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Parallel involute gears — ISO system of accuracy

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FOREWORD

ISO (the International Organization for Standardization) is a worldwide federation of national standards institutes (ISO Member Bodies). The work of developing International Standards is carried out through ISO Technical Committees. Every Member Body interested in a subject for which a Technical Committee has been set up has the right to be represented on that Committee. International organizations, governmental and non-governmental, in liaison with ISO, also take part in the work.

Draft International Standards adopted by the Technical Committees are circulated to the Member Bodies for approval before their acceptance as International Standards by the ISO Council.

International Standard ISO 1328 was drawn up by Technical Committee ISO/TC 60, *Gears*, and circulated to the Member Bodies in December 1967.

It has been approved by the Member Bodies of the following countries :

Australia	India	Sweden
Austria	Israel	Switzerland
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Chile	Korea, Rep. of	United Kingdom
Czechoslovakia	Netherlands	U.S.S.R.
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France	Poland	
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The Member Bodies of the following countries expressed disapproval of the document on technical grounds :

Brazil
Hungary
Japan

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Parallel involute gears — ISO system of accuracy

1 SCOPE AND FIELD OF APPLICATION

This International Standard establishes a system of accuracy for parallel involute gear pairs defined in ISO 53, *Cylindrical gears for general and heavy engineering — Basic rack*, and ISO/R 54, *Modules and diametral pitches of cylindrical gears for general engineering and for heavy engineering.*

It specifies all errors the control of which is provided for, whether they be on a single wheel or on the complete gear pair, and gives the corresponding tolerances.

NOTE — Certain types of gear pairs may require only a limited number of controls; these will be dealt with in special standards covering these types of gears.

2 DEFINITIONS

The logical order of manufacture of a gear pair is :

– machining tne blanks of the two gears;
– cutting the teeth of the two gears;
– assembling the two toothed wheels under operating conditions.

It is therefore normal to carry out the successive inspections in a corresponding order :

– inspection of the blanks of the two gears;
– inspection of the teeth of the two gears;
– inspection of the assembly conditions of the gear pair.

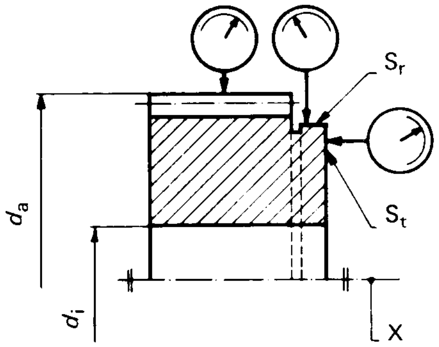
2.1 INSPECTION OF THE BODY OF THE WHEELS (OR BLANK)

2.1.1 *Reference axis*

2.1.1.1 In the case of pinions or wheels with bores, the axis of the bore shall be adopted as the reference axis.

2.1.1.2 In the case of pinions on shafts, the reference axis shall be the bearing axis of the bearings.

2.1.1.3 In order to facilitate the operations of machining, inspection and assembly of toothed wheels, it is recommended that radial and lateral auxiliary reference surfaces be indicated clearly on the working drawings (see figure 1).



d_a : tip diameter
d_i : bore diameter

X : reference axis
S_r : radial reference surface
S_t : axial reference surface

FIGURE 1

2.1.2 *Tip cylinder*

2.1.2.1 The value of the **tip diameter** is not of essential importance. It is as well, however, to give preference to a value tending to increase the bottom clearance. In cases where the apparatus for inspecting the tooth thickness rests on the tip cylinder, allowance must be made for the tip diameter error.

2.1.2.2 The **radial run-out** is the total amplitude of the deviation of the needle of a comparator the stylus of which is in contact with the tip cylinder during a complete revolution of the gear (figure 1). This check is important only in the case where certain tooth inspection instruments rest on the tip cylinder.

2.1.3 *Reference surfaces* (figure 1)

2.1.3.1 The **radial run-out** is the total amplitude of the deviation of the needle of a comparator the stylus of which is in contact with the radial cylindrical reference surface during a complete revolution of the gear.

2.1.3.2 The **axial run-out (wobble)** is the total amplitude of the deviation of the needle of a comparator the stylus of which is in contact with the axial reference surface during a complete revolution of the gear.

2.2 INSPECTION OF THE TEETH

2.2.1 *Division*

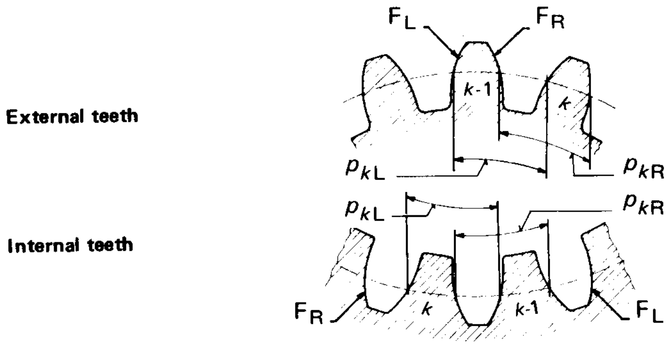
Looking at the wheel's axial reference surface, number the teeth in a clockwise direction

1, 2, 3, . . . etc. . . . to z.

