22. .... gauge checks the maximum metal condition and .... gauge checks the minimum metal condition.

(B) Match the two parts :

(*a*)

	I		II
(*a*)	Limits	(*i*)	The algebraic difference between the actual size and the corresponding basic size.
(*b*)	Fits	(*ii*)	The permissible variation in size.
(*c*)	Tolerance	(*iii*)	Prescribed difference between the dimensions of making parts to perform specific function.
(*d*)	Allowance	(*iv*)	The ranges of permissible variation in dimension of a part.
(*e*)	Deviation	(*v*)	Degree of lightness and looseness between the mating parts.

(*b*)

(*a*)	*H*7*g*6	(*i*)	Heavy press fit
(*b*)	*H*7*h*6	(*ii*)	close running fit
(*c*)	*H*7*n*6	(*iii*)	precision running firm
(*c*)	*H*7*s*6	(*iv*)	push fit (transition fit)

(C) Give reasons :

(*i*) Unilateral system is preferred in interchangeable manufacture.
(*ii*) Tight tolerances increases the cost of production.
(*iii*) Go end of the plug gauge is made longer than the No-Go end.
(*iv*) The hole basis system is most commonly used for obtaining different types of fits.
(*v*) Limit gauges are used in mass production.

10

GEAR METROLOGY

Introduction

Gears are toothed wheels commonly used to transmit power or motion, from one shaft to another, without slip. They are thus positive in action and provide constant velocity ratio, a feature which most machinery requires.

The transmission efficiency in case of gears is 99 percent. However, the errors present in the gears can interfere with the efficient operations of the equipment using them, *e.g.*, machine tools, vehicles etc. The accuracy of gears, both as to their geometrical forms, size has a considerable effect on smoothness of operation, freedom from noise and length of working life. In addition to this for higher efficiency the gears should be perfectly mounted on perfect shafts running in perfect bearings. It is thus obvious that major factor which decides the accuracy of gearing is the precision with which gears are manufactured.

Before use, therefore, it is essential to test and measure the gears precisely.

The distance between the two shafts should be just sufficient in order to obtain correct meshing of gear teeth. If the distance is much more, a complete set of gears, known as gear train, is used. The velocity in gear drive should not exceed 250 m/min.

The most commonly used forms of gear teeth are :

(*i*) Involute and (*ii*) Cycloidal

The cycloidal gears are not generally used in modern engineering, but used for some crude purposes where heavy and impact loads come on the machine. Involute gears are now almost entirely used for general purpose in precision engineering.

The involute gears also called as straight tooth or spur gears possess the following advantages over the cycloidal gears.

(*i*) The variations in the centre distance between two gears have no effect on the velocity ratio between a pair of involute gears.
(*ii*) The pressure angle is constant.
(*iii*) The involute rack has straight teeth. Thus the complex involute form can be generated by using a relatively simple cutter.
(*iv*) All the gears having the same pitch and pressure angle work correctly together.
(*v*) The face and flank of a tooth forms a continuous curve.

Types of Gears

The gears which are most commonly used for power transmission are :

(*i*) **Spur gears**. These gears have their teeth parallel to the centre line of the gear. These are used for transmitting the power between parallel shafts.

(*ii*) **Helical gears.** In these gears, the teeth are cut at an angle with the axis of gear, forming the part of helix around the gear. These gears are used for transmitting the power between parallel shafts as well as non-parallel and non-intersecting shafts. These are stronger and quicker in operation, as compared to spur gears. These gears run more smoothly and more quietly at high speeds than spur gears. However, some power is lost because of end thrust, and provision must be made to compensate for this thrust in bearings.

(*iii*) **Bevel gears.** In these gears the teeth are cut on the conical surface. These are used for connecting the driving shaft and driven shaft whose axes intersect and are generally at right angles. Bevel gears are made with either straight or spiral teeth when the shafts are at right angles and the two bevel gears have the same size, then such a pair of gears is known mitre gears.

(*iv*) **Worm and worm wheel.** These are used for connecting the driving shaft and driven shaft whose axes are non-parallel and non-intersecting. Worm gearing is used where a large speed reduction is required.

(*v*) **Rack and Pinion.** Rack gears are straight spur gears with infinite radius. Pinion is a small spur gear. Rack and pinion are used in combination to convert rotary motion into reciprocating motion or reciprocating motion into rotary motion.

Internal gears. These gears are employed for transmitting the power when the distance between two parallel shafts to be connected is less. Sufficient reduction in speed can be obtained in this case.

Since the involute profile gears are widely used, it is necessary to consider the involute curve in some details.

The Involute Curve

An involute curve is defined as the locus of a point on straight line which rolls around a cylinder without slipping. It can also be defined as a path described by a point on a thin inextensible cord when the latter is unwound from a given curve.

From the figure 10.1 it is seen that the length of the generator is equal to the arc length of the base circle from the point of tangency to the origin of the involute at A

i.e., $A_1B_1 = \text{arc } AB_1$

$A_2B_2 = \text{arc } AB_2$ and so on

It is also clear that the tangent to the involute at any point, e.g., A_2 is perpendicular to the generator at that point.

