MODULE-2

SPUR GEARS AND PARALLEL AXIS HELICAL GEARS

INTRODUCTION:
Mechanical drives are divided into two types:

1. Those that transmit power through friction, such as belt and rope drives.
2. Drives that convey power through engagement, such as chain or gear drives.

However, the selection of an appropriate mechanical drive for a given application is determined by a variety of parameters, including center distance, velocity ratio, shifting arrangement, maintenance, and cost.

GEAR DRIVES
Gears are toothed wheels that transmit power and motion from one shaft to another through the sequential engagement of teeth.
1. The central distance between the shafts is rather short.
2. It can transmit extremely high power.
3. It's a positive, and the velocity ratio is constant.
4. It can convey motion at extremely low speeds.

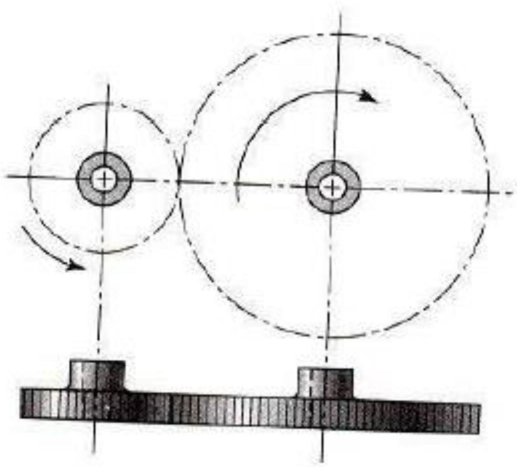
CLASSIFICATION OF GEARS:

