] y' \left(\frac{R - b}{R}\right) =$$

Where, b – face width in mm, (assume it to be 10 m_t); $[\sigma_b]$ – allowable static stress of gear material, N/mm^2, if not given may be taken as $[\sigma_b] = \sigma_u / 3$; y' – Form factor based on virtual number of teeth (z_v), taken from PSGDB pg. no. 8.50 based on the pressure angle.

Cone distance,

$$R = 0.5\, m_t \sqrt{z_1^2 + z_2^2} =$$

Step 7: To find transverse module, m_t

By taking the condition,

$$F_s \geq F_d$$

Find the transverse module, m_t and select the standard module from the PSGDB pg.no. 8.2.

Step 8: To find b, d and v

Face width, $b = 10\ m_t$

Pitch circle diameter of the gear,

$$d = m_t z =$$

Pitch line velocity, v

$$v = \frac{\pi\ d\ N}{60 \times 1000} =$$

Step 9: Revised beam strength, F_s

$$F_s = \pi\ m_t\ b\ [\sigma_b]\ y' \left(\frac{R - b}{R}\right) =$$

Step 10: To find accurate dynamic load, F_d

It is calculated using the Buckingham's equation,

$$F_d = F_t + F_I =$$

Where,

Tangential load, F_t

$$F_t = \frac{P}{v} =$$

Incremental load, F_I

$$F_I = \frac{21\ v\ (b\ c + F_t)}{21\ v + \sqrt{b\ c + F_t}} =$$

Where,

c – Deformation factor taken from PSGDB pg. no. 8.53.

Step 11: Check for beam strength

If $F_s \geq F_d$, design is safe, otherwise increase the face width and module or change the deformation factor value to the precision gears type to fulfill the condition.

Step 12: To find maximum wear load, F_w

$$F_w = \frac{d \times b \times Q' \times K_w}{\cos \delta_1} =$$

Where, Q – ratio factor

$$Q' = \frac{2\ z_{v2}}{z_{v1} + z_{v2}} =$$

And K_w – load stress factor based on the gear materials.

Step 13: check for wear strength

If $F_w \geq F_d$, design is safe

Step 14: To find the basic dimensions of the pinion and gear:

From PSGDB pg. no. 8.22, the formulas to find the basic dimensions are given

- Height Factor, $f_o = 1$ for full depth teeth

- Tip diameter, d_a

$$for\ pinion, d_{a1} = m_t(z_1 + 2\cos\delta_1) =$$

$$for\ gear, d_{a2} = m_t(z_2 + 2\cos\delta_2) =$$

- Clearance, c

$$c = 0.2$$

- Addendum angle

$$\theta_{a1} = \theta_{a2} = \tan^{-1}\left(\frac{m_t \times f_o}{R}\right) =$$

- Dedendum angle

$$\theta_{f1} = \theta_{f2} = \tan^{-1}\left(\frac{m_t \times (f_o + c)}{R}\right) =$$

- Tip angle

$$\delta_{a1} = \delta_1 + \theta_{a1} =$$

$$\delta_{a2} = \delta_2 + \theta_{a2} =$$

- Root angle

$$\delta_{f1} = \delta_1 - \theta_{f1} =$$

$$\delta_{f2} = \delta_2 - \theta_{f2} =$$

5. Design a right-angle bevel gear drive with speed of pinion shaft is 300 rpm and that of gear shaft is 150 rpm; pinion is to have 20 teeth of in volute profile with module of 20mm and a pressure angle of 20^o and is to be of suitable material. Gears is forged steel having allowable stress of 150 N/mm^2, kW at gear shaft = 56; Assume service factor = 2.

Given datas:

Solution:

Step 1: Selection of material

Step 2: To find z_1 and z_2

If number of teeth in the pinion is not given assume them to be, i.e, $z_1 \geq 17$, say 18 teeth

The number of teeth in the gear is given by, gear ratio or velocity ratio or transmission ratio, i

$$i = \frac{z_2}{z_1} = \frac{N_1}{N_2}$$

Step 3: To find the pitch angles and virtual number of teeth

Pitch angle for right angle bevel gears are given by

$$\tan\delta_2 = i \quad and \quad \delta_1 = 90° - \delta_2$$

The virtual number of teeth

$$z_{v1} = \frac{z_1}{\cos\delta_1}; \quad z_{v2} = \frac{z_2}{\cos\delta_2}$$

Step 4: To find the tangential load on the gear tooth, F_t

$$F_t = \frac{P}{v} \times k_o, \qquad N$$

$$v = \frac{\pi\, d\, N}{60} = \frac{\pi\, m_t\, z\, N}{60 \times 1000} =$$

Step 5: Calculation of initial dynamic load, F_d

$$F_d = \frac{F_t}{c_v} =$$

Where, c_v – velocity factor (from PSGDB pg.no. 8.54)

In general always take that to be,

$$c_v = \frac{5.6}{5.6 + v}, with\ v\ assumed\ as\ 5\ m/s$$

$$c_v = \frac{5.6}{5.6 + v} =$$

Step 6: Calculation of beam strength, F_s

According to Lewis beam strength equation,

$$F_s = \pi\, m_t\, b\, [\sigma_b] y' \left(\frac{R - b}{R}\right) =$$

Where, b – face width in mm, (assume it to be 10 m_t); $[\sigma_b]$ – allowable static stress of gear material, N/mm^2, if not given may be taken as $[\sigma_b] = \sigma_u / 3$; y' – Form factor based on virtual number of teeth (z_v), taken from PSGDB pg. no. 8.50 based on the pressure angle.

Cone distance,

$$R = 0.5\, m_t \sqrt{z_1^2 + z_2^2} =$$

Step 7: To find transverse module, m_t

By taking the condition,

$$F_s \geq F_d$$

Find the transverse module, m_t and select the standard module from the PSGDB pg.no. 8.2.

Step 8: To find b, d and v

Face width, $b = 10\ m_t$

Pitch circle diameter of the gear,

$$d = m_t z =$$

Pitch line velocity, v

$$v = \frac{\pi\ d\ N}{60 \times 1000} =$$

Step 9: Revised beam strength, F_s

$$F_s = \pi\, m_t\, b\, [\sigma_b]\, y' \left(\frac{R - b}{R}\right) =$$

Step 10: To find accurate dynamic load, F_d

It is calculated using the Buckingham's equation,

$$F_d = F_t + F_I =$$

Where,

Tangential load, F_t

$$F_t = \frac{P}{v} =$$

Incremental load, F_I

$$F_I = \frac{21\, v\, (b\, c + F_t)}{21\, v + \sqrt{b\, c + F_t}} =$$

Where,

c – Deformation factor taken from PSGDB pg. no. 8.53.

Step 11: Check for beam strength

If $F_s \geq F_d$, design is safe, otherwise increase the face width and module or change the deformation factor value to the precision gears type to fulfill the condition.

Step 12: To find maximum wear load, F_w

$$F_w = \frac{d \times b \times Q' \times K_w}{\cos \delta_1} =$$

Where, Q – ratio factor

$$Q' = \frac{2\, z_{v2}}{z_{v1} + z_{v2}} =$$

And K_w – load stress factor based on the gear materials.

Step 13: check for wear strength

If $F_w \geq F_d$, design is safe

Step 14: To find the basic dimensions of the pinion and gear:

From PSGDB pg. no. 8.22, the formulas to find the basic dimensions are given

- Height Factor, f_o = 1 for full depth teeth

- Tip diameter, d_a

  $$for\ pinion, d_{a1} = m_t(z_1 + 2\cos\delta_1) =$$

  $$for\ gear, d_{a2} = m_t(z_2 + 2\cos\delta_2) =$$

- Clearance, c

  $$c = 0.2$$

- Addendum angle

  $$\theta_{a1} = \theta_{a2} = \tan^{-1}\left(\frac{m_t \times f_o}{R}\right) =$$

- Dedendum angle

  $$\theta_{f1} = \theta_{f2} = \tan^{-1}\left(\frac{m_t \times (f_o + c)}{R}\right) =$$

- Tip angle

$$\delta_{a1} = \delta_1 + \theta_{a1} =$$

$$\delta_{a2} = \delta_2 + \theta_{a2} =$$

- Root angle

$$\delta_{f1} = \delta_1 - \theta_{f1} =$$

$$\delta_{f2} = \delta_2 - \theta_{f2} =$$

WORM GEAR

When the shafts are non-parallel and non-intersecting, worm and worm wheel drive is used. Worm drive can be treated as screw and nut pair. A segment of nut is coiled to form a wheel. The thread of a worm has trapezoidal profile in the axial direction.

Materials: Since the sliding occurs, the materials used should have low coefficient of friction.

Worm: Worms are made of steel. The threads are grounded and polished to reduce the surface roughness as low as possible.

Worm wheel: Worm wheels are made of bronze and cast iron.

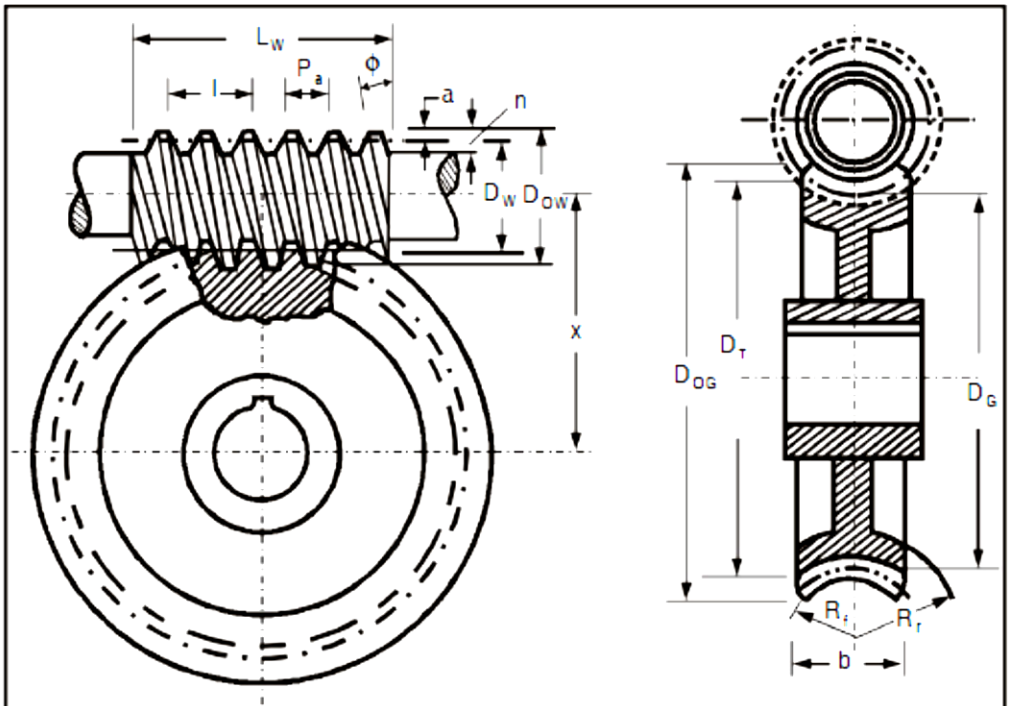


Fig.3.6. Worm and Worm Gear

TERMS USED IN WORM GEARING

1. Axial Pitch: *Pa*: It is the distance measured axially (i.e, parallel to the axis of worm) from a point on one thread to the corresponding point on the adjacent thread on the work

$$P_a = \pi m_x$$

where *mx*= axial module of the worm

2. Lead: It is the linear distance through which a point on a thread moves ahead in one revolution of the worm.

$$Lead = P_a \times z_1$$

where, *Pa* = axial pitch, *Z*1 = number of starts

3. Lead angle, γ: It is the angle between the tangent to the thread helix on the pitch cylinder and the plane normal to the axis of the worm.

$$\gamma = \tan^{-1}\left(\frac{z_1}{q}\right)$$

Where, q – diameter factor

$$q = \frac{d_1}{m_x}$$

4. Tooth Pressure Angle: It is measured in a plane containing the axis of the worm and equal to one half the thread profile angle. The pressure angle of 30° is recommended to obtain a high efficiency and to permit overhauling, for automotive applications.

5.Normal Pitch: The distance measured along the normal to the thread between two corresponding points on two adjacent threads of the worm.

$$P_N = P_a \cos\gamma$$

6. Helix Angle, β: It is the angle between the tangent to the thread helix on the pitch cylinder and the axis of the worm.

$$\beta = 90° - \gamma$$

EFFICIENCY OF WORM GEARING

The ratio of work done by the worm gear to the work done by the worm is known as efficiency of worm gearing.

$$Efficiency, \eta = \frac{\tan\gamma(\cos\alpha - \mu\tan\alpha)}{\cos\alpha\tan\gamma + \mu}$$

Where, α – pressure angle, γ – lead angle, μ – Coefficient of friction

Maximum efficiency occurs when,

$$\tan\gamma = \sqrt{1+\mu^2} - \mu$$

Or for square threads, α = 0 and tan ρ = μ, where ρ = friction angle

$$\eta_{maxi} = \frac{\tan\gamma}{\tan(\gamma+\rho)}$$

MERITS AND DEMERITS OF WORM GEAR DRIVES

Merits of Worm Gear Drives

1. It is used for very high velocity ratio of about 100.
2. Very smooth and noiseless operation.
3. Self-locking facility is available.
4. It is very compact when compared with equivalent spur or helical gears for the same speed reduction.

Demerits of Worm Gear Drives

1. Low Efficiency

2. More heat will be produced and hence this drive can only be operated inside an oil reservoir (or)an extra cooling fan is required in order to dissipate the heat from the drive.

3. Low power transmission (up to 100 kW)

4. Cost is very high, when compared with other gears drive.

Design of worm gear pair based on Beam Strength using Lewis and Buckingham equations:

Design Procedure:

Step 1: Selection of material

From PSGDB pg. no. 1.40, the suitable material for gear and pinion are selected if not given in the problem.

Step 2: To find z_1 and z_2

If number of threads in the worm is selected from PSGDB pg. no. 8.46

The number of teeth in the worm wheel is given by, gear ratio or velocity ratio or transmission ratio, i

$$i = \frac{z_2}{z_1} = \frac{N_1}{N_2}$$

Where, N_1 – speed of the worm, N_2 – speed of the worm wheel

Step 3: To find the diameter factor and lead angle

Diameter factor, q

$$q = \frac{d_1}{m_x}$$

If not given assume them to be q = 11

Lead angle, γ

$$\gamma = \tan^{-1}\left(\frac{z_1}{q}\right)$$

Step 4: To find the tangential load on the gear tooth, F_t

$$F_t = \frac{P}{v} \times k_o, \qquad N$$

Where, P – power transmitted,

v – pitch line velocity, m/s;

$$v = \frac{\pi\, d_2 N_2}{60} = \frac{\pi\, m_x z_2 N_2}{60 \times 1000}\ m/s$$

k_o – shock factor

Type of load	**Shock factor, k_o**
Steady	1.0
Light shock	1.25
Medium shock	1.5
Heavy Shock	2.0

Step 5: Calculation of initial dynamic load, F_d

$$F_d = \frac{F_t}{c_v}, \qquad N$$

Where, c_v – velocity factor (from PSGDB pg.no. 8.54)

In general always take that to be,

$$c_v = \frac{6}{6+v}, with\ v\ assumed\ as\ 5\ m/s$$

Step 6: Calculation of beam strength, F_s

According to Lewis beam strength equation,

$$F_s = \pi\, m_x\, b\, [\sigma_b] y'$$

Where, b – face width in mm, (From PSGDB pg.no. 8.48); $[\sigma_b]$ – allowable static stress of gear material, N/mm^2, if not given may be taken as $[\sigma_b] = \sigma_u / 3$; y' – Form factor taken from PSGDB pg. no. 8.52 based on the pressure angle.

Step 7: To find axial module, m_x

By taking the condition,

$$F_s \geq F_d$$

Find the axial module, m_x and select the standard module from the PSGDB pg.no. 8.2.

Step 8: To find b, d and v

Face width, $b = 10\ m_x$

Pitch circle diameter of the worm gear,

$$d_2 = m_x z_2$$

Pitch line velocity, v

$$v = \frac{\pi\, d_2 N_2}{60 \times 1000}\ m/s$$

Step 9: Revised beam strength, F_s

$$F_s = \pi\, m_x\, b\, [\sigma_b]\, y', \qquad N$$

Step 10: To find accurate dynamic load, F_d

It is calculated using the Buckingham's equation,

$$F_d = \frac{F_t}{c_v}, \qquad N$$

Velocity factor

$$c_v = \frac{6}{6+v}$$

Step 11: Check for beam strength

If $F_s \geq F_d$, design is safe, otherwise increase the face width and module or change the deformation factor value to the precision gears type to fulfill the condition.