The shape of the involute curve entirely depends upon the diameter of base circle from which the involute is generated. The curvature of the involute goes on decreasing as the base circle diameter goes on increasing and finally when the limit is reached for a base circle of infinite diameter, the involute becomes straight line.

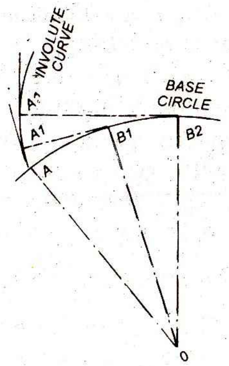

Fig. 10.1. The involute curve

Gear Terminology. The important terms used in connection with the measurement of gears are defined as follows.

Addendum Circle. The circle which limits the top of the gear teeth and represents its maximum diameter is called addendum circle.

Addendum. It is the radial height of the tool from the pitch circle to the tip of the tooth.

Dedendum. It is the radial depth of tooth from the pitch circle to the root of the tooth.

Clearance. The radial distance from the top of the tooth to the bottom of the tooth space in the mating gears is called as clearance.

Face of tooth. It is the side surface of the tooth above the pitch circle, perpendicular to the plane of the gear.

Flank of tooth. It is the side surface of the tooth below the pitch circle, perpendicular to the plane of the gear.

Tooth thickness. It is the thickness of the tooth, measured along the pitch circle.

Whole depth. It is the sum of the addendum and dedendum of a tooth.

Working depth. It is the greatest depth to which a tooth of one gear extends into the tooth space of a mating gear.

Working depth = Addendum + Dedendum − Clearance

Dedendum Circle. It is the circle which contains the bottoms of the tooth spaces. It is also called a root circle. Its diameter is the root diameter.

Pitch Circle. In every pair of gears in mesh, the two circles representing the two plain wheels in contact are always assumed to exist. Each of these circles is called a pitch circle. Its diameter is the pitch circle diameter.

Pitch point. It is the point of contact between the pitch circles of two gears in mesh.

Crest of tooth. It is the outside surface of the tooth, perpendicular to the plane of the gear.

Root of the tooth. It is the junction of the tooth with the material at the bottom of the tooth space.

Base Circle. It occurs only in involute gears and is the circle from which the involute curve of the tooth profile is generated.

Circular Pitch. It is the distance measured along the pitch circle, from a point on one tooth to a corresponding point on the adjacent tooth.

For a spur gear, Circular pitch, $P_c = \frac{\pi \times d}{T} = \pi m$

where, d = pitch circle diameter

T = number of teeth

and m = module .

Diametral Pitch. It is the number of teeth per unit length of the pitch circle diameter

$$\text{Diametral Pitch} = \frac{\text{Number of teeth}}{\text{Pitch circle diameter}} = \frac{T}{d}$$

Module. It is the linear distance in *mm* that each tooth of the gear would occupy, if the gear teeth were spaced along the pitch diameter.

$$\text{Module} = \frac{\text{Pitch circle diameter}}{\text{Number of teeth}} = \frac{d}{T}$$

Thus, it is the reciprocal of Diametral pitch.

Pressure angle or angle of obliquity. It is the angle which the common normal to the two teeth at the point of contact makes with the common tangent to the two pitch circles at the pitch point.

The standard proportions adopted by the Indian Standard System of the elements of an involute spur gear are given in Table 10.1.

Table 10.1

Name of tooth element	*Symbol*	*Gear tooth proportions (pressure angle 20°)*
Pitch diameter	d	2 m
Addendum	ha	m
Dedendum	hf	1.25 m
Working depth	$2\ ha$	2 m
Tooth depth	h	2.25 m
Outside diameter	$d' + 2\ ha$	$m(2 + 2)$
Tooth thickness	s	1.5708 m
Radius of fillet	r	0.4 m to 0.45 m
Clearance	$hf–ha$	0.25 m

The recommended series of modules adopted by the Indian Standard System are 1, 1.25, 1.5, 2, 2.5, 3, 4, 5, 6, 8, 10, 12, 16 and 20. The modules 1.125, 1.375, 1.75, 2.25, 3.5, 4.5, 5.5, 7, 9, 11, 14 and 18 are of second choice.

The recommended series of Diametral pitches are 20, 16, 12, 10, 8, 7, 6, 5, 4, 3, $2\frac{1}{2}$, 2, $1\frac{1}{2}$, $1\frac{1}{4}$ and 1.