1. Spur Gears

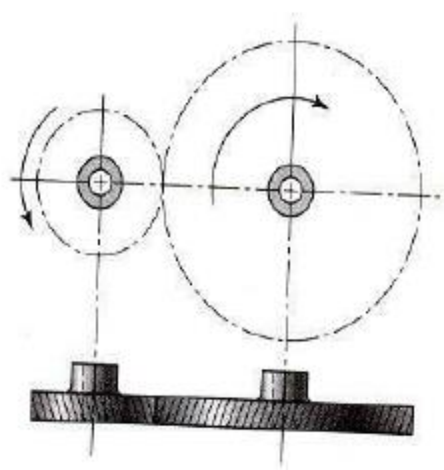
2. Helical gears

3. Bevel gears and

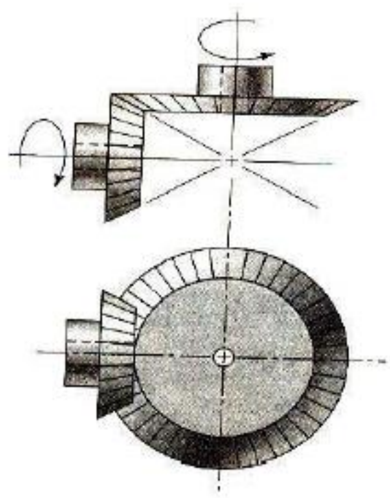
4. Worm Gears



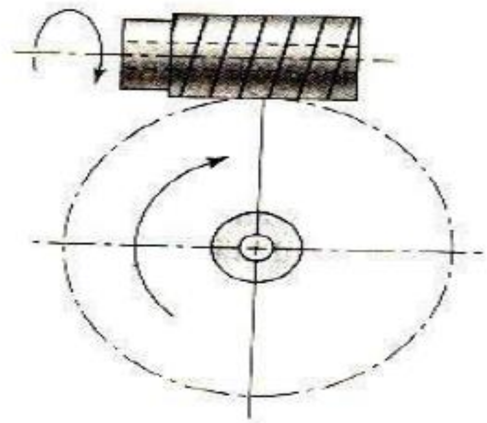
Spur Gear



Helical Gear



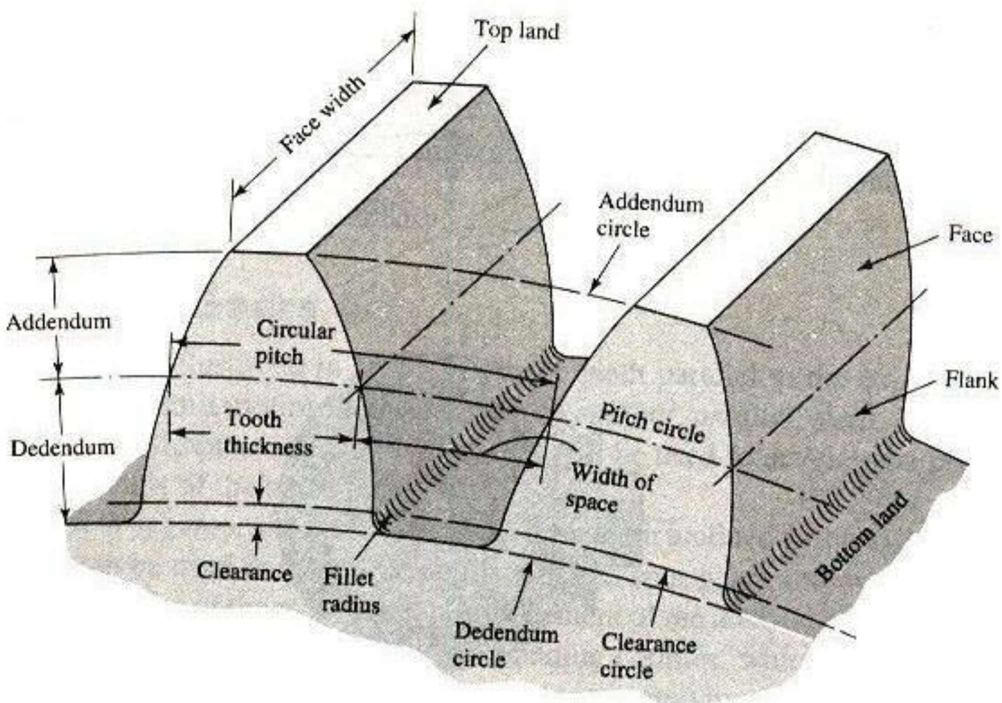
Bevel Gear



Worm Gear Set

Nomenclature

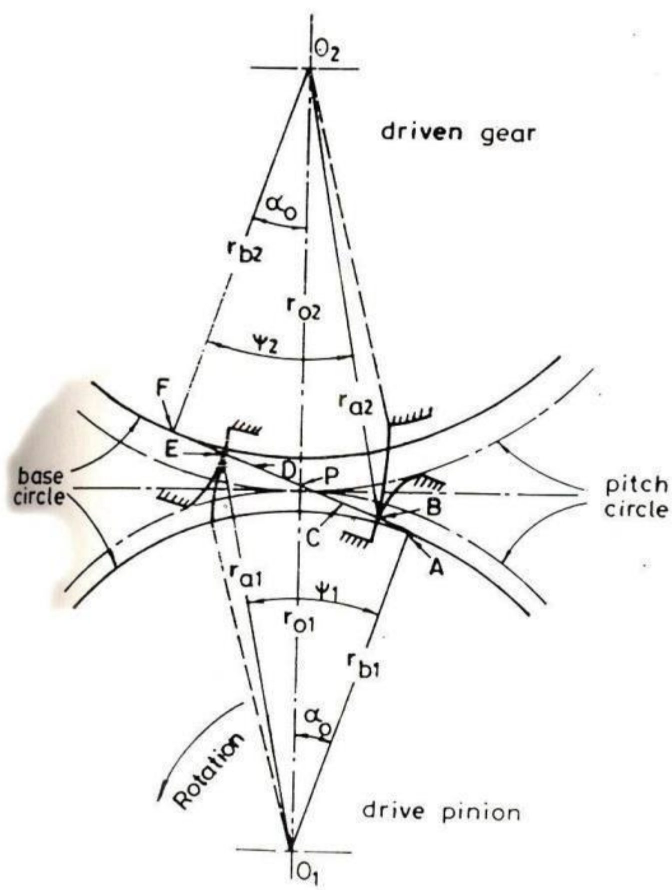
Spur gears are designed to convey rotary motion across parallel shafts. They are normally cylindrical in shape, with straight teeth that are parallel to the axis of rotation. In a pair of gears, the bigger is commonly termed the gear, and the smaller is called the pinion.