Step 12: To find maximum wear load, F_w

$$F_w = d_2 \times b \times k_w$$

And K_w – load stress factor selected from PSGDB pg. no. 8.54.

Step 13: check for wear strength

If $F_w \geq F_d$, design is safe

Step 14: Check for efficiency

$$\eta_{actual} = 0.95 \frac{\tan \gamma}{\tan(\gamma + \rho)}$$

Where,

$$friction\ angle, \rho = \tan^{-1} \mu$$

Where, μ is coefficient of friction (assumed as 0.03)

The actual efficiency calculated should be greater than or equal to the desired efficiency, for safe design

Step 15: To find the basic dimensions of the worm and worm gear:

From PSGDB pg. no. 8.43, the formulas to find the basic dimensions are given

- Center distance

$$a = 0.5\, m_x(q + z_2)$$

- Length of the worm, L (from DB pg.no.8.48)

$$L \geq (11 + 0.06\, z_2)m_x\,, \qquad for\ z_1 = 1\ or\ 2$$

$$L \geq (12.5 + 0.09\, z_2)m_x\,, \qquad for\ z_1 = 3\ or\ 4$$

- Height Factor, $f_o = 1$ for full depth teeth
- Bottom clearance, c

$$c = 0.25\, m_x$$

- Pitch diameter, d

$$d_1 = q\, m_x$$

$$d_2 = z_2 m_x$$

- Tip diameter, d_a

$$for\ worm, d_{a1} = d_1 + 2 f_o m_x$$

$$for\ worm\ gear, d_{a2} = (z_2 + 2\, f_o)\, m_x$$

- Root diameter, d_f

$$for\ worm, d_{f1} = \ d_1 - 2f_o m_x - 2\ c$$

$$for\ worm\ gear, d_{f2} = (z_2 - 2\ f_o)\ m_x - 2\ c$$

Step 16: To find power loss and cooling area required

$$heat\ generated\ (power\ loss) = heat\ emitted\ into\ the\ atmosphere$$

$$(1 - \eta) \times input\ power = K_t\ A\ (t_o - t_a)$$

K_t – overall heat transfer coefficient, W/m^2°C

t_o – temperature of lubricating oil, °C

t_a – temperature of the atmospheric air, °C

A – cooling area, m^2

Problems:

1. A harden steel worm rotates at 1440 rpm and transmit 12 kW to a phosphor bronze gear. Speed of the worm wheel should be 60±3 % rpm. Design the worm gear drive for an efficiency of at least 82%.

   Given Data:

   Solution:

   Step 1: Selection of material

   Step 2: To find z_1 and z_2

   If number of threads in worm is not given, then it is selected from PSGDB pg. no. 8.46

$$z_1 =$$

   The number of teeth in the worm wheel is given by, gear ratio or velocity ratio or transmission ratio, i

$$i = \frac{z_2}{z_1} = \frac{N_1}{N_2}$$

Step 3: To find the diameter factor and lead angle

Diameter factor, q

$$q = \frac{d_1}{m_x} =$$

If not given assume them to be q = 11

Lead angle, γ

$$\gamma = \tan^{-1}\left(\frac{z_1}{q}\right) =$$

Step 4: To find the tangential load on the gear tooth, F_t

$$F_t = \frac{P}{v} \times k_o =$$

Where, P – power transmitted,

v – pitch line velocity, m/s;

$$v = \frac{\pi\, d_2 N_2}{60} = \frac{\pi\, m_x z_2 N_2}{60 \times 1000} =$$

k_o – shock factor =

Step 5: Calculation of initial dynamic load, F_d

$$F_d = \frac{F_t}{c_v} =$$

Where, c_v – velocity factor (from PSGDB pg.no. 8.54)

In general, always take that to be,

$$c_v = \frac{6}{6+v}, with\ v\ assumed\ as\ 5\ m/s$$

Step 6: Calculation of beam strength, F_s

According to Lewis beam strength equation,

$$F_s = \pi\ m_x\ b\ [\sigma_b] y'$$

Where, b – face width in mm, (From PSGDB pg.no. 8.48); $[\sigma_b]$ – allowable static stress of gear material, N/mm^2, if not given may be taken as $[\sigma_b] = \sigma_u / 3$; y' – Form factor taken from PSGDB pg. no. 8.52 based on the pressure angle.

Step 7: To find axial module, m_x

By taking the condition,

$$F_s \geq F_d$$

Find the axial module, m_x and select the standard module from the PSGDB pg.no. 8.2.

Step 8: To find b, d and v

Face width, b = 10 m_x

Pitch circle diameter of the worm gear,

$$d_2 = m_x z_2 =$$

Pitch line velocity, v

$$v = \frac{\pi\, d_2 N_2}{60 \times 1000} =$$

Step 9: Revised beam strength, F_s

$$F_s = \pi\, m_x\, b\,[\sigma_b] y' =$$

Step 10: To find accurate dynamic load, F_d

It is calculated using the Buckingham's equation,

$$F_d = \frac{F_t}{c_v} =$$

Velocity factor

$$c_v = \frac{6}{6 + v} =$$

Step 11: Check for beam strength

If $F_s \geq F_d$, design is safe, otherwise increase the face width and module or change the deformation factor value to the precision gears type to fulfill the condition.

Step 12: To find maximum wear load, F_w

$$F_w = d_2 \times b \times k_w =$$

And K_w – load stress factor selected from PSGDB pg. no. 8.54.

Step 13: check for wear strength

If $F_w \geq F_d$, design is safe

Step 14: Check for efficiency

$$\eta_{actual} = 0.95 \frac{\tan \gamma}{\tan(\gamma + \rho)} =$$

Where,

$$friction\ angle, \rho = \tan^{-1} \mu =$$

Where, μ is coefficient of friction (assumed as 0.03)

The actual efficiency calculated should be greater than or equal to the desired efficiency, for safe design

Step 15: To find the basic dimensions of the worm and worm gear:

From PSGDB pg. no. 8.43, the formulas to find the basic dimensions are given

- Center distance

$$a = 0.5\, m_x (q + z_2)$$

- Length of the worm, L (from DB pg.no.8.48)

$$L \geq (11 + 0.06\, z_2) m_x\,, \qquad for\ z_1 = 1\ or\ 2$$

$$L \geq (12.5 + 0.09\, z_2) m_x\,, \qquad for\ z_1 = 3\ or\ 4$$

- Height Factor, f_o = 1 for full depth teeth

- Bottom clearance, c

$$c = 0.25\, m_x =$$

- Pitch diameter, d

$$d_1 = q\, m_x =$$

$$d_2 = z_2 m_x =$$

- Tip diameter, d_a

$$for\ worm, d_{a1} = d_1 + 2 f_o m_x =$$

$$for\ worm\ gear, d_{a2} = (z_2 + 2\, f_o)\, m_x =$$

- Root diameter, d_f

$$for\ worm, d_{f1} = d_1 - 2 f_o m_x - 2\, c =$$

$$for\ worm\ gear, d_{f2} = (z_2 - 2\ f_o)\ m_x - 2\ c =$$

2. A hardened steel worm rotates at 1440 rpm and transmits 12 kW to a phosphor bronze gear. The speed of the worm wheel should be 60±3% rpm. Design the Worm gear if an efficiency of at least 82% is desired. Also determine the required cooling area if the overall heat transfer coefficient for the housing can be assumed as 15W/m² °C and the temperature rise of the lubricant is restricted to 50 °C.

   Same as previous problem, but it requires this last step

   Step 16: To find power loss and cooling area required

$$heat\ generated\ (power\ loss) = heat\ emitted\ into\ the\ atmosphere$$

$$(1 - \eta) \times input\ power = K_t\ A\ (t_o - t_a)$$

   K_t – overall heat transfer coefficient, W/m²°C

   t_o – temperature of lubricating oil, °C

   t_a – temperature of the atmospheric air, °C

   A – cooling area, m²

3. A steel worm running at 240 rpm receives 1.5 kW from its shaft. The speed reduction is 10:1. Design the drive so as to have an efficiency of 80%. Also Determine cooling area required, if the temperature rise is restricted to 45°C. Take overall heat transfer coefficient as 10 W/m²°C.

   Given Data:

   Solution:

   Step 1: Selection of material

   Step 2: To find z_1 and z_2

   If number of threads in worm is not given, then it is selected from PSGDB pg. no. 8.46

$$z_1 =$$

   The number of teeth in the worm wheel is given by, gear ratio or velocity ratio or transmission ratio, i

$$i = \frac{z_2}{z_1} = \frac{N_1}{N_2}$$

Step 3: To find the diameter factor and lead angle

Diameter factor, q

$$q = \frac{d_1}{m_x} =$$

If not given assume them to be q = 11

Lead angle, γ

$$\gamma = \tan^{-1}\left(\frac{z_1}{q}\right) =$$

Step 4: To find the tangential load on the gear tooth, F_t

$$F_t = \frac{P}{v} \times k_o =$$

Where, P – power transmitted,

v – pitch line velocity, m/s;

$$v = \frac{\pi\, d_2 N_2}{60} = \frac{\pi\, m_x z_2 N_2}{60 \times 1000} =$$

k_o – shock factor =

Step 5: Calculation of initial dynamic load, F_d

$$F_d = \frac{F_t}{c_v} =$$

Where, c_v – velocity factor (from PSGDB pg.no. 8.54)

In general, always take that to be,

$$c_v = \frac{6}{6+v}, with\ v\ assumed\ as\ 5\ m/s$$

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According to Lewis beam strength equation,

$$F_s = \pi\ m_x\ b\ [\sigma_b] y'$$

Where, b – face width in mm, (From PSGDB pg.no. 8.48); $[\sigma_b]$ – allowable static stress of gear material, N/mm^2, if not given may be taken as $[\sigma_b] = \sigma_u / 3$; y' – Form factor taken from PSGDB pg. no. 8.52 based on the pressure angle.

Step 7: To find axial module, m_x

By taking the condition,

$$F_s \geq F_d$$

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Face width, b = 10 m_x

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Pitch line velocity, v

$$v = \frac{\pi\, d_2 N_2}{60 \times 1000} =$$

Step 9: Revised beam strength, F_s

$$F_s = \pi\, m_x\, b\, [\sigma_b] y' =$$

Step 10: To find accurate dynamic load, F_d

It is calculated using the Buckingham's equation,

$$F_d = \frac{F_t}{c_v} =$$

Velocity factor

$$c_v = \frac{6}{6 + v} =$$

Step 11: Check for beam strength

If $F_s \geq F_d$, design is safe, otherwise increase the face width and module or change the deformation factor value to the precision gears type to fulfill the condition.

Step 12: To find maximum wear load, F_w

$$F_w = d_2 \times b \times k_w =$$

And K_w – load stress factor selected from PSGDB pg. no. 8.54.

Step 13: check for wear strength

If $F_w \geq F_d$, design is safe

Step 14: Check for efficiency

$$\eta_{actual} = 0.95 \frac{\tan \gamma}{\tan(\gamma + \rho)} =$$

Where,

$$friction\ angle, \rho = \tan^{-1} \mu =$$

Where, μ is coefficient of friction (assumed as 0.03)

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- Height Factor, f_o = 1 for full depth teeth

- Bottom clearance, c

$c = 0.25\, m_x =$

- Pitch diameter, d

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Step 16: To find power loss and cooling area required

$$heat\ generated\ (power\ loss) = heat\ emitted\ into\ the\ atmosphere$$

$$(1-\eta) \times input\ power = K_t\ A\ (t_o - t_a)$$

K_t – overall heat transfer coefficient, W/m²°C

t_o – temperature of lubricating oil, °C

t_a – temperature of the atmospheric air, °C

A – cooling area, m²

4. Design a worm gear drive to transmit 225 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°C. Determine the required cooling area.

Given Data:

Solution:

Step 1: Selection of material

Step 2: To find z_1 and z_2

If number of threads in worm is not given, then it is selected from PSGDB pg. no. 8.46

$$z_1 =$$

The number of teeth in the worm wheel is given by, gear ratio or velocity ratio or transmission ratio, i

$$i = \frac{z_2}{z_1} = \frac{N_1}{N_2}$$

Step 3: To find the diameter factor and lead angle

Diameter factor, q

$$q = \frac{d_1}{m_x} =$$

If not given assume them to be q = 11

Lead angle, γ

$$\gamma = \tan^{-1}\left(\frac{z_1}{q}\right) =$$

Step 4: To find the tangential load on the gear tooth, F_t

$$F_t = \frac{P}{v} \times k_o =$$

Where, P – power transmitted,

v – pitch line velocity, m/s;

$$v = \frac{\pi\, d_2 N_2}{60} = \frac{\pi\, m_x z_2 N_2}{60 \times 1000} =$$

k_o – shock factor =

Step 5: Calculation of initial dynamic load, F_d

$$F_d = \frac{F_t}{c_v} =$$

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Step 7: To find axial module, m_x

By taking the condition,

$$F_s \geq F_d$$

Find the axial module, m_x and select the standard module from the PSGDB pg.no. 8.2.

Step 8: To find b, d and v

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Pitch circle diameter of the worm gear,

$$d_2 = m_x z_2 =$$

Pitch line velocity, v

$$v = \frac{\pi\, d_2 N_2}{60 \times 1000} =$$

Step 9: Revised beam strength, F_s

$$F_s = \pi\, m_x\, b\,[\sigma_b] y' =$$

Step 10: To find accurate dynamic load, F_d

It is calculated using the Buckingham's equation,

$$F_d = \frac{F_t}{c_v} =$$

Velocity factor

$$c_v = \frac{6}{6 + v} =$$

Step 11: Check for beam strength

If $F_s \geq F_d$, design is safe, otherwise increase the face width and module or change the deformation factor value to the precision gears type to fulfill the condition.

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$$F_w = d_2 \times b \times k_w =$$

And K_w – load stress factor selected from PSGDB pg. no. 8.54.

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$$\eta_{actual} = 0.95 \frac{\tan \gamma}{\tan(\gamma + \rho)} =$$

Where,

$$friction\ angle, \rho = \tan^{-1} \mu =$$

Where, μ is coefficient of friction (assumed as 0.03)

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- Height Factor, f_o = 1 for full depth teeth

- Bottom clearance, c

$$c = 0.25\ m_x =$$

- Pitch diameter, d

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$$for\ worm, d_{a1} = d_1 + 2 f_o m_x =$$

$$for\ worm\ gear, d_{a2} = (z_2 + 2\ f_o)\ m_x =$$

- Root diameter, d_f

$$for\ worm, d_{f1} = \ d_1 - 2 f_o m_x - 2\ c =$$

$$for\ worm\ gear, d_{f2} = (z_2 - 2\ f_o)\ m_x - 2\ c =$$

5. Design a worm gear drive with a standard center distance to transmit 7.5 kW from a worm rotating at1440 rpm to a worm wheel at 20 rpm.

Given Data:

Solution:

Step 1: Selection of material

Step 2: To find z_1 and z_2

If number of threads in worm is not given, then it is selected from PSGDB pg. no. 8.46

$$z_1 =$$

The number of teeth in the worm wheel is given by, gear ratio or velocity ratio or transmission ratio, i

$$i = \frac{z_2}{z_1} = \frac{N_1}{N_2}$$

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k_o – shock factor =

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Step 8: To find b, d and v

Face width, $b = 10\ m_x$

Pitch circle diameter of the worm gear,

$$d_2 = m_x z_2 =$$

Pitch line velocity, v

$$v = \frac{\pi\, d_2 N_2}{60 \times 1000} =$$

Step 9: Revised beam strength, F_s

$$F_s = \pi\, m_x\, b\, [\sigma_b] y' =$$

Step 10: To find accurate dynamic load, F_d

It is calculated using the Buckingham's equation,

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Velocity factor

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$$F_w = d_2 \times b \times k_w =$$

And K_w – load stress factor selected from PSGDB pg. no. 8.54.