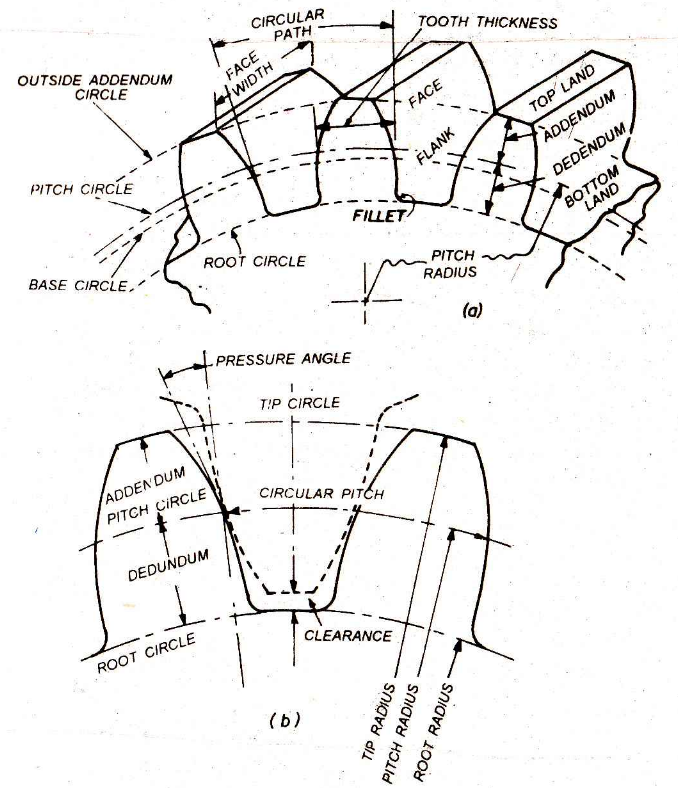

Fig. 10.2. Gear terminology for spur gear

Helical Gear elements

Certain additional elements related to helical gears are as described below :

Helix. A helix can be defined as a curve generated by the path of a point moving along the curved surface of a cylinder such that its movement parallel to and around the axis of the cylinder is uniform in each revolution.

Lead. The lead of the helix is defined as the distance travelled, measured parallel to the axis of rotation, in one revolution by a point moving along the curve.

Helix angle. It is the acute angle between the tangent to a helix and the straight generator of the cylinder on which it lies. Helix angle is given by :

$$\tan \beta = \frac{\pi D}{l}$$

β = helix angle

D = diameter of the work

l = lead of the helix.

For helical gear, D = the pitch circle diameter. For helical flutes, D = the mean diameter.

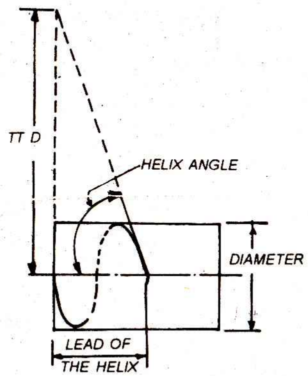

Fig. 10.3. Helix angle

Lead angle. It is the angle between the tangent to the helix and plane perpendicular to the axis of cylinder.

Back Lash. The diameter through which a gear can be rotated to bring its non-working flank in contact with the teeth of mating gear.

Bevel Gear elements. Bevel gears are used when two shafts, the axes of which intersect, are to be connected by gearing. Bevel gear elements are shown in Fig. 10.4.

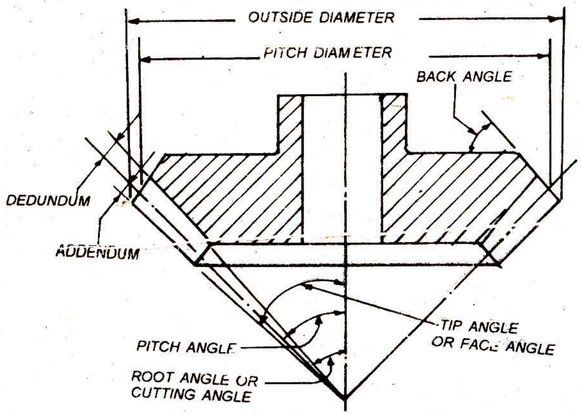

Fig. 10.4. Bevel gear elements

Spur Gear Measurement

The analytical inspection of the gears consists in determining the following teeth elements in which manufacturing errors may be present.

1. Runout
2. Pitch
3. Profile
4. Lead

5. Back lash
6. Tooth thickness
7. Concentricity
8. Alignment
9. Composite errors.

(1) *Runout.* Runout means the eccentricity in the pitch circle. Gears that are eccentric tend to have periodic variation in sound (vibration) during each revolution. A badly eccentric tooth may cause an abrupt gear failure. The runout in the gears is measured by means of gear eccentricity testers. The gear is held in the mandrel in the centres. The dial indicator of the tester possesses special tip depending upon the module of the gear to be checked. The tip is inserted in between the tooth spaces. The gear is rotated tooth by tooth. The maximum variation is noted from the dial indicator reading which gives the runout of the gear. The runout is twice the eccentricity.

Pitch Measurement. Errors in the tooth spacing or pitch of gear may be measured by :

(a) measuring the distance from a point on one tooth to a point on the next tooth (step by step method)

(b) measuring the position of a suitable point on a tooth after the gear has been indexed through a suitable angle (Direct angular measurement).

Tooth to Tooth Pitch Measurement (Step by step method)

This method involves the measurement of variations in pitch between successive teeth of the gears. The differences obtained in this way are relative to the tooth spacing of the arbitrarily chosen datum position.

The portable hand-hel instrument which measures the base pitch errors is shown in Fig. 10.5.