Nomenclature of Spur Gear

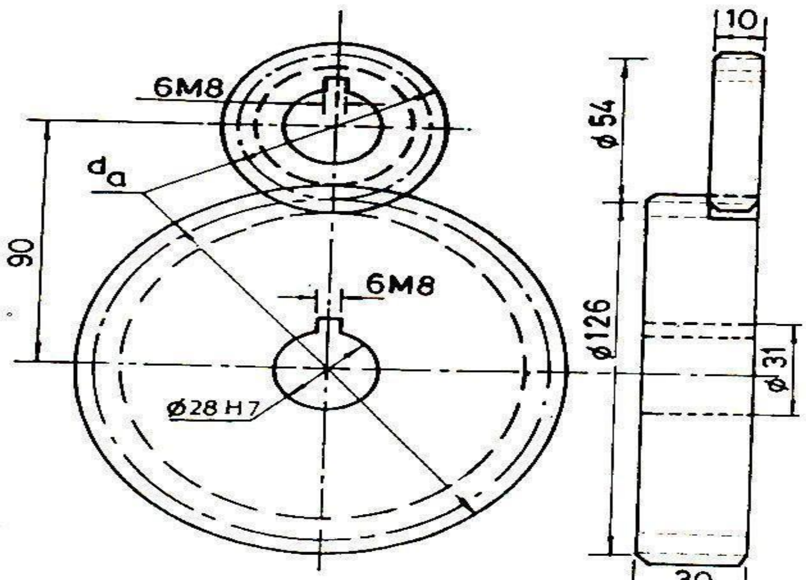
1. Pitch Surface: The pitch levels of the gears were notional planes, cylinders, or cones that may roll together without slipping.
2. Pitch circle: This is the theoretical circle on which computations are often based. It is an imagined circle that rolls while slipping along the pitch circle of a mating gear. Furthermore, the pitch circles of a mated gear are tangent to one another.
3. Pitch circle diameter: This is the pitch circle's diameter. Typically, the size of the gear is determined by the pitch circle diameter. This is signified by "d".
4. The top land refers to the outermost layer of the highest point of the gear tooth.
5. Base circle: The base circle is an imaginary circle that generates the involute curve of the tooth profile. (The base circles of two mating gears are parallel to the pressure line).
6. Addendum. The Addendum represents the radial distance among the pitch and addendum circles. The addendum denotes the height of the tooth above the pitch circle.
6. Dedendum: The dedendum refers to the axial distance between the pitch and dedendum circles. Dedendum refers to the tooth's depth beneath the pitch circle.
7. Whole Depth: The entire depth is the overall depth of the tooth space, calculated as the sum of the addendum and dedendum.

Failure Map of Involutes Gears



Specification of Test Pinions and Gears

Variable	Symbol	Unit	Values of variables used in the experiments		
			Pinion		Gears
Module	m	(mm)		3.0	
Pressure angle	α_o	(deg)		20°	
Number of teeth	z	(--)	18		42
Pitch circle diameter	d	(mm)	54.0		126.0
Centre distance	a_o	(mm)		90.0	
Addendum circle diameter	d_a	(mm)	60.0		132.0
Root circle diameter	d_r	(mm)	46.5		118.5
Face width	B	(mm)	10.0		30.0