Step 13: check for wear strength

If $F_w \geq F_d$, design is safe

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Where,

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- Height Factor, $f_o = 1$ for full depth teeth

- Bottom clearance, c

$$c = 0.25\, m_x =$$

- Pitch diameter, d

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- Root diameter, d_f

$$for\ worm, d_{f1} = d_1 - 2 f_o m_x - 2\, c =$$

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MULTIPLE CHOICE QUESTIONS AND ANSWERS

	QUESTION	OPTION 1	OPTION 2	OPTION 3	OPTION 4	ANSWER
1	Pitch circle diameter is equal to the product of……….	Circular pitch and no. of teeth	Working depth and no. of teeth	Module and no.of teeth	Clearance and no of teeth	Module and no.of teeth
2	If two gears having 24 and 96 teeth mesh each other they must have been cut with……	Same cutter	Cutter of same diameter as that of gear	Cutter of same module	One cutter	Cutter of same module
3	The product of diametral pitch and module is equal to…….	π	1	1/T	None of these	1
4	Backlash is……….	Sum of clearance of two gears	Mutual play between two gears	Amount by which tooth space exceeds the thickness of engaging teeth	None of these	Amount by which tooth space exceeds the thickness of engaging teeth
5	Ratio of dedendum circle diameter and pitch circle diameter is equal to……..	Cos ø	1/cos ø	Sin ø	1/sinø	Cos ø
6	Difference between the tooth space and the tooth thickness, measured on the pitch circle is known as…………	Module	Backlash	Total depth	Working depth	Backlash
7	Angle through which a gear turns from the beginning of contact of a pair of teeth until the contact reaches pit point is known as…………	Angle of contact	Angle of approach	Angle of action	Pressure angle	Angle of approach
8	Ratio of length of arc of contact to the circular pitch is known as…………..	Module	Contact ratio	Backlash	Addendum	Contact ratio
9	Contact ratio means…………..	No.of pairs of teeth in contact	Radial distance from pitch circle to the bottom of the tooth	Radial distance from pitch circle to top of the tooth	None of these	No.of pairs of teeth in contact
10	Contact ratio for gears is………	3	6	12	24	3
11	According to law of gearing the common normal at the point of contact between a pair of teeth must always pass through the…………	Pitch point	Blank of tooth	Face of tooth	None of these	Pitch point
12	Medium velocity gears has their peripheral velocity…………	1 to 3 m/s	3 to 15 m/s	15 to 30 m/s	13 to 50 m/s	3 to 15 m/s
13	The gears used to connect two non intersecting and non-coplanar shafts are known as ………	Spiral gears	Bevel gears	Spur gears	Worm Gears	Spiral gears
14	When the axis of shafts is non intersecting and non-parallel and teeth are straight the gearing is known as…………….	Bevel gears	Skew bevel gearing	Helical bevel gearing	Spur gearing	Skew bevel gearing
15	When the axes of shafts are non-intersecting and non-parallel and teeth are curved the gears are known as………	Bevel gears	Skew bevel gears	Helical bevel gears	Zero bevel gears	Zero bevel gears

16	The gears used to connect two non-parallel or intersecting but coplanar shafts are known as	Spiral gears	Bevel gears	Spur gears	Helical gears	Bevel gears
17	The range of pressure angle for spur gear is	5-15 degrees	15-20 degrees	20-30 degrees	30 -40 degrees	15-20 degrees
18	The face angle of bevel gears equal to……….	Pitch angle + addendum angle	Pitch angle - addendum angle	Pitch angle + dedendum angle	Pitch angle - dedendum angle	Pitch angle + addendum angle
19	Worm gear is used to get speed reduction between shafts while the axes are …….	Parallel	Perpendicular and intersecting	Perpendicular and non intersecting	None of these	Perpendicular and non intersecting
20	Gears used to keep minimum noise are……….	Bevel gears	Helical gears	Spur gears	Spiral gears	Helical gears
21	Gears used for non-intersecting particular shafts are…………	Spur gears	Helical gears	Double helical	Hypoid gears	Spur gears
22	In hypoid gears the axes of the shafts ………	Non- parallel, and non intersecting teeth are curved	Non-parallel, non intersecting teeth are straight	Intersecting and teeth are curved	Intersecting and teeth are straight	Intersecting and teeth are straight
23	Meter gears are…………	Right angled bevel gears with same no. of teeth	Spur gears with the same no.of teeth	Helical gears with same no.of teeth	None of these	Right angled bevel gears with same no. of teeth
24	Worm gears are made of……………	Mild steel	Cast iron	Forged steel	None of these	Mild steel
25	If no.of teeth on two bevel gears in mesh is 25 and 50 then, cone pitch angle of the gear will be……..	$(\pi/2) + \tan^{-1}(0.5)$	$(\pi/2)$ - tan-1(0.5)	$(\pi/4) + \tan^{-1}(0.5)$	$(\pi/4) - \tan^{-1}(0.5)$	$(\pi/2) - \tan^{-1}(0.5)$
26	If no.of teeth on two bevel gears in mesh is 25 and 50 then, cone pitch angle of the pinion will be…………	$\sin^{-1}(0.5)$	$\cos^{-1}(0.5)$	$\tan^{-1}(0.5)$	$\tan^{-1}(0.2)$	$\sin^{-1}(0.5)$
27	When the lead angle of a worm is 22.5degree then the helix angle will be……….	22.5 degree	45 degree	67.5 degree	None of these	None of these
28	Selection of gear material depends on…………….	Type of service	Peripheral speed	Shock and wear resistance	All of these	All of the above
29	Cycloidal profile gears are used in……………..	Air craft	Watches and clock	Machine tools	Toys	Watches and clock
30	Lewis equation is used to evaluate ………	Tensile stress in bending	Compressive stress in bending	Creep stress	Fatigue stress	Compressive stress in bending
31	Lewis form factor(y) depends upon	Number of teeth of a gear	Pressure angle	Size of teeth	Number of teeth of a gear & Pressure angle	Number of teeth of a gear & Pressure angle
32	When T is number of teeth, Lewis form factor for 14.5^o composite and full depth involute system is	0.124-0.684/T	0.154-0.912/T	0.175-0.841/T	None of these	0.154-0.912/T
33	When T is number of teeth, Lewis form factor for 20^o full depth involute system is	0.124-0.684/T	0.154-0.912/T	0.175-0.841/T	None of these	0.124-0.684/T
34	Permissible working stress (f_1) in the Lewis equation depends upon	Material of tooth	Load condition	Pitch line velocity	All of these	All of these

35	With the increase of pitch line velocity permissible working stress (f_1) in the Lewis equation	Increases	Decreases	Remains the same	None of these	Decreases
36	Worm gearing is used to obtain considerable speed reduction between shafts whose axes are	Perpendicular and do not intersect	Perpendicular and intersect	Inclined	Parallel	Perpendicular and do not intersect
37	Two shafts are connected by worm gears having D and d as diameters of worm wheel worm respectively. The centre distance will be	(D+d)/4	(D+d)/2	(D+d)	2(D+d)	(D+d)
38	In worm gearing system, the angle between the inclined faces in axial plane is ………… the pressure angle	Half	Equal	Two times	Three times	Two times
39	In single reduction, a large velocity ratio is required. The best transmission is …….	Spur gear drive	Helical gear drive	Bevel gear drive	Worm gear drive	Worm gear drive
40	The bevel gears are used to connect …………	Two parallel shaft	Two intersecting shaft	Two non-intersecting shaft	None of these	Two intersecting shaft
41	Bevel gears with shafts angle of 90^o are termed as……….	Zero gears	Angular bevel gears	Miter gears	Hypoid gears	Miter gears
42	Bevel gears used for connecting non-intersecting shafts are ……………	Miter gears	Hypoid gears	Spiral bevel gears	Zero gears	Hypoid gears
43	Face width of the bevel gears is equal to	Pitch cone radius/2	Two module	Pitch cone radius/3	None of these	Pitch cone radius/3
45	Ratio factor Q in wear load equation of bevel gear is given by……….	2(gear ratio G)/ (G+1)	2(ratio of formative no.of teeth of gear and pinion G')/ (G'+1)	(G+1)/ (G'+1)	2G'/G	2(ratio of formative no.of teeth of gear and pinion G')/ (G'+1)
46	Bevel factor should not be less than …………..	0.75	0.8	0.67	0.76	0.67
47	Pitch cone angle of pinion of straight tooth bevel gear pair with gear ratio 1.732 is	25^o	30^o	60^o	None of these	60^o
48	The face width of bevel gear is 0.3 times the radius of pitch cone. Here the bevel factor must be…………	1.3	3	0.7	1.7	0.7
49	Interchangibility is possible only in ……….	Bevel gears	Helical gears	Spur gear	Mitre gear	Spur gear
50	Worm and worm gear is a type of …………	Bevel gears	Spur gear	Helical gears	Spiral gear	Spiral gear
51	Worm gear with double start worm, pressure angle should not be less than	20^o	30^o	10^o	40^o	20^o
52	Shaft angle in case of worm and gear is	60^o	90^o	180^o	30^o	90^o
53	The type of gear with minimum axial thrust is	Bevel gear	Worm gear	Helical gear	Herring bone gear	Herring bone gear
54	Heat dissipation is important criteria of design in case ……………	Spur gear	Bevel gear	Worm gear	Helical gear	Worm gear

55	Material used for gear is always bronze in a pair of ……….	Spur gear	Helical gear	Hypoid gears	Worm gear	Worm gear
56	For transmission ratio of 25 and triple start worm the no. of teeth in worm gear is…….	38	75	100	50	75
57	Type of gearing used to connect two non-parallel shafts with shaft angle other than 90° is ……….	Miter gear	Herring bone gears	Spiral gears	Worm gear	Spiral gears
58	Usual proportion of face width b and diameter of worm d_w is ………..	b = 0.5	b = d_w /3	b = 0.73 d_w	b = 0.75 d_w	b = 0.73 d_w
59	Worm gear drive is used for hoists because it is…………	Very efficient	Noiseless	Self locking	Capable of taking large load	Self locking
60	Efficiency of worm gear is less than……..	90%	50%	60%	70%	50%

UNIT IV

DESIGN OF GEAR BOXES

Geometric progression – Standard step ratio – Ray diagram, kinematics layout –Design of sliding mesh gear box –Constant mesh gear box. – Design of multi speed gear box.

GEAR BOX

A gearbox is a mechanical device used for transferring energy from one device to another and is used to increase torque while reducing speed. A group of gears put together in a manner to increase the torque of an engine and reduce the speed is known as gear box. Generally, gear boxes are also known as speed reducers, or gear reducers.

TYPES OF GEAR BOXES

- According to the type of drive used: Spur, helical, bevel and worm gears.
- According to the No. of stages:

  (i) Single stage Gear box:

  It means one pair of gear wheels, such as one pinion and one wheel can be operated to reduce the speed.

  (ii) Multi-stage Gear box:

  It means two or more pairs of gears can be operated to reduce the speed.

Applications

Speed reducers are widely used for reduction of speed in turbine generator set; from motor to machine tool spindles; in rolling mills; from engine to road wheels in automobiles, etc. Speed reducers are widely used for reduction of speed in conveyors, crushers, cranes, elevators, feeders, small and large ball mills, mixers, towers and coal pulverizing units.

Requirements of Speed Gear Boxes

- It should provide silent operation of the power transmission.
- To reduce / increase the rpm.
- Change the direction of rotation (Clockwise / Anticlockwise)
- Shift the axis of rotation (linear or angular)
- To transmit the required power to the spindle.

Method of changing speed in gear boxes

Mainly two important methods are used,

(i) Sliding mesh gear box

In case of sliding mesh gear box, the gears on the main shaft mesh with appropriate gears on the spindle shaft to provide variety in the speed. Since they move left or right and slide along the gears, they are named as sliding mesh gear box.

(ii) Constant mesh gear box.

In this case, whether the counter shaft or main shaft, the gears are in constant mesh always with each other. They are also known to be silent or quite gear box. It is achieved by employing helical gears. To achieve variation in the spindle speed a different sliding mechanism is implemented.

Requirements to obtain optimum design

To reduce large diameter of the gear wheels and also to limit the pitch line velocity of gear drive, the following principles are to be followed.

1. Number of gears on the last stage should be minimum.

2. Number of gears on shafts should not be more than three. (But in some cases it may be four)

3. It is necessary to have,

$$N_{max} \geq N_{input} \geq N_{mini}$$

in all stages, except in the first stage.

4. The transmission ratio between driver and driven shaft should be the maximum.

$$\frac{N_{maxi}}{N_{mini}} \leq 2 \; and \; \frac{N_{mini}}{N_{input}} \geq \frac{1}{4}$$

(For all stages, except for first stage)

Main components of gear box

- Shafts for mounting gears.
- Oil seals for lubrication.
- Bearings to support shafts.
- Gears for getting different speeds.
- Gear box housing used for covering all inner components.

Preferred Numbers

It is the conventionally rounded off values derived from geometric progression series. There are five basic series available and they are R 5, R 10, R 20, R 40, R 80. The symbol R is used as a tribute to a French engineer Charles Renard. It is available in PSGDB pg. no. 7.19 and 7.20.

Step ratio or Series ratio or Progression ratio.

It is the ratio between the two adjacent speeds. It is denoted by ϕ

$$\frac{N_{maxi}}{N_{mini}} = \phi^{n-1}$$

Where, n is the number of steps of speed.

Structural formula:

$$n = P_1(X_1)\,.\ P_2(X_2)\,.\ P_3(X_3)\,.\ldots\ldots\ldots$$

Where,

n – Number of speeds available at the spindle

P_1, P_2, P_3 …. – stage numbers in the gear box

X_1, X_2, X_3 …. – characteristics of the stage

$X_1 = 1$ for first stage, $X_2 = P_1$ for second stage, $X_3 = P_2$ for third stage and so on.

Preferred Structural formula

6 speeds	$\underset{P_1}{3} \times \underset{P_2}{2}$ *or* $\underset{P_1}{2} \times \underset{P_2}{3}$
9 speeds	$\underset{P_1}{3} \times \underset{P_2}{3}$
12 speeds	$\underset{P_1}{3} \times \underset{P_2}{2} \times \underset{P_3}{2}$ (or) $\underset{P_1}{2} \times \underset{P_2}{3} \times \underset{P_3}{2}$ (or) $\underset{P_1}{2} \times \underset{P_2}{2} \times \underset{P_3}{3}$
18 speeds	$\underset{P_1}{3} \times \underset{P_2}{3} \times \underset{P_3}{2}$ (or) $\underset{P_1}{2} \times \underset{P_2}{3} \times \underset{P_3}{3}$ (or) $\underset{P_1}{3} \times \underset{P_2}{2} \times \underset{P_3}{3}$
16 speeds	$\underset{P_1}{2} \times \underset{P_2}{2} \times \underset{P_3}{2} \times \underset{P_4}{2}$

Kinematic layout or Kinematic arrangement

It is the figure showing the arrangement of gears in a gear box. To draw a kinematic layout, we need to find out the number of gears required in the gear box. This is approximately calculated as shown in the below table.

No. of speeds	Factors	Approximate No. of gears
2	(2×1)	$2\ (2) = 4$
3	(3×1) do not consider 1	$2\ (3) = 6$
4	2×2	$2\ (2 + 2) = 8$
6	2×3	$2\ (2 + 3) = 10$
8	$2 \times 2 \times 2$	$2\ (2 + 2 + 2) = 12$
9	3×3	$2\ (3 + 3) = 12$
12	$2 \times 3 \times 2$	$2\ (2 + 3 + 2) = 14$
16	$2 \times 2 \times 2 \times 2$	$2\ (2 + 2 + 2 + 2) = 16$
18	$3 \times 3 \times 2$	$2\ (3 + 3 + 2) = 16$

Ray diagram or Speed diagram

It is a graphical representation of the drive arrangement in general form or the graphical representation of the structural formula. It provides us with the data's like, number of stages, number of speeds in each stage, order of kinematic arrangement and the total number of speeds available at the spindle.

PROBLEMS:

1. Design nine speed gear box for milling machine with speed ranging from 56 – 900 rpm. The output speed is 720 rpm. Draw the speed diagram and kinematic layout.