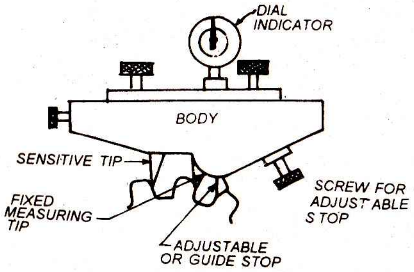

Fig. 10. 5. Portable hand-held base pitch measuring instrument

The instrument has three tips. One is fixed measuring tip, the second is sensitive tip whose position can be adjusted by a screw and the further movement of it is transmitted through a leverage system to the dial indicator. The third tip is adjustable or guide stop. It is meant for the stability of the instrument and its position can also be adjusted by a screw.

The distance between the fixed and sensitive tip is set to be equivalent to the base pitch of the gear with the help of slip gauges. This properly set instrument is applied to the gear so that all the three tips contact the tooth profile. The reading on the dial indicator is the error in the base pitch.

Another method is to use two dial gauges on adjacent teeth with the gear mounted in centres. The gear is indexed through successive pitches to give a constant reading on dial A. Any change in the reading on dial B indicates that pitch errors are present. The actual error can be determined by deducting the individual reading on a dial B from the mean of the readings.

Direct Angular Measurement. The simplest method of determining pitch errors is to set a dial gauge against a tooth and note the reading. If gear is not indexed through the angular pitch the reading differs from the original reading. The difference between these is the cumulative pitch error. The problem is to index through the exact angular pitch because an error in indexing will induce an error in pitch. It is, therefore, necessary to use suitable indexing device to obtain accurate results.

Profile checking. The following methods are used to check the involute profile of a spur gear.

(1) *Optical projection Method.* In this method, the profile of the gear under test is magnified by optical means and projected on the screen. It is then compared with master profile. This method is quick and suitable for checking the profile of small thin instrument gears.

(2) *Using involute measuring machine.* The principle of involute measuring machine is illustrated in Fig. 10.6. If a straight edge is rolled around a base circle without slipping the styles of the dial gauge attached to the straight edge would traverse a true involute.

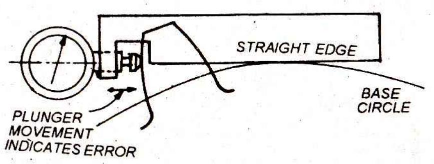

Fig. 10.6. A possible arrangement for profile measurement

In this method the gear to be tested is held on a mandrel. A ground circular disc having exactly the same diameter as the base circle of gear under test is also mounted on the mandrel. The straight edge of the instrument is brought in contact with the base circle of the disc. As the gear and disc are rotated, the straight edge moves over the disc without slip. The stylus of the dial gauge is brought in contact with a tooth profile. When the gear and disc are rotated, the stylus moves over the tooth profit and the deviations from the true involute profile are indicated on the dial gauge.

This method is rapid and accurate up to 0.002 mm.

(3) Tooth displacement method. When the involute measuring machines are not available, the profile of large gear is checked by using a dividing head and a vertical measuring machine (height gauge). The gear is rotated through small angular increments and the readings of the vertical measuring machine or the height gauge are compared with the theoretically calculated values at about five to ten places along the tooth flank and the required increments of angular setting may be established by trial. The calculated values of the vertical setting may be calculated by reference to Figs. 10.7 (*a*), (*b*), and (*c*), which show three angular positions.

$$L_1 - L_2 = rp \cos \phi\, (\theta_1 - \theta_2)$$

$$L_3 - L_1 = rp \cos \phi\, (\theta_3 - \theta_1)$$

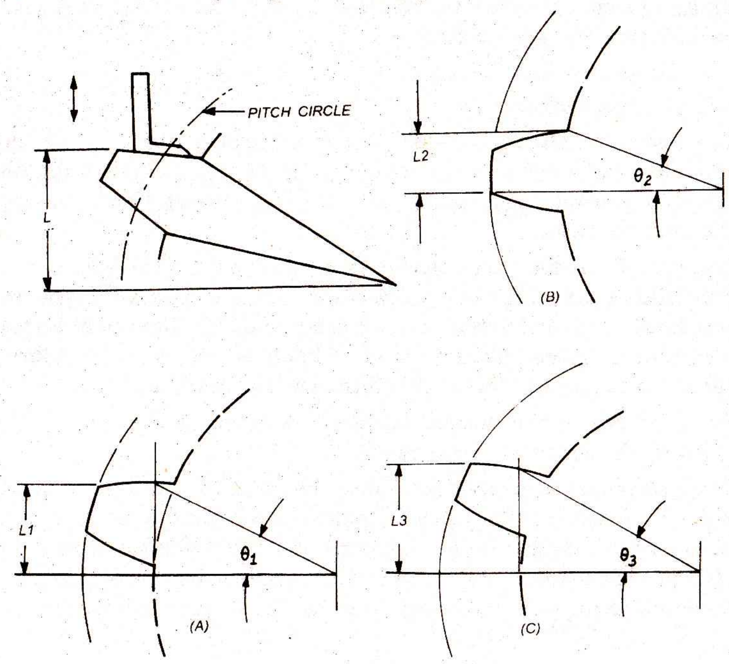

Fig. 10.7. Tooth displacement method

This method is very time consuming but is best suited for calibration of master involute, so it is used only for very precision components.