$\alpha_o = 20°;\ m = 3\ mm;\ z_1 = 1.8;\ z_2 = 42$

Design Considerations for a Gear Drive

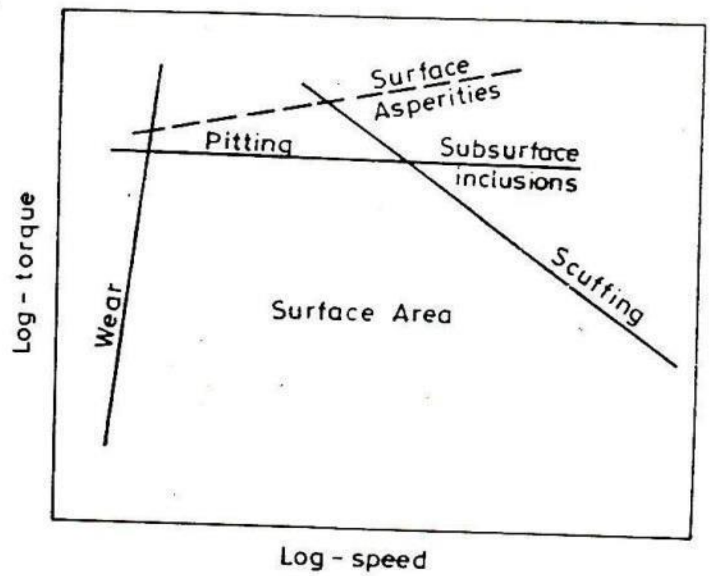
Typically, the following information is provided while designing a gear drive:

i. Transmission power

ii. Driving gear speed

iii. Driven gear velocity ratio

iv. The Center Distance



The requirements that follow must be addressed while designing a gear drive:

(a) The gear teeth should have adequate strength to not fail under static or dynamic loading during typical operating circumstances.
(b) The gear teeth ought to possess wearing properties that allow them to last for an appropriate amount of time.
(c) The utilization of area and material should be advised. The gear alignment and shaft deflections must be considered since they affect the efficiency of the gears.
(d) The gears must be well lubricated.

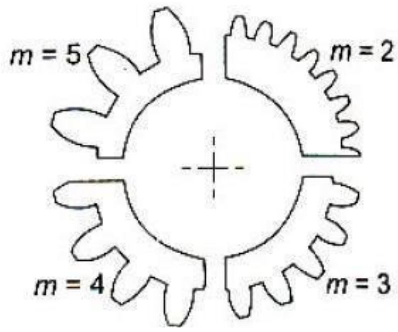
Selection of Gears:
Designing a gear drive begins with selecting the appropriate gear type for the application.
The elements to consider when deciding on the type of gear are

The overall layout of shafts
Speed Ratio
Power to be transmitted.
Input speed and cost.

Spur and Helical Gears: When the shafts are parallel.
Bevel gears connect at right angles,
While Worm & Worm gears have perpendicular shafts that do not intersect. Crossed helical gears are used in a specific case where the axes of the two shafts are not intersecting or perpendicular.

Law of Gearing:
The Basic Law of Gearing states "To achieve a constant velocity ratio, the common normal to both profiles at the point of contact must always pass through a set location known as the pitch point.

Module: The module sets the size of the gear teeth. Figure depicts the exact diameters of gear teeth for four separate modules. It is noted that as the number of modules increases, so does the size of the gear teeth. The module can be thought of as an index for the size of the gear tooth.



Recommended Series of Modules (mm)

Preferred (1)	Choice 2 (2)	Choice 3 (3)	Preferred (1)	Choice 2 (2)	Choice 3 (3)
1			8	7	(6.5)
1.25	1.125		10	9	
1.5	1.375		12	11	
2	1.75		16	14	
2.5	2.25		20	18	
3	2.75	(3.25)	25	22	
4	3.5		32	28	
5	4.5	(3.75)	40	36	
6	5.5		50	45	

Note: The modules given in the above table apply to spur and helical gears. In case of helical gears and double helical gears, the modules represent normal modules

The module given under choice 1, is always preferred. If that is not possible under certain circumstances module under choice 2, can be selected.