Solution:

Step 1: To find the standard step ratio or progression ratio, ϕ

We know that,

$$\frac{N_{maxi}}{N_{mini}} = \phi^{n-1}$$

$$\frac{900}{56} = \phi^{9-1} \ or \ \phi = 1.415$$

We can write,

$$\phi = 1.12 \times (1.12 \times 1.12) = 1.40,$$

$hence, the\ standard\ step\ ratio\ to\ be\ followed\ is\ \phi = 1.12, R20\ series.$

Step 2: Selecting speed

$by\ skipping\ two\ speed, the\ selected\ speed\ are\ marked\ in\ the\ ray\ diagram$

Step 3: Structural formula

$$3 \times 3 = p_1 . p_2$$

$$Structural\ formula = p_1(X_1) . p_2(X_2) = 3\,(1). 3\,(3)$$

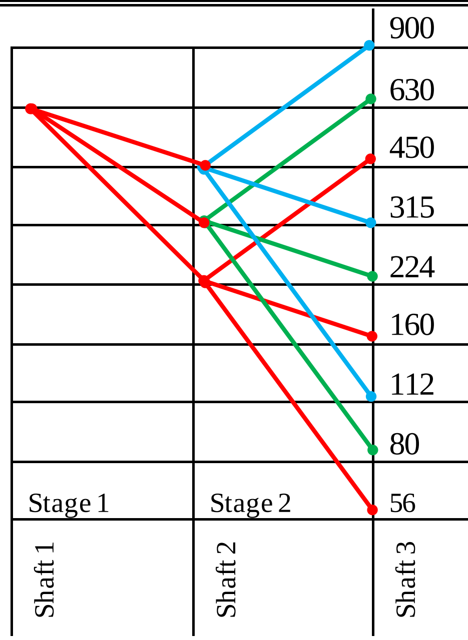
$$where, X_1 = 1, X_2 = p_1 = 3,$$

Step 4: RAY DIAGRAM

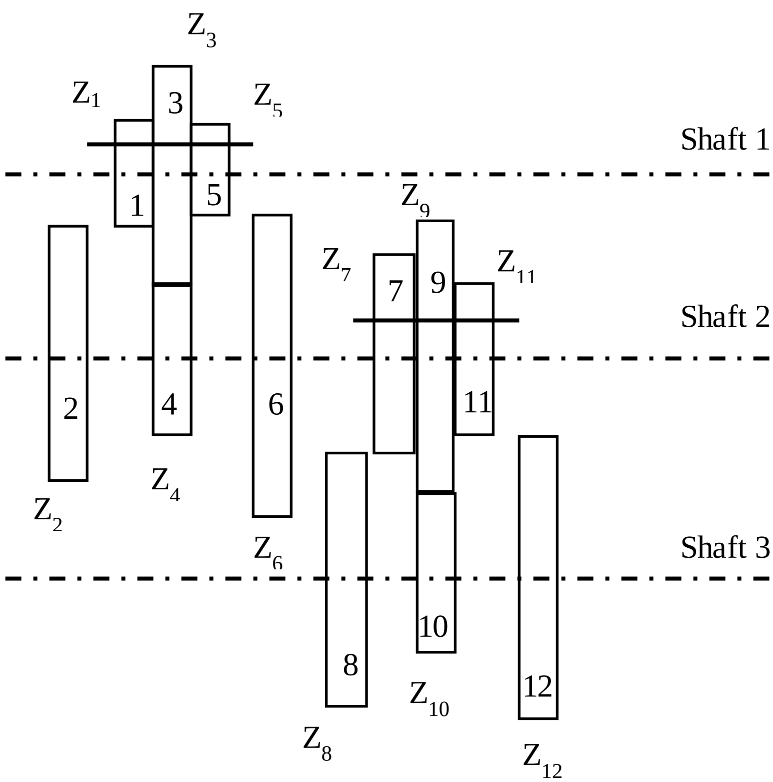
Stage 2:

$$\frac{N_{mini}}{N_{input}} = \frac{56}{224} = 0.25 \geq \frac{1}{4} \ and \ \frac{N_{maxi}}{N_{input}} = \frac{450}{224} = 2.00 \leq 2$$

Ratio requirement are satisfied



Step 5: KINEMATIC LAYOUT



2. Design a 12-speed gear box for an all geared headstock of a lathe. Maximum and minimum speeds are 600 rpm and 25 rpm respectively. The drive is from an electric motor giving 2.25 kW at 1440 rpm.

   Given Data:

   Step 1: To find the standard step ratio or progression ratio, ϕ

We know that,

$$\frac{N_{maxi}}{N_{mini}} = \phi^{n-1}$$

$$\frac{600}{25} = \phi^{12-1} \; or \; \phi = 1.334$$

The above ratio does not match with any of the std step ratio, hence the above value is to be used to calculate the speeds

Step 2: Selecting speed

$N_1 = 25\ rpm,$

$N_2 = N_1 \times \phi = 25 \times 1.334 = 33.35\ rpm = 34\ rpm$

$N_3 = N_2 \times \phi = 34 \times 1.334 = 45.35\ rpm = 46\ rpm$

$N_4 = N_3 \times \phi = 46 \times 1.334 = 61.35\ rpm = 62\ rpm$

$N_5 = N_4 \times \phi = 62 \times 1.334 = 82.7\ rpm = 83\ rpm$

$N_6 = N_5 \times \phi = 83 \times 1.334 = 110.72\ rpm = 111\ rpm$

$N_7 = N_6 \times \phi = 111 \times 1.334 = 148\ rpm$

$N_8 = N_7 \times \phi = 148 \times 1.334 = 197.43\ rpm = 198\ rpm$

$N_9 = N_8 \times \phi = 198 \times 1.334 = 264.13\ rpm = 265\ rpm$

$N_{10} = N_9 \times \phi = 265 \times 1.334 = 353.5\ rpm = 354\ rpm$

$N_{11} = N_{10} \times \phi = 354 \times 1.334 = 472.23 rpm = 473\ rpm$

$N_{12} = N_{11} \times \phi = 473 \times 1.334 = 630\ rpm$

Step 3: Structural formula

Given in the problem,

$$3 \times 2 \times 2 = p_1.p_2.p_3$$

$$Structural\ formula = p_1(X_1)\,.p_2(X_2)\,.p_3(X_3) = 3\,(1).2\,(3).2\,(6)$$

$$where, X_1 = 1, X_2 = p_1 = 3, X_3 = P_1 \times P_2 = 3 \times 2 = 6$$

Step 4: RAY DIAGRAM

Stage 3:

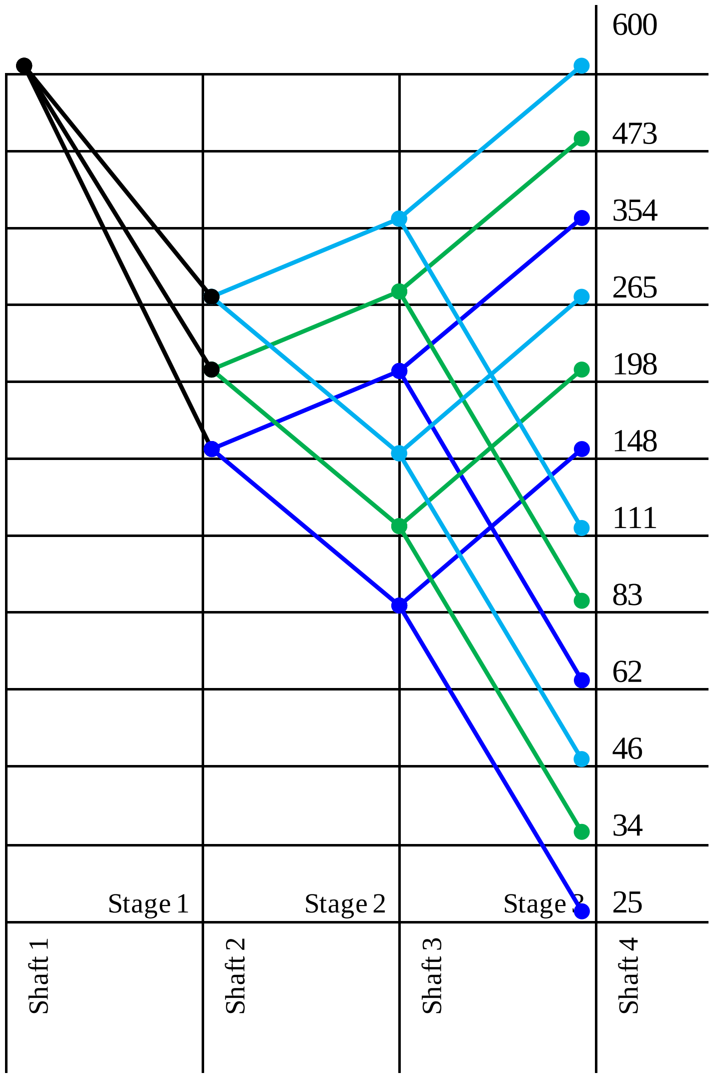
$$\frac{N_{mini}}{N_{input}} = \frac{25}{83} = 0.3 \geq \frac{1}{4} \;\; and \;\; \frac{N_{maxi}}{N_{input}} = \frac{148}{83} = 1.78 \leq 2$$

Ratio requirement is satisfied

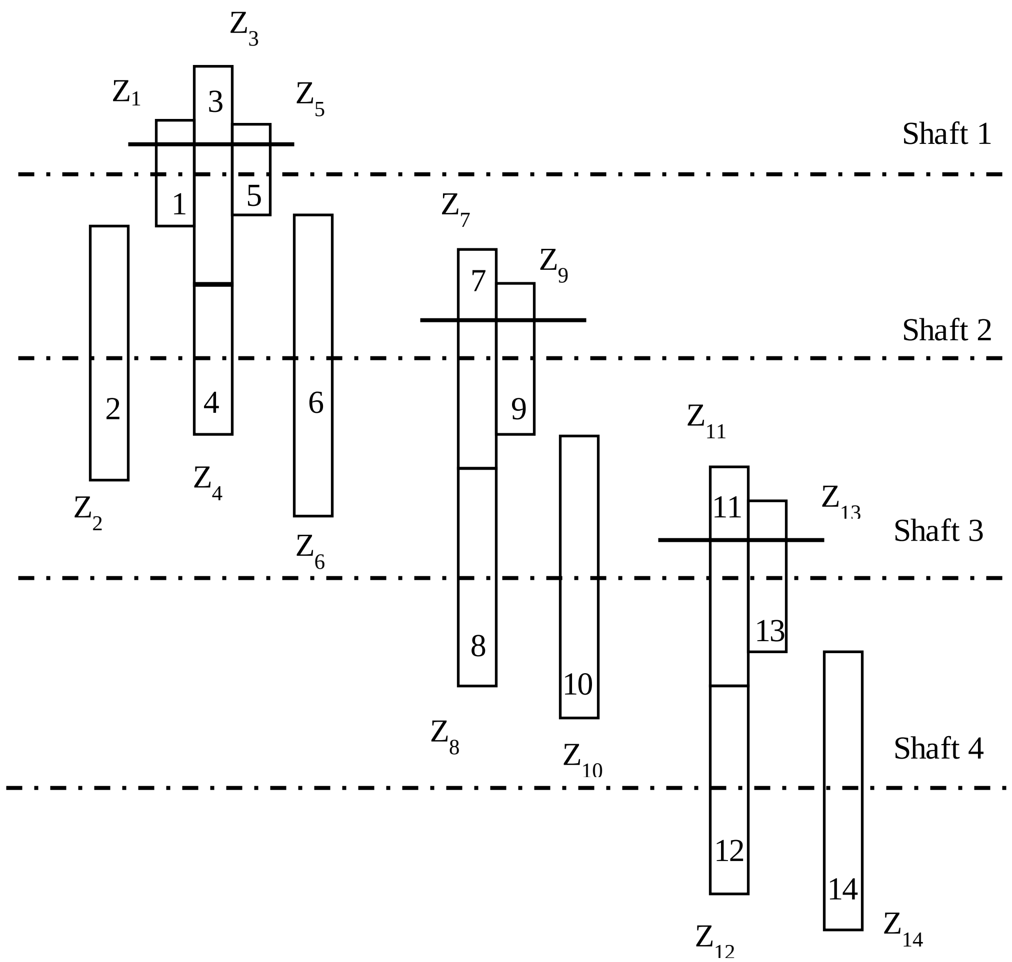
Stage 2:

$$\frac{N_{mini}}{N_{input}} = \frac{83}{148} = 0.56 \geq \frac{1}{4} \;\; and \;\; \frac{N_{maxi}}{N_{input}} = \frac{198}{148} = 1.33 \leq 2$$

Ratio requirement is satisfied



Step 5: KINEMATIC LAYOUT



3. A nine-speed gear box, used as a head stock gear box for a turret lathe, is to provide a speed range of 180rpm to 1800 rpm. Using standard step ratio draw the speed diagram, and kinematic layout.

Given Data:

Solution:

Step 1: To find the standard step ratio or progression ratio, ϕ

We know that,

$$\frac{N_{maxi}}{N_{mini}} = \phi^{n-1}$$

$$\frac{1800}{180} = \phi^{9-1} \; or \; \phi = 1.333$$

We can write,

$$\phi = 1.06 \times (1.06 \times 1.06 \times 1.06 \times 1.06) = 1.338,$$

$hence, the\ standard\ step\ ratio\ to\ be\ followed\ is\ \phi = 1.06, R40\ series.$

Step 2: Selecting speed

$by\ skipping\ one\ speed, the\ selected\ speed\ are\ marked\ in\ the\ ray\ diagram$

Step 3: Structural formula

$$3 \times 3 = p_1 . p_2$$

$$Structural\ formula = p_1(X_1)\ .p_2(X_2)\ = 3\ (1). 3\ (3)$$

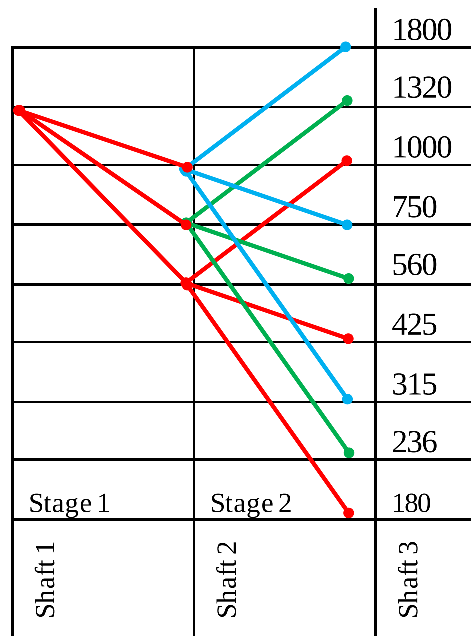
$$where, X_1\ = 1, X_2 = p_1\ = 3,$$

Step 4: RAY DIAGRAM

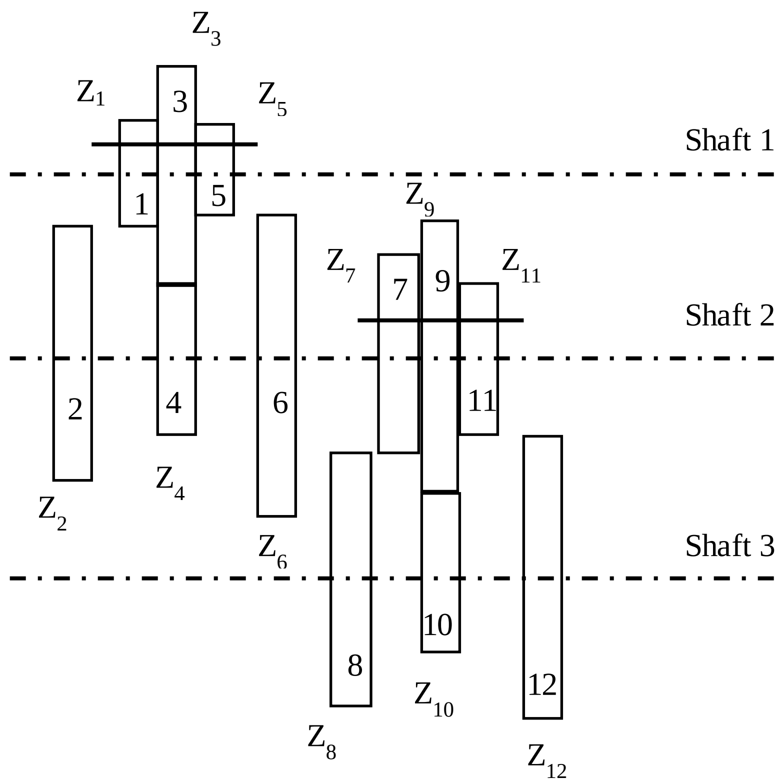
Stage 2:

$$\frac{N_{mini}}{N_{input}} = \frac{180}{560} = 0.32 \geq \frac{1}{4}\ \ and\ \ \frac{N_{maxi}}{N_{input}} = \frac{1000}{560} = 1.78 \leq 2$$

Ratio requirement are satisfied



Step 5: KINEMATIC LAYOUT



4. 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 = 2x3x3. List the speeds of all the shafts when the output speed is 16 rpm.