Lead checking. Lead is the axial advance of a helix for one complete turn, as in the threads of cylindrical worms and teeth of helical gears.

Control of lead is necessary to ensure adequate contact across the face with gear and pinion are properly mounted with axes parallel and in the same plane.

Lead may be checked by lead checking instruments. The instrument advances a probe along a tooth surface, parallel to the axis while the gear rotates in a specified time revolution, based on the specified lead.

Backlash checking. Backlash in gears is the play between measuring tooth surfaces. Backlash is defined as the amount by which tooth space exceeds the thickness of an engaging tooth. Numerical values of backlash are measured at the tightest point of mesh on the pitch circle, in a direction normal to the tooth surface.

However, a tight mesh is objectionable, because of gear sound, increased power losses, overheating and rupture of lubricant film, overloaded bearing and premature gear failure. Hence, some backlash is necessary. I.S. specifies two types of backlash.

1. Circumferential backlash
2. Normal backlash.

The desired amount of backlash is difficult to evaluate. It is, therefore, recommended that when a designer user, or purchaser includes a reference to backlash in gearing specification and drawing, consultation be arranged with the manufacturer.

I.S., therefore, does not specify the tolerance etc. Backlash is determined as follows, one of the two gears of the pair is locked, while the other is rotated backwards and forward as far as possible, the maximum displacement recorded by a comparator whose stylus is locked near the reference cylinder and a tangent to this is called the circular backlash.

For spur gears, the normal backlash is equal to circular backlash multiplied by the cosine of pressure angle.

Tooth thickness measurement. Since the tooth thickness is defined as the length of an arc, it is not possible to measure it directly. It is generally measured at pitch circle and is, therefore, the pitch line thickness of the tooth. In most of the cases, it is sufficient to measure the chordial thickness i.e., the chord joining the intersection of the tooth profile with the pitch circle.

There are various methods of measuring the gear tooth thickness.

(*i*) Chordal thickness method (measurement of tooth thickness by gear tooth vernier calliper).
(*ii*) Constant chord method.
(*iii*) Base tangent method.
(*iv*) Measurement over pins or balls.

Chordal thickness method. In this method, gear tooth vernier calliper is used to measure the thickness of gear tooth at the pitch line. The gear tooth vernier calliper consists of two perpendicular vernier arms with vernier scale on each arm. One of the arms is used to measure the thickness of gear teeth and other for measuring depth. The calliper is so set that it slides on the top of tooth of gear under test and the lower ends of the calliper jaws touch the sides of the tooth at the pitch line. The reading on the horizontal vernier scale gives the value of chordal thickness (W) and the reading on the vertical vernier scale gives the value of chordal addendum. These measured values are then compared with the calculated values.

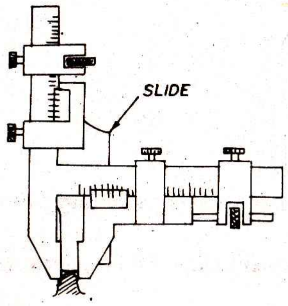

Fig. 10.8. Vernier gear tooth calliper

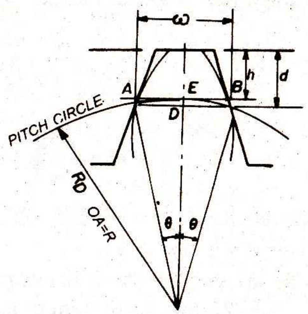

Fig. 10.9. Chordal thickness method

Considering one gear tooth, the theoretical values of w and d can be found out. In Fig. 10.9, W is a chord ADB, but tooth thickness is specified as an arc distance AEB. Also, the distance d adjusted on instrument is slightly greater than the addendum E, W is therefore called chordal thickness and is called chordal addendum.

Now, from Fig. 10.9

$$w = AB = 2\,AD$$

angle $$AOD = \theta = \frac{360}{4\,T}$$

where T = number of teeth,

$$w = 2\,AD = 2 \times A\,O \sin\theta$$

i.e., $$w = 2R \sin \frac{360}{4\,T} \quad (R = \text{Pitch circle radius})$$

Module $$m = \frac{\text{P.C.D.}}{\text{No. of teeth}} = \frac{2R}{T}$$

Therefore, $$R = \frac{Tm}{2}$$

and $$w = 2\,\frac{Tm}{2}\,.\,\sin\left(\frac{360}{4T}\right)$$

i.e., $$w = T.m \sin\left(\frac{90}{T}\right) \qquad \text{...(1)}$$

Also from Fig. 10.9

$$d = OC - OD$$

But $$OC = OE + \text{addendum} = R + m$$
$$= \frac{Tm}{2} + m$$

and $$OD = R\cos\theta$$
$$= \frac{Tm}{2}\cos\left(\frac{90}{T}\right)$$

Therefore, $$h = \frac{Tm}{2} + m - \frac{Tm}{2}\cos\left(\frac{90}{T}\right)$$

i.e., $$h = \frac{Tm}{2}\left(1 + \frac{2}{T} - \cos\left(\frac{90}{T}\right)\right) \quad \text{...(2)}$$

or $$h = m + \frac{Tm}{2}\left[1 - \cos\left(\frac{90}{T}\right)\right]$$

The vernier method described above is not very satisfactory because of the following reasons.