Standard proportions of gear tooth in terms of module m, for 20º full depth system. Addendum = m
Dedendum = 1.25 m
Clearance (c) = 0.25 m
Working depth = 2 m
Whole depth = 2.25 m
Tooth thickness = 1.5708 m=1.5708 m
Tooth space = 1.5708 m
Fillet radius = 0.4 m

Standard tooth proportions for involutes spur gear

Gear Terms	Circular pitch p	Diametral pitch P	Module m
Addendum	0.3183 p	1/P	m
Dedendum	0.3977 p	1.25/P	1.25 m
Tooth thickness	0.5 p	1.5708/P	1.5708 m
Tooth space	0.5 p	1.5708/P	1.5708 m
Working depth	0.6366 p	2/P	2 m
Whole depth	0.7160 p	2.25/P	2.25 m
Clearance	0.0794 p	0.25/P	0.25 m
Pitch diameter	zp/π	z/P	zm
Outside diameter	$(z+2)p/\pi$	(z+2)/P	(z+2) m
Root diameter	$(z - 2.5)p/\pi$	(z – 2.5)/P	(z – 2.5) m
Fillet radius	0.1273p	0.4/P	0.4 m

Selection of Material:

- The gear tooth's load carrying capability is determined by the material's ultimate tensile or yield strength.
- When a gear tooth is subjected to variable forces, its endurance strength is the decisive factor.
- The gear materials should be strong enough to withstand tooth breaking.
- Gear tooth size are often determined by wear rating, not strength rating.
- Resistance to wear is determined by alloying elements, grain size, carbon content, and surface hardness.
- The gear material must have adequate surface endurance strength to prevent failure due to destructive pitting.
- To avoid score failure, high-speed power transmission requires high sliding velocities and a low coefficient of friction.
- The degree of thermal distortion or wrapping during the heat treatment process is a significant issue in gear applications.
- Wrapping concentrates the load on one corner of the gear tooth.
- Alloy steels are superior to plain carbon steel in this respect (Thermal distortion)

Load-Distribution Factor Km (KH)

The load-distribution component altered the stress calculations to reflect the non-uniform distribution of load along the line of contact. The objective is to situate the gear "mid span" between two bearings at zero slope points when the load is applied. Unfortunately, this is not always possible. The following procedure applies to

• Net face width to pinion pitch diameter ratio $F/d \leq 2$

• Gear elements mounted between the bearings

• Face widths up to 40 in

• Contact, when loaded, across the full width of the narrowest member

The load-distribution factor under these conditions is currently given by the *face load* distribution factor, Cmf , where

$$C_{mc} = \begin{cases} 1 & \text{for uncrowned teeth} \\ 0.8 & \text{for crowned teeth} \end{cases}$$

$$C_{pf} = \begin{cases} \dfrac{F}{10d} - 0.025 & F \leq 1 \text{ in} \\ \dfrac{F}{10d} - 0.0375 + 0.0125F & 1 < F \leq 17 \text{ in} \\ \dfrac{F}{10d} - 0.1109 + 0.0207F - 0.000\,228F^2 & 17 < F \leq 40 \text{ in} \end{cases}$$

Note that for values of $F/(10d) < 0.05$, $F/(10d) = 0.05$ is used.

$$C_{pm} = \begin{cases} 1 & \text{for straddle-mounted pinion with } S_1/S < 0.175 \\ 1.1 & \text{for straddle-mounted pinion with } S_1/S \geq 0.175 \end{cases}$$

$$C_{ma} = A + BF + CF^2 \qquad \text{(see Table 14–9 for values of } A, B, \text{ and } C\text{)}$$

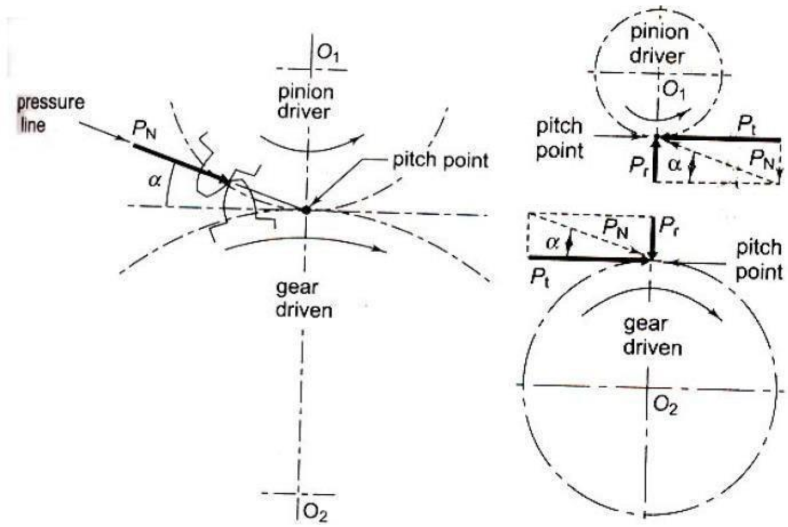
$$C_e = \begin{cases} 0.8 & \text{for gearing adjusted at assembly, or compatibility} \\ & \text{is improved by lapping, or both} \\ 1 & \text{for all other conditions} \end{cases}$$