Given Data:

Solution:

Step 1: To find the standard step ratio or progression ratio, ϕ

We know that,

$$\frac{N_{maxi}}{N_{mini}} = \phi^{n-1}$$

$$\frac{800}{16} = \phi^{18-1} \; or \; \phi = 1.258$$

We can write,

$$\phi = 1.12 \times 1.12 = 1.254\,,$$

$hence, the\ standard\ step\ ratio\ to\ be\ followed\ is\ \phi = 1.12, R20\ series.$

Step 2: Selecting speed

$by\ skipping\ one\ speed, the\ selected\ speed\ are\ marked\ in\ the\ ray\ diagram$

Step 3: Structural formula

Given in the problem,

$$2 \times 3 \times 3 = p_1 . p_2 . p_3$$

$$Structural\ formula = p_1(X_1) . p_2(X_2) . p_3(X_3)$$

$$where, X_1 = 1, X_2 = p_1 = 2, X_3 = P_1 \times P_2 = 2 \times 3 = 6$$

Step 4: RAY DIAGRAM

Stage 3:

$$\frac{N_{mini}}{N_{input}} = \frac{16}{18} = 0.2 < \frac{1}{4} \quad and \quad \frac{N_{maxi}}{N_{input}} = \frac{250}{18} = 3.125 > 2$$

In this case condition is not satisfied and can be treated as special case.

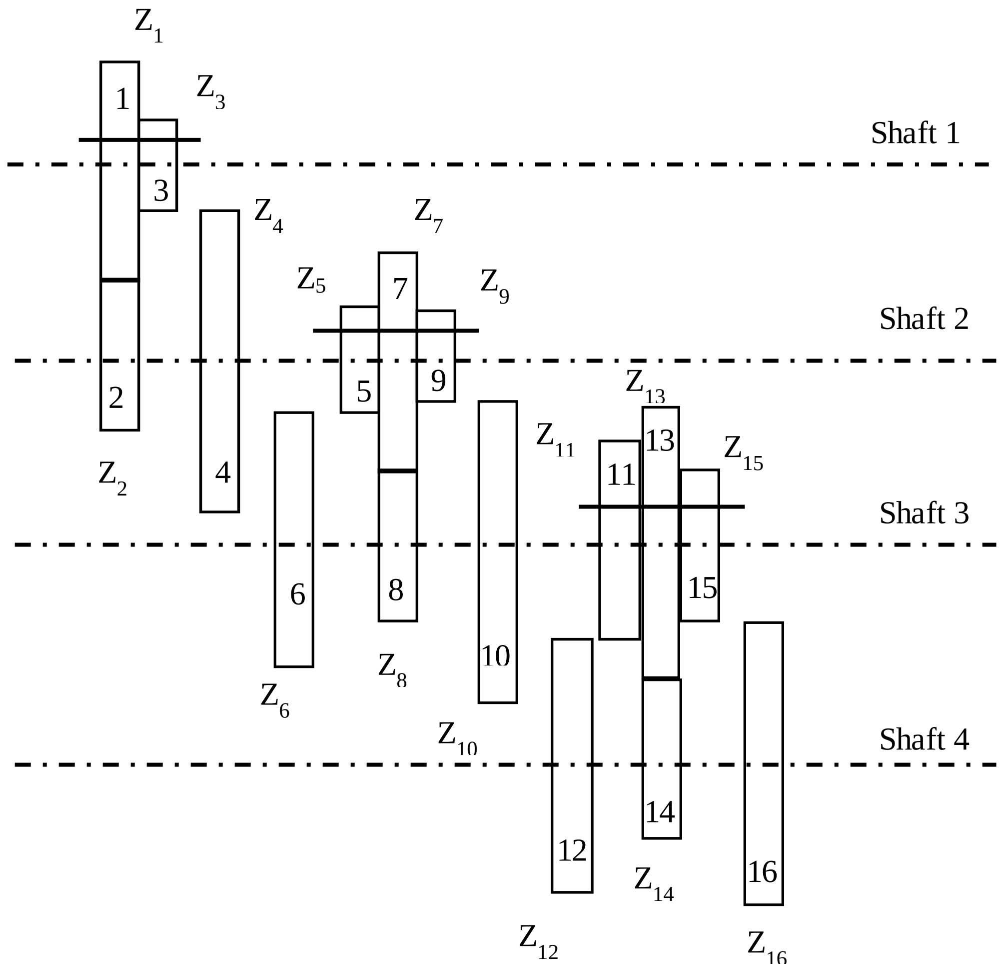
Stage 2:

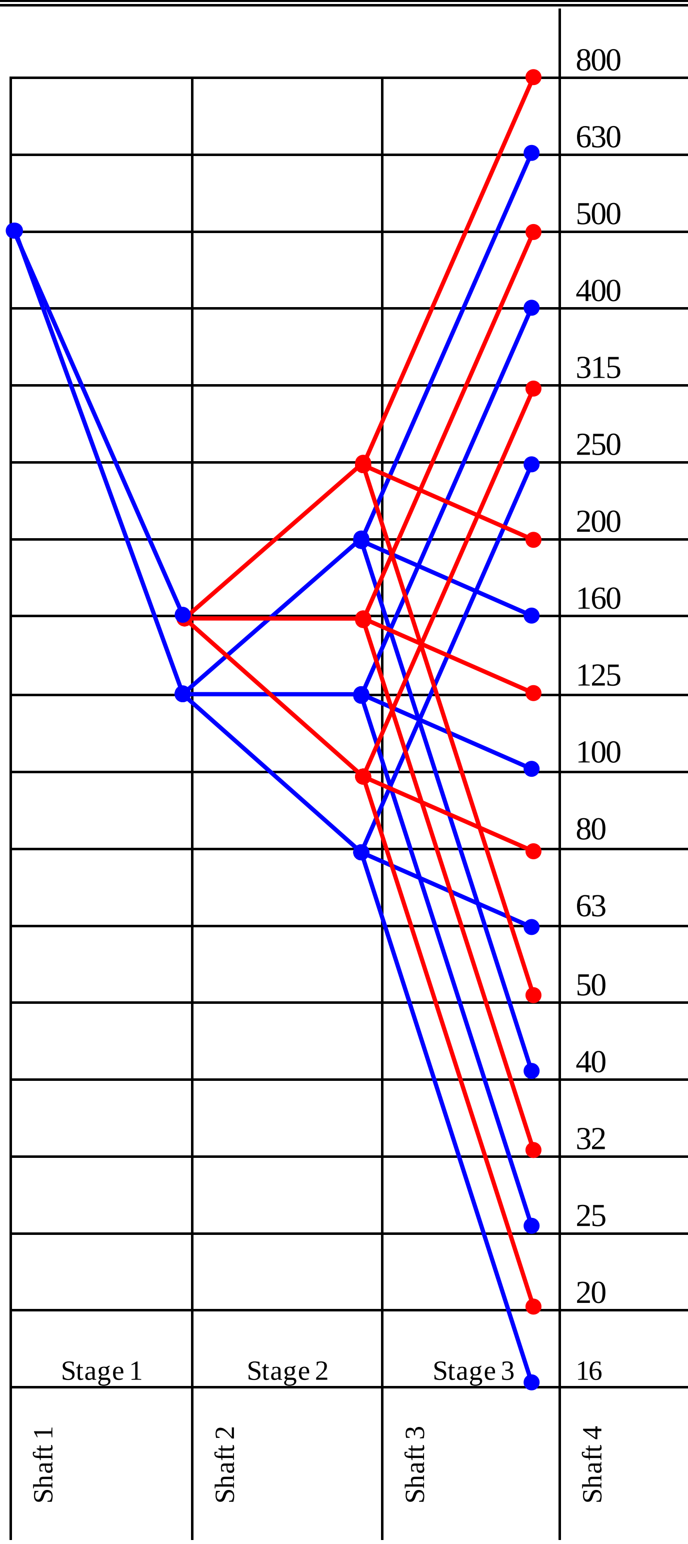
$$\frac{N_{mini}}{N_{input}} = \frac{80}{125} = 0.64 \leq \frac{1}{4} \quad and \quad \frac{N_{maxi}}{N_{input}} = \frac{250}{125} = 2 \geq 2$$

Stage 1:

$$\frac{N_{mini}}{N_{input}} = \frac{125}{200} = 0.25 \leq \frac{1}{4} \quad and \quad \frac{N_{maxi}}{N_{input}} = \frac{160}{200} = 0.32 \geq 2$$

Step 5: KINEMATIC LAYOUT




800
630
500
400
315
250
200
160
125
100
80
63
50
40
32
25
20
16
Stage 1
Stage 2
Stage 3
Shaft 1
Shaft 2
Shaft 3
Shaft 4

5. A six-speed gear box is required to provide output speeds in the range of 125 to 400 r.p.m. with a step ratio of 1.25 and transmit a power of 5 kW at 710 r.p.m. Draw the speed diagram and Kinematics diagram. Determine the number of teeth on all gears.
6. Design a 9-speed gear box for a machine to provide speeds ranging from 200 rpm to 1000 rpm the input is from a motor of 5 kW at 1440 rpm. Draw the ray diagram and the kinematic arrangement.
7. Design a 12-speed gearbox the speed range required is 100 to 355rpm
   a) Draw the ray diagram.
   b) Draw the kinematic arrangement.
8. The minimum and maximum speeds of a six-speed gear box are to be 160 and 500rpm. Construct the Kinematic arrangement and the ray diagram of the gearbox.

MULTIPLE CHOICE QUESTIONS

	QUESTION	OPTION 1	OPTION 2	OPTION 3	OPTION 4	ANSWER
1	For high gear reduction ratio……… is preferred.	Spur gear	Worm gear	Bevel Gears	Helical Gears	Worm gear
2	Worm gears are used in	Hoisting	Table Fan	Steering Rod	All of these	All of these
3	To the corresponding point on an adjacent tooth distance, measured axially, from a point on one tooth is	Pitch	Lead	Pressure angle	Reference point	Pitch
4	In which gear-drive the self-locking is available	Bevel gear	Helical gear	Spur gear	Worm gear	Worm gear
5	The number of gears employed in a gear-box is kept to the minimum by arranging the speed of the spindle is ……… series.	Panoramic	Logarithmic	Geometric	Quantum	Geometric
6	The ratio of adjacent speeds are constant in spindle speed are arranged in geometric progression is	Gear ratio	Step ratio	Velocity ratio	Standard ratio	Step ratio
7	………is known as the ratio of circular pitch to the length of arc of contact	Module	Addendum	Backlash	Contact ratio	Contact ratio
8	. ………….. is the graphical represent of different speeds of output shafts motor shaft and intermediate shafts.	Speed diagram	Space diagram	Kinematic diagram	Free body diagram	Speed diagram
9	……….. is the ratio between the speed of the shaft of a to its previous lower speed	Gear ratio	Standard ratio	Velocity ratio	Step ratio	Step ratio
10	Difference between the tooth space and the tooth thickness, measured on the pitch circle is known as……	Module	Backlash	Total depth	Working depth	Backlash
11	The speed range ratio is given by (where **V** is cutting speed, and **d** is work-piece diameter)	$(V_{max} / V_{min}) \times (d_{max} / d_{min})$	$(V_{max} / V_{min}) \times (d_{min} / d_{max})$	$(V_{min} / V_{max}) \times (d_{max} / d_{min})$	$(V_{min} / V_{max}) \times (d_{min} / d_{max})$	(V max / V min) x (d max / d min)
12	If from a geometric progression having progression ratio f, the members are removed in such a manner that only every x^{th} members remains, the remaining members from the new GP, with the ratio	$f^{1/x}$	f^{x}	f^{x2}	f^{2x}	fx
13	To obtain the 12 speeds in 3 stages, the recommended distribution can be	2x2x3	3x2x2	2x3x2	Any one of these	Any one of these
14	In a sliding mesh type gear box	Entire range of speed obtain with out stopping the motor	Least possible gears and shafts, are used	Different module gear are used	Entire range of speed obtain with out stopping the motor & Least possible gears and	Entire range of speed obtain with out stopping the motor & Least possible gears

					shafts, are used	and shafts, are used
15	Backlash is……….	Sum of clearance of two gears	Mutual play between two gears	Amount by which tooth space exceeds the thickness of engaging teeth	None of the above	Amount by which tooth space exceeds the thickness of engaging teeth
16	Ratioof length of arc of contact to the circular pitch is known as…………..	Module	Contact ratio	Backlash	Addendum	Contact ratio
17	Pitch circle diameter is equal to the product of………	Circular pitch and no. of teeth	Working depth and no. of teeth	Module and no. of teeth	Clearance and no of teeth	Module and no. of teeth
18	In skew bevel gearing, the axes of shafts are	Intersecting and the teeth are curved	Non-intersecting and non-parallel and the teeth are curved	Non-intersecting and non-parallel and the teeth are straight	None of these	Non-intersecting and non-parallel and the teeth are straight
19	In helical bevel gearing, the axes of shafts are ………… and the teeth are curved.	Intersectin	Non-intersecting and non-parallel	Parallel	All of these	Non-intersecting and non-parallel
20	When two non-intersecting and non-coplanar shafts are connected by gears, the arrangement is known as	Spur gearing	Helical gearing	Bevel gearing	Spiral gearing	Spiral gearing
21	The gears are termed as medium velocity gears, if their peripheral velocity is	1 – 3 m/s	3 – 15 m/s	15 – 30 m/s	30 - 50 m/s	3 – 15 m/s
22	An imaginary circle which by pure rolling action gives the same motion as the actual gear, is called	Addendum circle	Dedendum circle	Pitch circle	Clearance circle	Pitch circle
23	The size of a gear is usually specified by	Pressure angle	Pitch circle diameter	Circular pitch	Diametral pitch	Pitch circle diameter
24	The faces of the tooth is the	Surface of the top of the tooth	Surface of the toot above the pitch circle	Surface of the tooth below the pitch circle	Width of the tooth measured along the pitch circle	Surface of the toot above the pitch circle
25	The Dedendum circle diameter is equal to	Pitch circle diameter x cos f s	Addendum circle diameter x cos f	Clearance circle diameter x cos f	Pitch circle diameter x sin f	Pitch circle diameter x cos f s
26	An involute pinion and gear are in mesh. If both have the same size of addendum, then there will be interference between the	Tip of the gear and flank of pinion	Tip of the pinion and flank of gear	Flanks of both gear and pinion	Tip of both gear and pinion	Tip of the gear and flank of pinion
27	The minimum number of teeth on the pinion which will mesh any gear without interference for 20° full depth involute teeth will be	12	14	18	24	18

28	Which of the following statement is correct for gears?	The addendum is less than Dedendum	The pitch circle diameter is equal to the product of module and number of teeth	The pitch circle is always greater than the base circle	All of these	All of these
29	If the centre distance of the mating gears having involute teeth is varied within limits, the velocity ratio	Increases	Decreases	Remains unchanged	Increases or Decreases	Remains unchanged
30	Lewis equation in gears is used to find the	Tensile stress in bending	Shear stress	Compressive stress in bending	Fatigue stress	Compressive stress in bending
31	The velocity factor for very accurately cut and ground metallic gears operating at velocities up to ………….. is equal to 6 / 6 + V	10 m/s	12.5 m/s	15 m/s	20 m/s	20 m/s
32	The allowable static stress for steel gears is approximately ……….. of the ultimate tensile stress.	One – fourth	One – third	One – half	Double	One – third
33	The dynamic tooth load is due to	Inaccuracies of tooth spacing	Irregularities in tooth profiles.	Deflection of the teeth under load	All of these	All of these
34	Surface endurance limit of gear material is dependent upon its	Elastic strength	Yield strength	Brinell hardness number	Toughness	Brinell hardness number
35	In a helical gears, the distance between similar faces of adjacent teeth along a helix on the pitch cylinders normal to the teeth, is called	Normal pitch	Axial pitch	Diametral pitch	Module	Normal pitch
36	The helix angle for single helical gears ranges from	10 ° to 15 °	15 ° to 20 °	20 ° to 35 °	35 ° to 50°	20 ° to 35 °
37	The form factor of a helical gear …………. which the increases in the helix angle.	Increases	Decreases	Remains constant	Increases or Decreases	Increases
38	When the bevel gears having equal teeth and equal pitch angles connect two shafts whose axes intersect at right angles, then they are known as	Angular bevel gears	Crown bevel gears	Internal bevel gears	Mitre gears	Mitre gears
39	When the bevel gears connect two shafts whose axes intersect at an angles greater than a right angle and one of the bevel gears has a pitch angle of 90, they are known as	Angular bevel gears	Crown bevel gears	Internal bevel gears	Mitre gears	Crown bevel gears
40	For a bevel gear having the pitch angle q, the ratio of formative number of the teeth (T_E) to actual number of teeth (T) is	1/ sin q	1/ cos q	1/ tan q	Sin q cos q	1/ cos q

41	In worm, gears, pitch lead angle is	Half the angle between two inclined faces in axial plane	The angle between the tangent to the pitch helix and the plane of rotation	The angle between the tangent to the pitch helix and an element of the cylinder	None of these	The angle between the tangent to the pitch helix and the plane of rotation
42	In the ratio of axial module and pitch diameter is	Lead and pitch	Diameter factor	Dynamic Factor	None of these	Diameter factor
43	To avoid interruption between the gear and bearings or one gear and another gear	Spacers	Sleeves	Bushes	Clamps	Spacers, sleeves
44	In which the arrangements of various gears in various shafts of the gear boxin order to obtain the different output speeds from the single speed is ……	Space diagram	Kinematic diagram	Free body diagram	Ray diagram	Free body diagram
45	Ray diagram of a gear box indicate the arrangement of the various gears in shafts in order to obtain the different ……… from the single speed of the	Output speed	Output torque	Output power	Output gears	Output speed
46	For machining a component of any arbitrary diameter, at required speed, the machine tool should be run with	Finite number of speed	Step less speed	Infinitely variable speed	None of these	Step less speed

UNIT V – DESIGN OF CLUTCH AND BRAKES

Design of plate clutches –axial clutches–cone clutches–internal expanding rim clutches–internal and external shoe brakes.