(*i*) the vernier itself is not reliable to closer than 0.05 mm or perhaps 0.025 mm with practice,

(*ii*) the measurements depend on two vernier readings, each of which is a function of the other,

(*iii*) measurement is made with an edge of the measuring jaw, not its face, which again does not lead itself to accurate measurement.

These problems can be overcome by measuring the span of a convenient number of teeth with a review calliper.

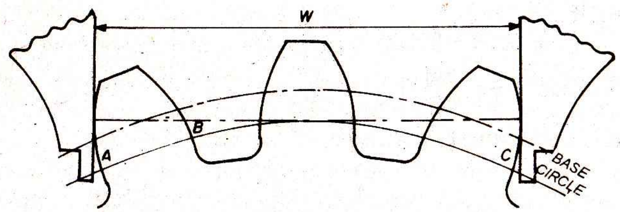

Fig. 10.10. Measurement over a number of teeth

Constant Chord Method. In gear tooth collinear method, both the chordal thickness and chordal addendum are dependent upon the number of teeth. Hence, for measuring a large number of gears for set, each having different number of teeth would involve separate calculations. Thus the procedure becomes laborious and time consuming.

The constant chord method does away with these difficulties. It enables to employ one setting for all the gears having the same pitch and pressure angle irrespective of the number of teeth.

The constant chord is defined as the chord joining those points, on opposite faces of the tooth, which make contact with the mating teeth when the centre line of the tooth lies on the line of the gear centres.

In Fig.10.10. *AB* is known as constant chord. The value of *AB* and its depth from the tip, where it occurs, can be calculated mathematically and then verified by instrument.

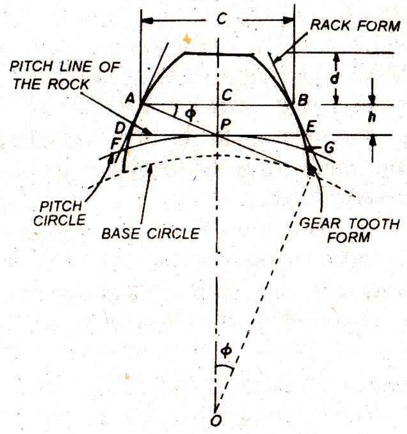

Fig. 10.11

The advantage of the constant chord method is that for all number of teeth (of same module) value of constant chord is same. Secondly, it readily lends itself to a form of comparator which is more sensitive than the gear tooth vernier.

Base Tangent Method. (David Brown tangent comparator). Measurement by this method uses either a micrometer with flanked anvils or the David Brown Tagent computer as shown in Fig. 10.12. This instrument essentially consists of a fixed anvil and a movable anvil. There is a micrometer on the moving anvil side and this has a very limited movement on either side of the setting. The distance is adjusted by setting the fixed anvil at a desired place with the help of locking ring and setting tubes.

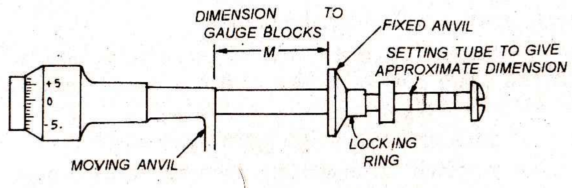

Fig. 10.12. David Brown Tangent Comparator

Measurement over Rollers. A convenient method of checking tooth thickness and obtaining some indication of accuracy of involute profile is to measure a gear over rollers placed in opposite tooth spaces. Two or three different sizes of rollers can be used so that viriations at several places on the tooth flanks can be detected.

Measurement of Concentricity

The centre about which the gear is mounted should be coincident with the centre from which the gear is generated. Otherwise, the gear wheel will not function correctly because of eccentricity of mounting.

The concentricity of the gear may be readily checked by mounting the gear between centres and measuring the variation in height of a roller placed between the successive teeth.

The variation in reading obtained will be a function of the eccentricity present and also of any variation which may be there in the tooth thickness.

By rotating the gear, tooth by tooth, dial gauge readings over the rollers can be noted and plotted in the form of a graph as shown in Fig. 10.13.

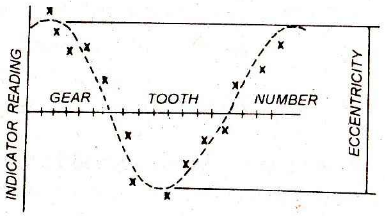

Fig. 10.13. Graphical representation of eccentricity

Alignment checking. The alignment of gear with respect to the axis of mounting may be checked by placing a parallel bar between the gear teeth. The gear being mounted between centres.

Readings are taken at the two ends of the bar. Differences in the readings (*i.e.*, height) of either end of the parallel bar will indicate presence of misalignment.

Gear Errors. Variation in manufacturing conditions may lead to many types of errors in gears. Some of the possible types of errors on spur, helical, bevel and worm gears are as described below.

Pitch errors. Pitch error is a source of gear noise and the character of noise will depend upon how pitch errors are produced and how they are distributed.