Force Analysis

In spur gearing, power is delivered through a force applied by the driving gear's tooth on the driven gear's meshing tooth along a pressure line. (The driving pinion exerts force PN on the tooth of the driven gear).

According to the fundamental law of gear, the resultant force, PN, always acts along the pressure line.

This resultant force PN, can be resolved into two components – tangential component Pt and radial components Pr at the pitch point.



The tangent component Pt is an important component (load) since it controls the scale of the torque and thus the power that is transmitted.

The radial portion of Pr serves no beneficial use (it is a separation force) and is always pointed towards the center of the gear.

The torque transmitted by the gear is given by

$$M_t = \frac{P \times 60}{2\,\pi N_1} \; N - m$$

Mt = Torque transmitted gears (N- m)
Power in KW = Power transmitted by gears
N1 = Speed of rotation (rev / mn)
The tangential component Ft acts at the pitch circle radius.

$$\therefore \quad M_t = F_t \frac{d}{2}$$

OR

$$F_t = \frac{2M_t}{d}$$

Where,
Mt = Torque transmitted gears N- mm d = Pitch Circle diameter, mm
Further, we know,
Power transmitted by gears = 2 πN Mt /60 (KW)
Where Fr= Ft tanα
Resultant force,

$$FN = \frac{F_t}{Cos\alpha}$$

The previous examination of gear tooth force is predicated on the following assumption.
i) As the location of contact moves, the amount of the resultant force, PN, changes. This effect is ignored.
ii) It presumes that only one set of teeth carries the entire burden. At times, two pairs are in contact and sharing the weight. This component is also disregarded.
iii) This study is valid in static settings, such as when the gears are running at extremely low speeds. In practice, dynamic forces exist alongside power transmission forces.

For gear tooth forces, it is always necessary to determine the amount and direction of two components. The values are determined using formulae.

$$M_t = \frac{P \times 60}{2\pi N_1}$$

$$F_t = \frac{2M_t}{d_1}$$

Furthermore, the direction of two parts Ft and Fr is determined by drawing a free body diagram. The smallest number of teeth on a pinion to prevent interference is provided by

$$Z_{\min} = \frac{2}{\sin^2 \alpha}$$

For a 20° full depth involute system, it is always reasonable to estimate the number of teeth is 18 or 20. After determining the number of teeth on the pinion, the velocity ratio is used to compute the number of teeth on the gear.

Face Width:

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

When designing gears, it is necessary to convey the face width as a number of modules. In practice, the ideal range for face width is 9.5 m ≤ b ≤ 12.5m.

In general, face width is expected to be ten times module.

$$\therefore \quad \boxed{b = 12.5m}$$

Table Standard proportions of gear systems.

S. No.	*Particulars*	*$14\frac{1}{2}°$ composite or full depth involute system*	*20° full depth involute system*	*20° stub involute system*
1.	Addendum	1*m*	1*m*	0.8 *m*
2.	Dedendum	1.25 *m*	1.25 *m*	1 *m*
3.	Working depth	2 *m*	2 *m*	1.60 *m*
4.	Minimum total depth	2.25 *m*	2.25 *m*	1.80 *m*
5.	Tooth thickness	1.5708 *m*	1.5708 *m*	1.5708 *m*
6.	Minimum clearance	0.25 *m*	0.25 *m*	0.2 *m*
7.	Fillet radius at root	0.4 *m*	0.4 *m*	0.4 *m*