DESIGN OF CLUTCH

A clutch is a device used to engage and disengage the driving shaft to the driver shaft according to the requirement. In case of automobiles, the clutch is required during changing of gears. By operating a lever, the clutch engages and disengages the driver and driven shafts. While changing gears, the driven shaft should be disengaged from the driving shaft.

Classification of Clutch

Clutches are classified in two ways, based on

1. Method of energy transfer (actuation).
2. Method of engagement.

According to the method of energy transfer, the clutches are categorized into

1. Mechanically actuated clutches.
2. Hydraulically actuated clutches.
3. Pneumatically actuated clutches.
4. Electrically actuated clutches.

According to the method of engagement, clutches are categorized into

1. Friction clutches.
2. Positive contact clutches.
3. over running clutches.
4. Magnetic clutches.

The actuating force or the force required to engage the clutches is supplied mainly by springs.

Material for friction surface

The materials used for lining of friction surfaces should have the following properties.

1. The material should have high coefficient of friction.
2. The material should withstand high temperatures caused during operation.

3. The material should have high heat conductivity, high resistance to wear and should not be effected by moisture or oil.

Types of friction clutches

1. Disc or plate clutches.

(a) Single disc clutch

(b) Multiple disc clutch

2. Cone clutches.

3. Centrifugal clutches.

Advantages of friction clutches.

1. They have a very little shock during engagement as they can slip relative to each other.

2. It can be used for high speed engagement applications.

Disadvantages of friction clutches

1. They are not suitable for application that require positive transmission because they do slip.

2. Replacement of friction material is often required as they wear out.

3. External cooling is required as they generate heat during engagement.

Design considerations for friction clutches

1. Sufficient torque transmitting capacity.

2. Light weight.

3. Smooth engagement without shock and fast disengagement with drag.

4. Proper friction material (high co-efficient of friction) that should not be affected by moisture and oil, etc.

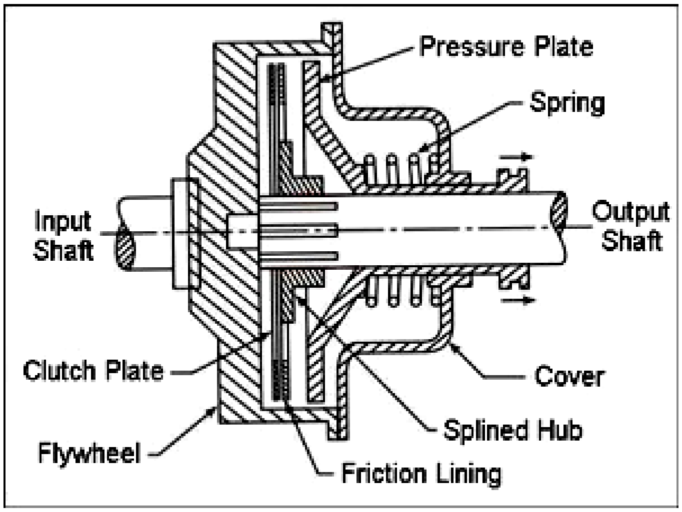
5. Provision for taking up wear of the contact surfaces.

6. Provision for transmission of heat which is generated during operation.

Single Disc or Plate Clutch

A single disc or plate clutch, as shown in Fig, consists of a clutch plate whose both sides are faced with a frictional material (usually of Ferrodo). It is mounted on the hub which is free to move axially along the splines of the driven shaft. The pressure plate is mounted inside the clutch body which is bolted to the flywheel. Both the pressure plate and the flywheel rotate with the engine crankshaft or the driving shaft. The pressure plate pushes the clutch plate towards the flywheel by a set of strong springs which are arranged radially inside the body. The three levers (also known as release levers or fingers) are carried on pivots suspended from the case of the body. These are arranged in such a manner so that the pressure plate moves away from the flywheel by the inward movement of a thrust bearing. The bearing is mounted upon a forked shaft and moves forward when the clutch pedal is pressed.

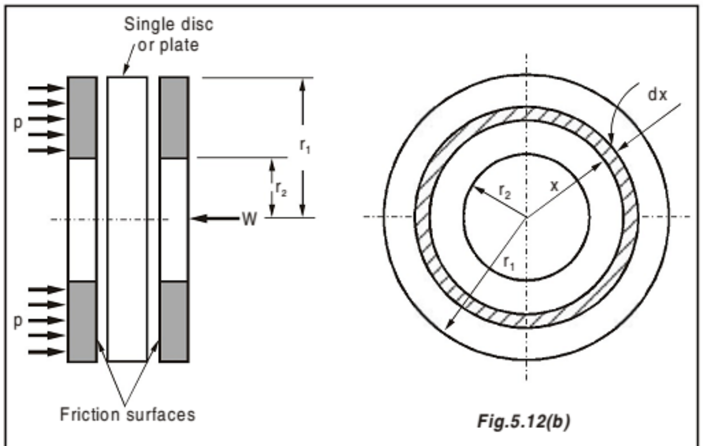
When the clutch pedal is pressed down, its linkage forces the thrust release bearing to move in towards the flywheel and pressing the longer ends of the levers inward. The levers are forced to turn on their suspended pivot and the pressure plate moves away from the flywheel by the knife edges, thereby compressing the clutch springs. This action removes the pressure from the clutch plate and thus moves back from the flywheel and the driven shaft becomes stationary. On the other hand, when the foot is taken off from the clutch pedal, the thrust bearing moves back by the levers. This allows the springs to extend and thus the pressure plate pushes the clutch plate back towards the flywheel.



The axial pressure exerted by the spring provides a frictional force in the circumferential direction when the relative motion between the driving and driven members tends to take place. If the torque due to this frictional force exceeds the torque to be transmitted, then no slipping takes place and the power is transmitted from the driving shaft to the driven shaft.

Design of a Disc (or) plate clutch

Consider two friction surfaces



Forces on a Disc Clutch.

Let,

W = Axial thrust,

p = Intensity of axial pressure,

$r1$ = External radius of friction disc

$r2$ = Internal radius of friction disc.

r = Mean radius of friction disc

μ = Coefficient of friction

$F\ t$ = Frictional torque transmitted by the clutch

Consider the following two conditions

1. Uniform pressure distributed over the entire area of the friction surface.

2. Uniform axial wear due to the sliding friction.

1. By considering uniform pressure

$$intensity\ of\ pressure, p = \frac{W}{\pi(r_1^2 - r_2^2)}$$

$$Total\ frictional\ torque, F_t = \mu\, W\, r$$

Where,

$$mean\ radius\ of\ the\ friction\ surface, r = \frac{2}{3}\left[\frac{r_1^3 - r_2^3}{r_1^2 - r_2^2}\right]$$

2. By considering uniform wear

Let p = Intensity of pressure

x = radial distance from axis of the clutch

Intensity of pressure varies inversely with the radial distance (x);

$$\therefore p \propto \frac{1}{x}$$

$$p\ x = constant$$

(a) The intensity of pressure is maximum at inner radius

$$p_{max} r_2 = C$$

$$p_{max} = \frac{C}{r_2}$$

(b) The intensity of pressure is minimum at outer radius $r1$

$$p_{min} r_1 = C$$

$$p_{min} = \frac{C}{r_1}$$

(c) The average pressure is given by,

$$p_{average} = \frac{W}{\pi(r_1^2 - r_2^2)}$$

$$Total\ frictional\ torque, F_t = \mu\, W\, r$$

Where,

$$mean\ radius\ of\ the\ friction\ surface, r = \frac{1}{2}\,(r_1 + r_2)$$

(d) Axial thrust,

$$W = 2\pi\ C\ (r_1 - r_2)$$

The total frictional torque acting on the clutch (Ft)

$$F_t = n\ \mu W\ r$$

Where n = no. of pairs of contact surfaces

Note:

1. for a single plate or disc clutch, n = 2 (both sides of the plate are effective)

2. for a multiple disc clutch

$$n = n_1 + n_2 - 1$$

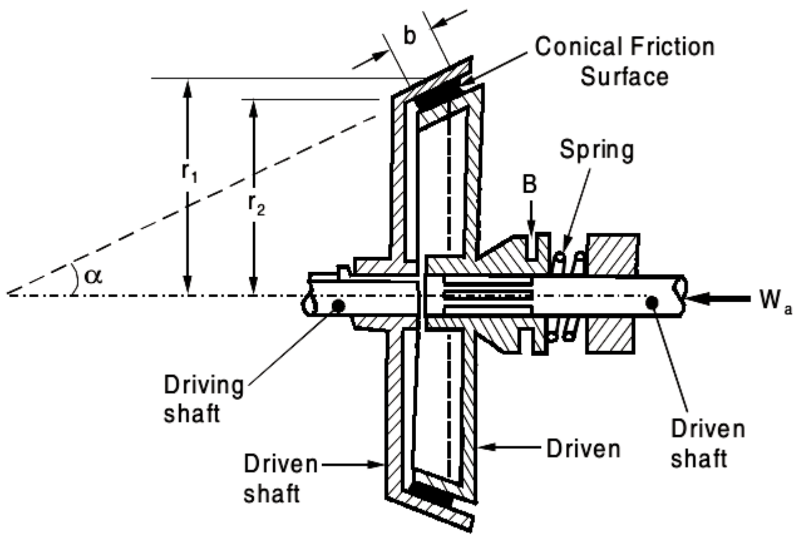
Where

n_1 = no. of discs on driving shaft

n_2 = no. of discs on driven shaft.

CONE CLUTCH

A cone clutch, as shown in Fig, was extensively used in automobiles, but now-a-days it has been replaced completely by the disc clutch. It consists of one pair of friction surface only. In a cone clutch, the driver is keyed to the driving shaft by a sunk key and has an inside conical surface or face which exactly fits into the outside conical surface of the driven. The driven member resting on the feather key in the driven shaft, may be shifted along the shaft by a forked lever provided at *B*, in order to engage the clutch by bringing the two conical surfaces in contact.



Due to the frictional resistance set up at this contact surface, the torque is transmitted from one shaft to another. In some cases, a spring is placed around the driven shaft in contact with the hub of the driven. This spring holds the clutch faces in contact and maintains the pressure between them, and the forked lever is used only for disengagement of the clutch. The contact surfaces of the clutch may be metal to metal contact, but more often the driven member is lined with some material like wood, leather, cork or asbestos etc. The material of the clutch faces (*i.e.* contact surfaces) depends upon the allowable normal pressure and the coefficient of friction.

Design of cone clutch:

Consider a pair of friction surfaces of cone clutch.

Consider the following two conditions

1. Uniform pressure

2. Uniform wear

1. Uniform pressure

Frictional torque

$$F_t = \frac{2\,\pi\,\mu\,p_n}{\sin\alpha}\left[\frac{r_1^3 - r_2^3}{3}\right]$$

Normal force, W_n

$$W_n = p_n \times 2\,\pi\,r\,b$$

Axial force, W_a

$$W_a = W_n \times \sin\alpha = 2\,\pi\,p_n r\,b\sin\alpha$$

Where

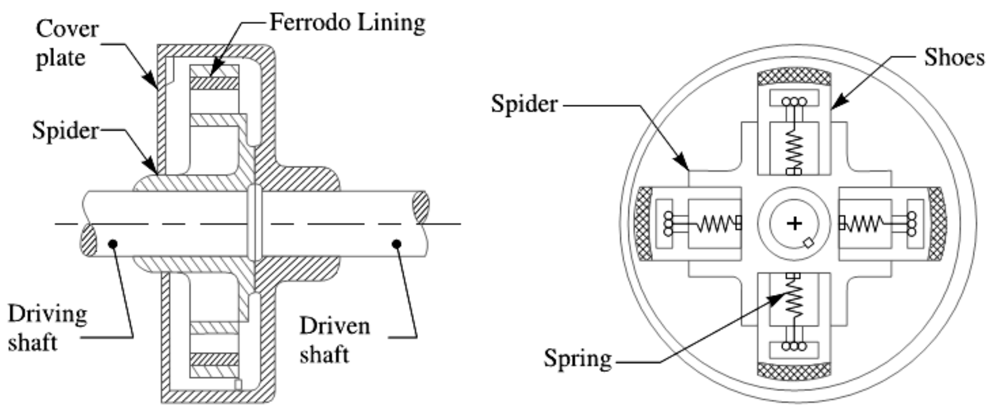
$$\sin\alpha = \frac{r_1 - r_2}{b}\ and\ r = \frac{r_1 + r_2}{2}$$

Therefore, normal pressure

$$p_n = \frac{W_a}{\pi[r_1^2 - r_2^2]}$$

Centrifugal Clutch

The centrifugal clutches are usually incorporated into the motor pulleys. It consists of a number of shoes on the inside of a rim of the pulley, as shown in Fig. The outer surface of the shoes are covered with a friction material. These shoes, which can move radially in guides, are held against the boss (or spider) on the driving shaft by means of springs. The springs exert a radially inward force which is assumed constant. The weight of the shoe, when revolving causes it to exert a radially outward force (*i.e.* centrifugal force). The magnitude of this centrifugal force depends upon the speed at which the shoe is revolving.



A little consideration will show that when the centrifugal force is less than the spring force, the shoe remains in the same position as when the driving shaft was stationary, but when the centrifugal force is equal to the spring force, the shoe is just floating. When the centrifugal force exceeds the spring force, the shoe moves outward and comes into contact with the driven member and presses against it. The force with which the shoe presses against the driven member is the difference of the centrifugal force and the spring force. The increase of speed causes the shoe to press harder and enables more torque to be transmitted.

Design of Brakes

Introduction

A brake is a device by means of which artificial frictional resistance is applied to a moving machine member, in order to retard or stop the motion of a machine. In the process of performing this function, the brake absorbs either kinetic energy of the moving member or potential energy given up by objects being lowered by hoists, elevators etc. The energy absorbed by brakes is dissipated in the form of heat. This heat is dissipated in the surrounding air (or water which is circulated through the passages in the brake drum) so that excessive heating of the brake lining does not take place. The design or capacity of a brake depends upon the following factors:

1. The unit pressure between the braking surfaces,

2. The coefficient of friction between the braking surfaces,

3. The peripheral velocity of the brake drum,

4. The projected area of the friction surfaces, and

5. The ability of the brake to dissipate heat equivalent to the energy being absorbed. The major functional difference between a clutch and a brake is that a clutch is used to keep the driving and driven member moving together, whereas brakes are used to stop a moving member or to control its speed.

Difference between clutch and brake

Clutch is used to rotate the driven member along with driving member. Whereas, brake is used to control or stop the moving member. In other words, clutch is used to connect two moving members of the machine, i.e. driven member along with driving members. Whereas brake connects a moving member to a stationary member. Therefore we can say that, if one member of the clutch is kept fixed (stationary), then the clutch becomes a brake.