(*a*) Adjacent pitch error = Design pitch – actual pitch.

(*b*) Cumulative pitch error = design length value – actual value between corresponding flanks of teeth not adjacent to each other.

(*c*) *Profile error.* It is the maximum distance of any point on the tooth profile form and normal to the design profile when the two coincide at the reference circle.

Tooth to tooth composite error. Single flank

The range of difference between the displacement at the pitch circle of a year and that of a master gear meshed with it at fixed centre distance when moved through a distance corresponding to one pitch with only the driving and driven flanks in contact is called tooth to tooth composite error.

Total composite error. Single flank. It is the range of difference between the displacement at the pitch circle of a gear and that of a master gear meshed with it at fixed centre distance when moved through one revolution with only the driving and driven flanks in contact.

Tooth to tooth composite error. Double flank.

It is the range of variation in the minimum centre distance between a gear and a master gear when rotated through a distance corresponding to the pitch of the teeth.

Total Composite error. Double flank.

It is the range of variation in minimum centre distance between a gear and a master gear when the gear is rotated through one revolution.

Tooth alignment error. The distance of any point on a tooth trace from the design tooth trace passing through a selected reference point on that tooth is called tooth alignment error.

Tooth thickness error. It is the value obtained by subtracting the design tooth thickness from the actual tooth thickness measured along the surface of the reference cylinder.

Cyclic error. It is the error occurring during each revolution of the element under considerations.

Periodic error. An error occurring at regular intervals not necessarily corresponding to one revolution of the component.

Runout. It is the total range of reading of a fixed indicator with the contact point applied to a surface rotated, without axial movement, about a fixed axis.

Radial runout. It is the runout measured along a perpendicular to the axis of rotation.

Axial runout. It is the runout measured parallel to the axis of rotation, at a specified distance from the axis.

Eccentricity. It is half the radial runout.

The presence of these errors causes interference in efficient operation of gears. These result in non-smooth and noisy operation which ultimately effect the working life.

Checking of composite errors. Composite errors can be checked by measuring the variations of centre distance when the gear under test is rolled under spring pressure against a master gear. The test is generally known as 'rolling gear test' or functional test. Its principle is illustrated in Fig. 10.14. Total composite variation is centre distance variation in one complete revolution of the gear being inspected; whereas tooth to tooth variation is the centre distance variation as the gear is rotated through an increment of $\frac{360}{N}$.

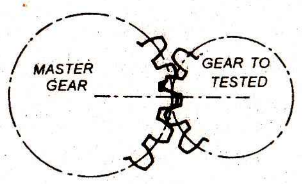

Fig. 10.14

A uniform tooth to tooth variation shows profile variation while a sudden jump indicates pitch variation.

Composite type of checking takes care of the errors in the gears. It is specially very much suited for large gears as it also ensures control over the tooth spacing. The composite method of checking is very much suitable for checking worn gears.

Master Gears. These gears are made with sufficient accuracy and can be used as the basis for comparing the accuracy of other gears. Master gears are mostly used in composite errors determination in which the master gears are rotated in close mesh (double flank) or in single contact with the gears under test. These can also be used for calibration of gear checking instruments used in shop floor.

Master gears are made from chromium-manganese tool steel or good quality gauge steel and are hardened to 62 HRC. These are properly stabilized to relieve internal stresses.

Parkinson Gear Tester

Working Principle. A standard gear (master gear) is mounted on a fixed vertical spindle and the gear to be tested on another similar spindle mounted on a sliding carriage. These gears are maintained in mesh by spring pressure. As the gears are rotated the movements of the sliding carriage are indicated by a dial indicator, and these variations are a measure of any irregularities in the gear under test; alternatively a recorder can be fitted, in the form of a waxed circular chart and records made of the gear variation in accuracy of mesh.

Fig. 10.15 shows a gear tester for testing spur gears. (Testers are available for bevel, helical and worm gases also). The gears are mounted on two spindles so that they are free to rotate without measurable clearance. The master gear is mounted as an adjustable carriage whose position can be adjusted to enable a wide range of gear diameters to be accommodated and it can be clamped in any desired position. The gear under test is mounted on a floating spring loaded carriage so that the master gear and the gear under test may be meshed together under controlled spring pressures.

The two spindles can be adjusted so that their axial distance is equal to the designed gear centre distance. A scale is attached to one carriage and vernier to the other, this enables centre distance to be measured to within 0.025 mm.