Brake friction materials

The brake lining material should have the following properties.

1. It should have high coefficient of friction.

2. It should have high wear and heat resistance.

3. It should have high heat dissipation.

4. It should not react with moisture and oil.

Types of Brakes

1. Simple block shoe brake

2. Pivoted block shoe brake

3. Double block shoe brake

4. Simple band brake

5. Differential band brake

6. Band and block brake

7. Internal expanding brake

Single Block Shoe Brake

It consists of a block which is pressed against the rotating wheel. The friction between the block and wheel, it causes a tangential braking force on the wheel. This tangential braking force retards or stops the rotation of the wheel. The block is attached to a lever, which is pivoted at one end and the force is applied on the other end.

Let,

P = Applied force

R_N = Normal Reaction force

r = Wheel radius

2θ = Angle of contact between block and wheel

μ = Coefficient of friction

Ft = Tangential braking force (or) frictional force

l = Length of the lever;

x = distance between centre of wheel to the fulcrum.

It is assumed that, the normal pressure is uniform between the block and the wheel when the angle of contact $2\theta < 60°$

Then, frictional braking force,

$$F_t = \mu R_N$$

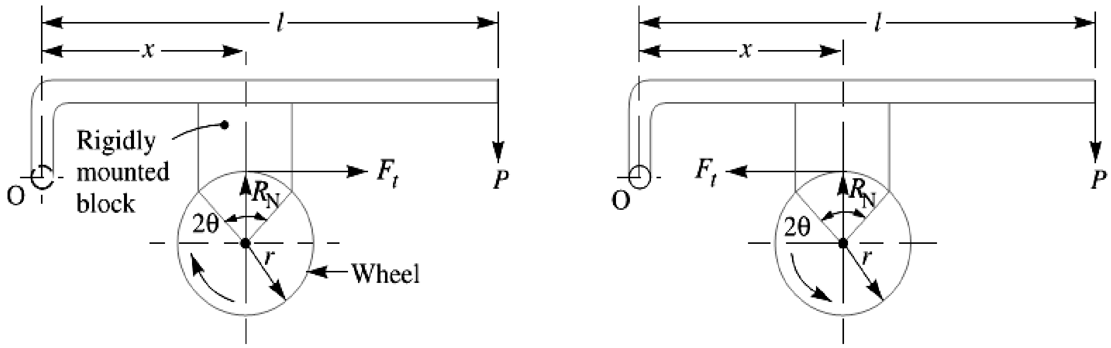
And the braking torque,

$$T_B = F_t r = \mu R_n r$$

There are three conditions in the arrangement of contact surface and position of pivot.

Case: 1

When the tangential braking force (F_t) passes through the fulcrum of the lever



(*a*) Clockwise rotation of brake wheel. (*b*) Anticlockwise rotation of brake wheel.

The normal reaction force for case 1,

a) For clockwise rotation of brake wheel

$$R_N = \frac{p \times l}{x}$$

b) For Anticlockwise rotation of brake wheel

$$R_N = \frac{p \times l}{x}$$

The braking torque for case 1,

a) For clockwise rotation of brake wheel

$$T_B = \frac{\mu \times p \times l \times r}{x}$$

b) For Anticlockwise rotation of brake wheel

$$T_B = \frac{\mu \times p \times l \times r}{x}$$

Case II

When the tangential force (F_t) passes through a distance ‘a’ below the fulcrum.

The normal reaction force for case 2,

c) For clockwise rotation of brake wheel

$$R_N = \frac{p \times l}{\mu\, a + x}$$

d) For Anticlockwise rotation of brake wheel

$$R_N = \frac{p \times l}{x - \mu\, a}$$

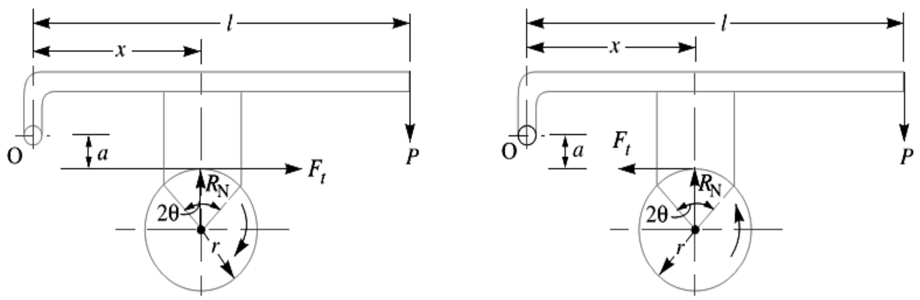
The braking torque for case 2,

c) For clockwise rotation of brake wheel

$$T_B = \frac{\mu \times p \times l \times r}{\mu\, a + x}$$

d) For Anticlockwise rotation of brake wheel

$$T_B = \frac{\mu \times p \times l \times r}{x - \mu\, a}$$



(*a*) Clockwise rotation of brake wheel. (*b*) Anticlockwise rotation of brake wheel.

Case III

When the tangential force (F_t) passes through a distance 'a' above the fulcrum.

The normal reaction force for case 3,

e) For clockwise rotation of brake wheel

$$R_N = \frac{p \times l}{x - \mu\, a}$$

f) For Anticlockwise rotation of brake wheel

$$R_N = \frac{p \times l}{\mu\, a + x}$$

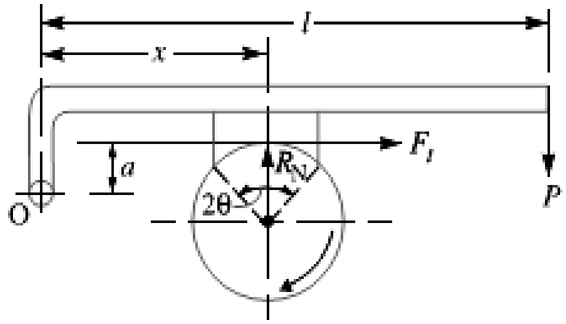
The braking torque for case 2,

e) For clockwise rotation of brake wheel

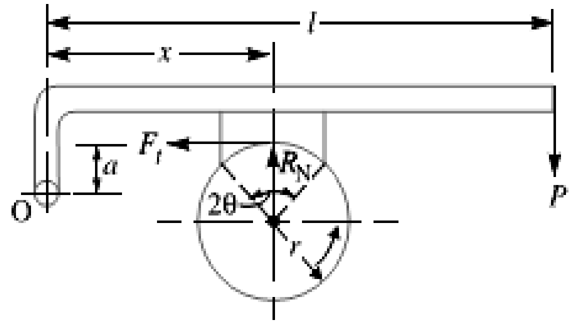
$$T_B = \frac{\mu \times p \times l \times r}{x - \mu\, a}$$

f) For Anticlockwise rotation of brake wheel

$$T_B = \frac{\mu \times p \times l \times r}{\mu\, a + x}$$



(*a*) Clockwise rotation of brake wheel.



(*b*) Anticlockwise rotation of brake wheel.

When the frictional force is greater enough to apply the brake without any external force, then the brake is called self-locking brake.

Condition for self locking

if $x\,\mu \leq a$; the P will be –ve or equal to zero.

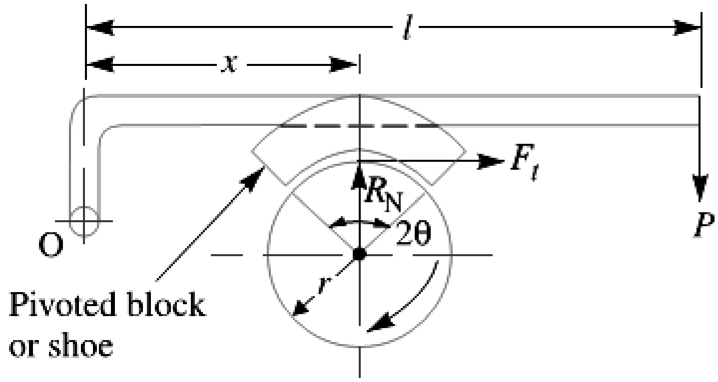
No external force is required to apply the brake. The self locking brake is used only in back – stop applications. In order to avoid self locking,

$$x\,\mu \geq a$$

The brake should be self energizing but not self locking

Pivoted Block or Shoe Brake

When the angle of contact is less than 60°, then it may be assumed that the normal pressure between the block and the wheel is uniform. But when the angle of contact is greater than 60°, then the unit pressure normal to the surface of contact is less at the ends than at the centre. In such cases, the block or shoe is pivoted to the lever as shown in Fig., instead of being rigidly attached to the lever. This gives uniform wear of the brake lining in the direction of the applied force. The braking torque for a pivoted block or shoe brake (*i.e.* when 2θ > 60°) is given by



$$T_B = F_t \times r = \mu' R_N r$$

Where

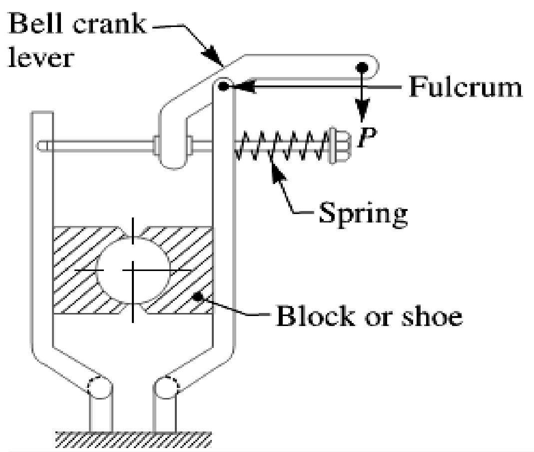
$$\mu' = equivalent\ coefficient\ of\ friction = \frac{4\,\mu \sin\theta}{2\,\theta + \sin 2\theta}$$

μ = Actual coefficient of friction.

These brakes have more life and may provide a higher braking torque.

Double Block or Shoe Brake

When a single block brake is applied to a rolling wheel and additional load is thrown on the shaft bearings due to the normal force (R_N). This produces bending of the shaft. In order to overcome this drawback, a double block or shoe brake as shown in Fig., is used. It consists of two brake blocks applied at the opposite ends of a diameter of the wheel which eliminate or reduces the unbalanced force on the shaft. The brake is set by a spring which pulls the upper ends of the brake arms together. When a force P is applied to the bell crank lever, the spring is compressed and the brake is released. This type of brake is often used on electric cranes and the force P is produced by an electromagnet or solenoid. When the current is switched off, there is no force on the bell crank lever and the brake is engaged automatically due to the spring force and thus there will be no downward movement of the load.



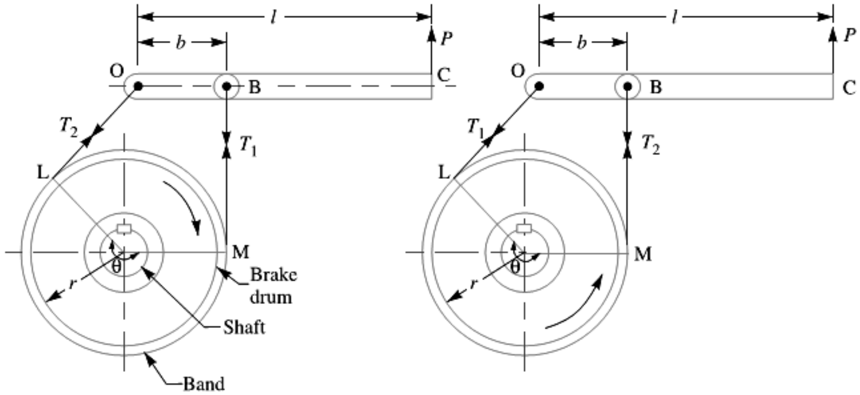
In a double block brake, the braking action is doubled by the use of two blocks and the two blocks may be operated practically by the same force which will operate one. In case of double block or shoe brake, the braking torque is given by

$$T_B = (F_{t1} + F_{t2})\, r$$

Where, F_{t1} and F_{t2} are the braking forces on the two blocks.

Simple band brake

A band brake consists of a flexible band of leather, one or more ropes, or steel lined with friction material, which embraces a part of the circumference of the drum. A band brake, as shown in Fig., is called a ***simple band brake*** in which one end of the band is attached to a fixed pin or fulcrum of the lever while the other end is attached to the lever at a distance b from the fulcrum.



(*a*) Clockwise rotation of drum. (*b*) Anticlockwise rotation of drum.

Where

T_1 = Tension in the tight side of the band, T_2 = Tension in the slack side of the band, θ = Angle of lap (or embrace) of the band on the drum, μ = Coefficient of friction between the band and the drum, r = Radius of the drum, t = Thickness of the band, and r_e = Effective radius of the drum = r + t / 2.

Limiting ratios of tensions is given by the relation,

$$\frac{T_1}{T_2} = e^{\mu\theta}$$

Braking torque on the drum,

$$T_B = (T_1 - T_2) \times r, \quad Nm$$

$$T_B = (T_1 - T_2) \times r_e, \quad Nm$$

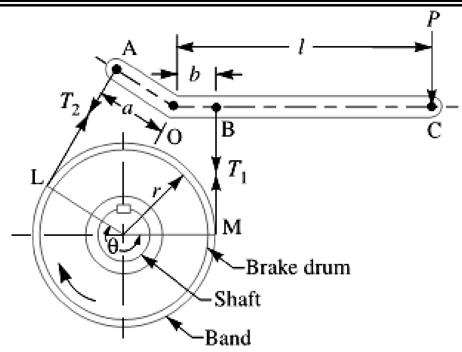
To find the width of the band,

$$T_1 = \sigma_t\, b\, t, \quad N$$

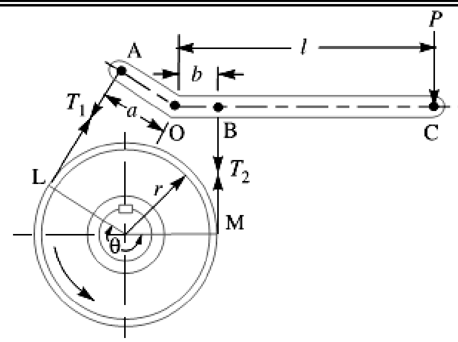
Where, σ_t = tensile stress in the band material.

Differential Band Brake

In a differential band brake, as shown in Fig., the ends of the band are joined at *A* and *B* to a lever *AOC* pivoted on a fixed pin or fulcrum *O*. It may be noted that for the band to tighten, the length *OA* must be greater than the length *OB*.



(a) Clockwise rotation of the drum.



(b) Anticlockwise rotation of the drum.

The braking torque on the drum may be obtained in the similar way as discussed in simple band brake. Now considering the equilibrium of the lever AOC. It may be noted that when the drum rotates in the clockwise direction, as shown in Fig (a), the end of the band attached to A will be slack with tension T_2 and end of the band attached to B will be tight with tension T_1. On the other hand, when the drum rotates in the anticlockwise direction, as shown in Fig (b), the end of the band attached to A will be tight with tension T_1 and end of the band attached to B will be slack with tension T_2.

For clockwise rotation of the drum,

$$P\,l = T_2\,a - T_1\,b$$

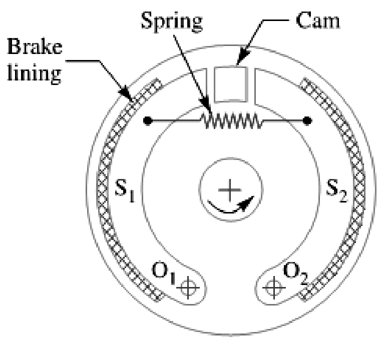
For anticlockwise rotation of the drum,

$$P\,l = T_1\,a - T_2\,b$$

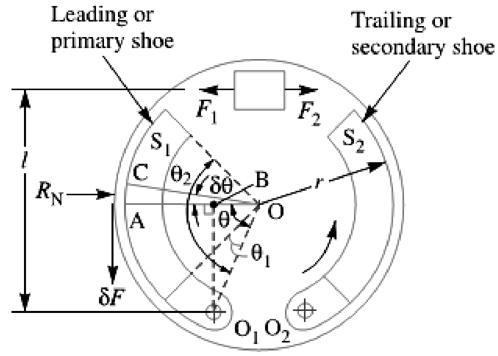
When the force P is negative or zero, then brake is self locking. Thus for differential band brake and for clockwise rotation of the drum, the condition for self- locking is $T_2 .a \leq T_1.b$ or $T_2/T_1 \leq b / a$ and for anticlockwise rotation of the drum, the condition for self-locking is $T_1 .a \leq T_2.b$ or $T_1/T_2 \leq b / a$

Internal Expanding Brake

An internal expanding brake consists of two shoes S_1 and S_2 as shown in Fig. (a). The outer surface of the shoes are lined with some friction material (usually with Ferodo) to increase the coefficient of friction and to prevent wearing away of the metal. Each shoe is pivoted at one end about a fixed fulcrum O_1 and O_2 and made to contact a cam at the other end. When the cam rotates, the shoes are pushed outwards against the rim of the drum. The friction between the shoes and the drum produces the braking torque and hence reduces the speed of the drum. The shoes are normally held in off position by a spring as shown in Fig (a). The drum encloses the entire mechanism to keep out dust and moisture. This type of brake is commonly used in motor cars and light trucks.



(a) Internal expanding brake.



(b) Forces on an internal expanding brake.

Problems:

1. 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 the sides of the plate are effective and the Coefficient of friction is 0.25. Determine (i) the diameter of the plate (ii) the axial force to engage the clutch. Assume the uniform wear theory.

   Solution:

   To find the inner diameter of the plate,

$$power\ transmitted, P = \frac{2\pi N\,T}{60}$$

$$25 \times 10^3 = \frac{2\pi \times 900 \times T}{60}$$

$$T = 265.26\ Nm$$

   The intensity of pressure is maximum at the inner radius,

$$p_{maxi}.r_2 = C$$

$$C = 85 \times 10^3.r_2$$

   The axial thrust transmitted to the frictional surface,

$$W = 2\pi C\ (r_1 - r_2) = 2\pi \times 85 \times 10^3 \times r_2\ (1.25r_2 - r_2)$$

$$W = 1.335 \times 10^5 \times r_2^2$$

   The mean radius for uniform wear is given by

$$R = \frac{r_1 + r_2}{2} = \frac{1.25\,r_2 + r_2}{2} = 1.125\,r_2$$

   We know that, the torque transmitted is given by

$$T = n\,.\mu.W.R$$

$$265.26 = 2 \times 0.25 \times 1.335 \times 10^5 \times r_2^2 \times 1.125\,r_2$$

$$r_2 = 0.1523\ m = 152.3\,mm$$

$$\boldsymbol{r_1 = 1.25\,r_2 = 190.375\,mm}$$

   To find the axial force required to engage the clutch,

$$W = 2\pi C(r_1 - r_2) = 1.335 \times 10^5 \times r_2^2 = 1.335 \times 10^5 \times (0.1523)^2$$

$$\boldsymbol{W = 3096.6\,N}$$

2. A single plate clutch, with both sides of the plate being effective, is used to transmit power at 1440 rpm. It has outer and inner radii 80 mm and 60 mm respectively. The maximum intensity is limited to 10 x 10⁴ N/m². If the coefficient of friction is 0.3, determine: (i) Total pressure exerted on the plate, and (ii) Power transmitted.

Solution:

The intensity of pressure is maximum at the inner radius,

$$p_{maxi}.r_2 = C$$

$$C = 10 \times 10^4 \times 0.06 = 6000\ N/m$$

The axial thrust transmitted to the frictional surface,

$$W = 2\pi C\ (r_1 - r_2) = 2\pi \times 6000 \times (0.08 - 0.06)$$

$$W = 754\ N$$

We know that, the torque transmitted is given by

$$T = n\,.\mu.W.R$$

The mean radius for uniform wear is given by

$$R = \frac{r_1 + r_2}{2} = \frac{0.08 + 0.06}{2} = 0.07\ m$$

$$T = 2 \times 0.3 \times 754 \times 0.07 = 31.7\ N.m$$

The power transmitted is given by

$$power\ transmitted, P = \frac{2\pi N\ T}{60} = \frac{2\pi \times 1440 \times 31.7}{60}$$

$$P = 4.643\ kW$$

3. Determine the maximum, minimum and average pressure in a plate clutch when the axial force is 5000 N. The outer and inner diameters of the friction surfaces are 200 mm and 100 mm respectively. Assume uniform wear.

Solution:

(i) Maximum pressure

The intensity of pressure is found to be maximum at the inner radius,

$$p_{maxi}.r_2 = C$$

$$C = 50 \times 10^{-3}.p_{maxi}$$

The axial thrust transmitted to the frictional surface,

$$W = 2\pi C\ (r_1 - r_2)$$

$$5000 = 2\pi \times 50 \times 10^{-3}.p_{maxi} \times (100 - 50) \times 10^{-3}$$

$$p_{maxi} = 31.83 \times 10^4\ {}^{N}/_{m^2}$$

(ii) Maximum pressure

The intensity of pressure is minimum at the inner radius,

$$p_{mini}.r_1 = C$$

$$C = 100 \times 10^{-3}.p_{mini}$$

The axial thrust transmitted to the frictional surface,

$$W = 2\pi C\ (r_1 - r_2)$$

$$5000 = 2\pi \times 100 \times 10^{-3}.p_{mini} \times (100 - 50) \times 10^{-3}$$

$$p_{mini} = 15.92 \times 10^4\ {}^{N}/_{m^2}$$

(iii) Average pressure

$$p_{avg} = \frac{W}{\pi[r_1^3 - r_2^3]} = \frac{5000}{\pi[100^3 - 50^3] \times 10^{-6}}$$

$$p_{avg} = 21.22 \times 10^4\ {}^{N}/_{m^2}$$

4. A multi – disc clutch consists of five steel plates and four bronze plates. The inner and outer diameters of friction discs are 75 mm and 150 mm 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.

Solution:

The intensity of pressure is maximum at the inner radius,

$$p_{maxi}.r_2 = C$$

$$C = 0.3 \times 10^6 \times 0.075 = 22500\ N/m$$

The axial thrust transmitted to the frictional surface,

$$W = 2\pi C\ (r_1 - r_2) = 2\pi \times 22500 \times (0.15 - 0.075)$$

$$W = 10602.9\ N$$

We know that, the torque transmitted is given by

$$T = n\,.\mu.W.R$$

The mean radius for uniform wear is given by

$$R = \frac{r_1 + r_2}{2} = \frac{0.15 + 0.075}{2} = 0.113\ m$$

The number of pairs of contact surfaces,

$$n = n_1 + n_2 - 1 = 5 + 4 - 1 = 8$$

Therefore,

$$T = 8 \times 0.1 \times 10602.9 \times 0.113 = 958.5\ N.m$$

The power transmitted is given by

$$power\ transmitted, P = \frac{2\pi N\ T}{60} = \frac{2\pi \times 750 \times 958.5}{60}$$

$$P = 75.3\ kW$$

5. A multi – disc clutch has 3 discs on the driving shaft and two on the driven shaft. The inner and outer diameters of friction disks are 120 mm and 240 mm respectively. The co-efficient of friction is 0.3. Find the maximum axial intensity of pressure between the discs for transmitting 25 kW at 1575 rpm. Assume the uniform wear theory.

The power transmitted is given by

$$power\ transmitted, P = \frac{2\pi N\ T}{60}$$

$$25 \times 10^3 = \frac{2\pi \times 1575 \times T}{60}$$

$$T = 151.58\ Nm$$

The intensity of pressure is maximum at the inner radius,

$$p_{maxi}.r_2 = C$$

$$C = 0.12\ p_{maxi}$$

The axial thrust transmitted to the frictional surface,

$$W = 2\pi C\ (r_1 - r_2)$$

$$W = 2\pi \times 0.12\ p_{maxi} \times (0.24 - 0.12)$$

$$W = 0.0905\ p_{maxi}, \qquad N$$

The mean radius for uniform wear is given by

$$R = \frac{r_1 + r_2}{2} = \frac{0.24 + 0.12}{2} = 0.18\ m$$

We know that, the torque transmitted is given by

$$T = n\,.\mu.W.R$$

The number of pairs of contact surfaces,

$$n = n_1 + n_2 - 1 = 3 + 2 - 1 = 4$$

Therefore,

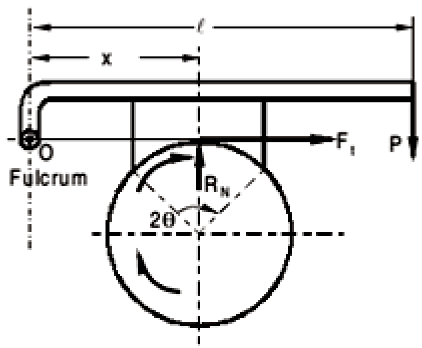
$$151.58 = 4 \times 0.3 \times 0.0905\ p_{maxi} \times 0.18$$

$$p_{maxi} = 7.754 \times 10^3, \qquad N/m^2$$

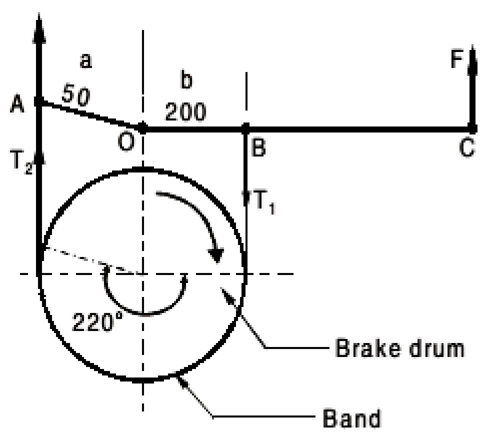
6. A cone clutch with a semi cone angle of 15° transmits 10 kW at 600 rpm. The normal pressure between the surfaces in contact is not to exceed 1000 kN/m². The width of the friction surfaces is half of the mean

diameter. Assume μ = 0.25. Determine: 1) The outer and inner diameter of the plate, and 2) The axial force to engage the clutch.

7. What are the functions of a clutch? Explain with a neat sketch the working of a single plate clutch.
8. What are the various types of clutches? Describe with a neat sketch the working of a multiplate clutch.
9. Describe with the help of a neat sketch the principles of operation of an internal expanding shoe.
10. A single block brake shown in the diagram has the diameter 300 mm. The angle of contact is 120°. The coefficient of friction is 0.3; If the torque transmitted by the brake is 80 N – m, find the force P required. Assume length of the lever (l) = 300 mm and distance between centre of wheel to fulcrum (x) = 150 mm



11. A differential band brake shown in the diagram has a width of 80 mm and thickness of 2 mm. The permissible tensile stress in the band material is limited to 60 N/mm². The Coefficient of friction is 0.25 between drum and the friction lining. The brake drum diameter is 500 mm. Calculate for clockwise direction (a) Tension in the band (b) Actuating force (c) Braking torque (d) Whether the brake is self-locking or not.



12. A double shoe brake shown in the diagram, is capable of absorbing a torque of 1500 N-m. Brake drum diameter = 400 mm; Angle of contact for each shoe 110°; μ = 0.4. Find: (a) Spring force necessary to set the brake (b) Width of the shoe brake, if the bearing pressure on the lining material is not to exceed 0.5 N/mm²

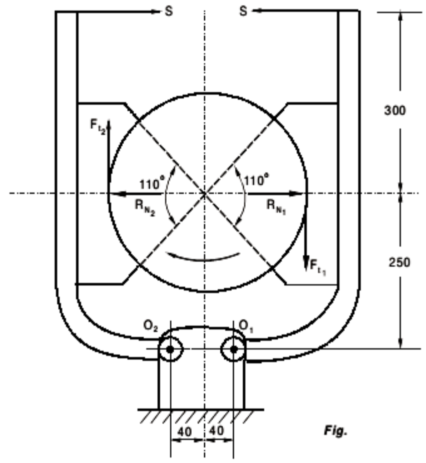


Fig.

MULTIPLE CHOICE QUSTIONS

	QUESTION	OPTION 1	OPTION 2	OPTION 3	OPTION 4	ANSWER
1	Pitch circle diameter is equal to the product of………..	Flat pivot bearing	Flat collar bearing	Conical pivot bearing	Truncated conical pivot bearing	Flat collar bearing
2	In a disc clutch, if there are n_1 number of discs on the driving shaft and n_2 number of discs on the driven shaft, then the number of pairs of contact surfaces will be	n_1+n_2	n_1+n_2+1	n_1+n_2-1	n_1+n_2-2	n_1+n_2-1
3	The frictional torque transmitted by a cone clutch is same as that of	Flat pivot bearing	Flat collar bearing	Conical pivot bearing	Truncated conical pivot bearing	Truncated conical pivot bearing
4	Claw clutch is	Positive drive because there is no slip in power transmission	Mostly preferred in automobiles because of smooth engagement	A friction drive because it consists of 2 pairs of friction surfaces	Seldom used in machine tools because it transmit power in one direction only	Positive drive because there is no slip in power transmission
5	The cone clutches	Have become obsolete because of small cone angles	Have become obsolete because of small cone angles	Are used widely because it is a positive drive	Has two pairs of friction discs because it is a multiple disk drive	Have become obsolete because of small cone angles

6	The dynamic friction is the friction experienced by a body, when the body	Is in motion	Is at rest	Just begins to slide over the surface of the other body	None of these	Is in motion
7	Which of the following statements regarding laws governing the friction between dry surfaces are correct?	The frictional force is dependent on the materials of the contact surfaces	The frictional force is directly proportional to the normal force	The frictional force is independent of the area of contact	All of these	All of these
8	In a screw jack the load cup	Is made as integral part because it reduces the friction	is made separate from the spindle because it will prevent the rotation of load being lifted	Enhancers the capacity of screw jack became it improves the mechanical advantages	Prevents the toppling of load because it rotates along with the screw rod	Is made separate from the spindle because it will prevent the rotation of load being lifted
9	In a screw jack, the effort required to lift the load W is given by	P = W tan (α - φ)	P = W tan (α + φ)	P = W tan (φ - α)	P = W cos (α + φ)	P = W tan (α + φ)
10	(S1) The static friction is independent of the area of contact, between the two surface	S1 is right	S2 is right	Both S1 & S2 are right	Both S1 & S2 are wrong	Both S1 & S2 are right
11	In a screw jack, the effort required to lower the load W is given by	P = W tan (α - φ)	P = W tan (α + φ)	P = W tan (φ - α)	P = W cos (α + φ)	P = W tan (φ - α)
12	Frictional torque for square thread at mean radius r while raising load W is given by	T = W.r tan (φ - α)	T = W.r tan (φ + α)	T = W.r tan α	T = W.r tan φ	T = W.r tan (φ + α)
13	The frictional torque transmitted in a conical pivot bearing, considering uniform wear, is	1/2 μ *W R cosec α*	2/3 μ *W R cosec α*	3/4 μ *W R cosec α*	μ *W R cosec α*	1/2 μ *W R cosec α*
14	The frictional torque transmitted by a disc or plat clutch is same as that of	Flat pivot bearing	Flat collar bearing	Conical pivot bearing	Truncated conical pivot bearing	Flat collar bearing
15	Brake is a device used for bringing a moving body	To rest	To retard the motion	To keep it in a state of rest against the external forces	All of these	All of these

16	Material used for brake lining should not have	High heat dissipation capacity	Low heat resistance	High strength	High coefficient of friction	Low heat resistance
17	In a Geneva wheel mechanism, the number of slots in the wheel is 4. For one full revolution of the wheel, the driver disc has to make	16 revolutions	¼ revolutions	4 revolutions	8 revolutions	4 revolutions
18	A Geneva wheel has 6 slots. Driving crank radius is 100 mm. the distance between centres is	200 mm	100 mm	50 mm	141.44 mm	200 mm
19	Properties of a good brake lining material are	High heat dissipation capacity	High coefficient of friction	High strength	All of these	All of these
20	Reduce the impact load in Geneva wheel…	Increase the number of slots	Reduces the number of slots	Increase the pin diameter	Increase the pin diameter	Increase the number of slots
21	Dwell period (in caser of cam and follower) is the tim	That cam rotates	During which the follower moves from its lowest position to highest posit	That cam not rotates.	None of these	None of these
22	The dynamic friction is the friction experienced by a body, when the body	Is rolling	Is at rest	Just begins to slide over the surface of the other body	None of these	None of these
23	To determine the geometric dimensions of a Geneva wheel, the most important parameter to be known are	Diameter of the Geneva wheel and its shaft.	Slot length and width	Diameter of the driving disc and its shafts	Number of slots and distance between centers.	Number of slots and distance between centers.
24	The pawl of the ratchet wheel mechanism is subjected to	Direct load	Bending moment	Twisting moment	Direct load & Bending moment	Direct load & Bending moment
25	The factor affects the life of the cam is	Base circle	Cam angle	Hub size	None of these	None of these