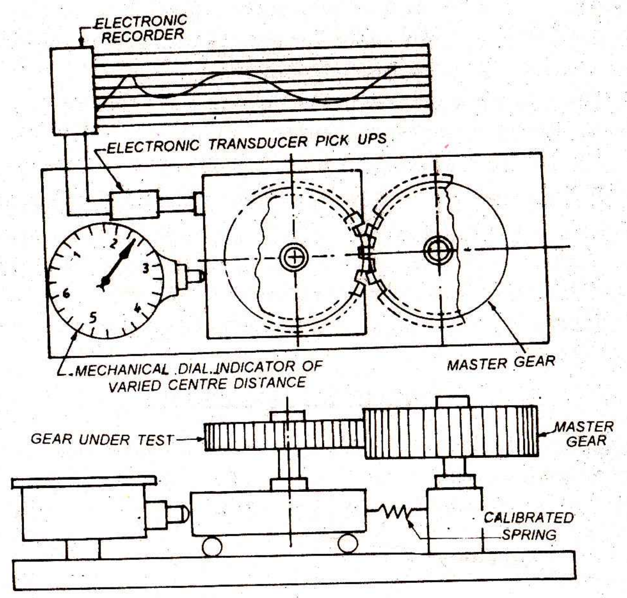

Fig. 10.15. Parkinson Gear Tester

When these gears are in close mesh and rotated any errors in the tooth form, pitch or concentricity of pitch line, will cause a variation in centre distance. Thus, movements of the carriage, as indicated to the dial gauge, show the errors in the gear under test.

Fig. 10.16

Alternatively a recorder can be fitted as shown in Fig. 10.14 in the form of a waved circular or rectangular chart and records made of the irregularities in the gear under test. Fig. 10.16 shows a reproduction of a few typical charts with a reduced scale and the magnified radial errors. Gear 1 is an unsatisfactory, gear 2 is moderate gear and gear 3 is fully satisfactory.

Some limitations of Parkinson gear tester are :

1. Generally 300 mm diameter gear is maximum, usually 150 mm or smaller diameter gears are also tested.

2. There is a low friction in the movement of the floating carriage and a high sensitivity of the sensing unit is important.

3. The accuracy is of the order of ± 0.001 mm.

4. Rolling test does not reveal all errors, since the device is sensitive to cumulative position errors.

5. Errors are not clearly identified for type profile, pitch, helix and tooth thickness and are indistinguishably mixed.

6. Measurements are directly dependent upon the master gear or reference gear.

SOLVED PROBLEMS

Problem 1. *Calculate the setting for a straight spur gear having 40 teeth of module 3 pitches.*

Sol. Setting of the vernier for a gear of 40 teeth of module 3 pitches

Now, $Tm \sin \frac{90}{T}$

where, w = width (thickness of teeth)

m = module

T = number of teeth

Therefore, $w = 40 \times 3 \sin \frac{90}{40}$

$= 120 \sin 2°15'$

depth of tooth, $h = m + \frac{Tm}{2}\left[1 - \cos\left(\frac{90}{T}\right)\right]$

i.e., $h = 3 + \frac{40 \times 3}{2}\left[\frac{1 - \cos 90°}{40}\right]$

$= 3 + 60(1 - 0.9993) = 3.042$ mm.

Problem 2. (*a*) *Give the elements for specifying the gear*

(*b*) *Give formulae for* (*i*) *circular pitch* (*ii*) *module* (*iii*) *dimetral pitch* (*iv*) *Base pitch* (*v*) *Addendum* (*vi*) *base diameter.*

Sol. (*a*) A gear can be specified by indicating the following elements

(*i*) Number of teeth (T)

(*ii*) Diametral pitch (Pd) or module (m)

(*iii*) Pressure angle ϕ

(*b*)

(*i*) Circular pitch, $P_c = \pi m$

(*ii*) Module $m = \frac{dp}{T}$ where dp = pitch circle diameter

(*iii*) Diametral pitch $p_d = \frac{T}{dp}$

(*iv*) Base pitch $P_b = P_c \cos\ \phi$ where ϕ = pressure angle.

(*v*) Addendum $= \frac{1}{Pd} = \frac{dp}{T}$

(*vi*) Base diameter $d_b = mT \cos\ \phi$

Problem 3. *A spur gear of 3 mm module has 40 teeth. Calculate the following proportions. Pitch Circle diameter, addendum, dedendum working height and base pitch for a pressure angle 20°.*

Sol. (*i*) Pitch circle diameter, $dp = m \times t$

$= 3 \times 40 = 120$ mm

(*ii*) Addendum $= m = 3$ mm

(*iii*) Dedendum = Addendum + Clearance

$= m + 0.25$ m

$= 1.25$ m

$= 1.25 \times 3 = 3.75$ mm.

(*iv*) Tooth working height = Addendum + Dedendum

$= 3 + 3.75 = 6.75$ mm

(*v*) Base pitch, $= m\pi \cos\ \phi$

$= 3\ \pi \cos 20°$

$= 3\pi \times 0.9397$

$= 8.87$ mm.

Problem 4. *Calculate the chord length and its distance below the tooth tip for a gear of module 4 and 20° pressure angle.*

Sol. Chord length $= m\frac{\pi}{2}\cos^2 \phi$

$= 4 \times \frac{\pi}{2} \times \cos^2 20°$

$= 4 \times \frac{\pi}{2} \times (0.9397)^2$

$\simeq 5.55$ mm.

Distance below the tip of gear

$h = m - \frac{m\pi}{4} \sin\ \phi\ \cos\ \phi$

$= m - \frac{m\pi}{8} \times \sin 2\ \phi$

$= 4 - 4 \times \frac{\pi}{8} \times (0.3214)$

$\simeq$ mm'

Problem 5. *State the various sources of errors in manufacturing gears.*