Then adopt the terminology below (figures 2 and 3), which is valid for the control of external and internal teeth.

a) **right flank :** flank bounding a tooth to the right when this tooth is seen with its tip above its root.

b) **left flank :** flank bounding a tooth to the left, in the above circumstances.



p_{kL} : left pitch k
p_{kR} : right pitch k

F_L : left flank
F_R : right flank

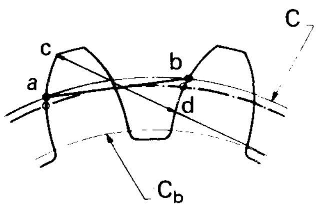
FIGURE 2

c) **pitch** k **:** pitch between one profile of tooth k-1 and the similar profile of tooth k.

d) **right pitch :** pitch between two consecutive right flanks.

e) **left pitch :** pitch between two consecutive left flanks.

f) **circular pitch :** term designating the value of the pitch round the checking circle which has the same centre as the reference circle and is generally adjacent to it (figure 3).



C : checking circle

C_b : base circle

$\overset{\frown}{ab}$: circular pitch

ab : chord of the circular pitch

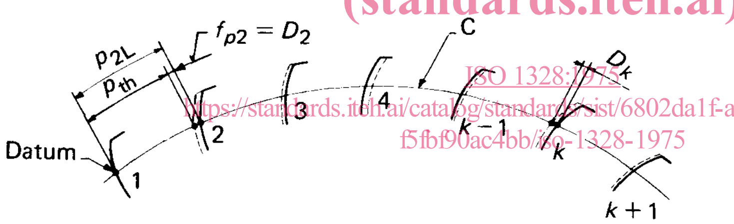
cd : base pitch

FIGURE 3

g) **chord of the circular pitch :** chord corresponding to the circular pitch (figure 3).

h) **base pitch :** distance between two consecutive similar flanks measured along a tangent to the base circle (figure 3); in the case of involute teeth without profile errors, the base pitch is equal to the pitch around the base circle.

2.2.1.1 The **circular pitch individual error** is the (algebraic) difference between the actual circular pitch and the theoretical circular pitch. The theoretical circular pitch is, furthermore, the mean value of all the actual circular pitches (figure 4).



p_{th} : theoretical pitch

p_{2L} : actual left pitch (2)

D_k : displacement k (positive)

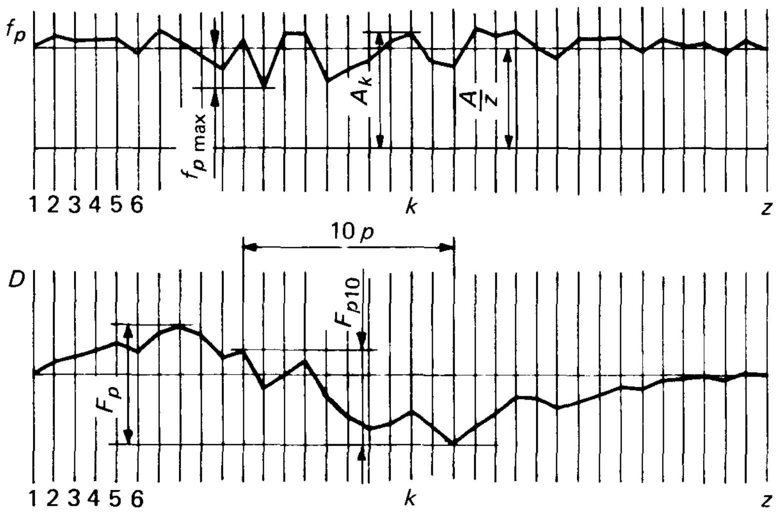
f_{p2} : individual error (2)

D_2 : displacement (2)

C : checking circle

FIGURE 4

Figure 5 gives an example of a curve of circular pitch individual errors, with an indication of the maximum individual error. If we call A the algebraic sum of readings from the checking apparatus for z successive pitches (z = number of teeth on wheel), the abscissae of the curves of individual errors (corresponding to the theoretical pitch) will be defined by the reading A/z.



f_p : individual adjacent transverse pitch deviation

D : displacement

$f_{p\ max}$: maximum individual transverse pitch deviation

z : number of gear teeth

A : algebraic sum of readings from checking apparatus

p : pitch

A_k : reading k on the checking apparatus

F_p : total cumulative error

F_{p10} : cumulative error over 10 pitches

FIGURE 5

2.2.1.2 The **base pitch error** is the (algebraic) difference between the actual base pitch and the theoretical base pitch.

2.2.1.3 Cumulative error over a certain sector : Refer in figure 4 to the division of the family of "left" flanks.

The datum profile is that of tooth 1. The perfect wheel is shown by a broken line.

2.2.1.3.1 The **circular displacement** of any profile, number k (same number as that of the tooth), is the positioning error between the actual profile and the theoretical profile measured on the control circle. It is positive or negative, depending on whether the actual profile is ahead of or behind its theoretical position.

Figure 5 gives the curve of circular displacements corresponding to that of circular pitch individual errors.

The circular displacement of any profile k is the algebraic sum of the circular pitch individual errors from the datum profile. Conversely, the individual error of any circular pitch k is the algebraic difference between the displacement of the profile k and that of the profile $k - 1$.

2.2.1.3.2 The **cumulative error over a sector of k pitches** is the difference between the actual length of the arc of the control circle between two similar profiles and the theoretical length of this arc. It is also the algebraic sum of the individual errors of k circular pitches. It may be determined directly from the curve of circular displacements for any sector : for instance on figure 5 the cumulative error over a given sector of 10 pitches has been shown.

The **total cumulative pitch error** is the total amplitude of the displacement curve. It is the maximum cumulative error over any sector of one half circumference ($k = z/2$).

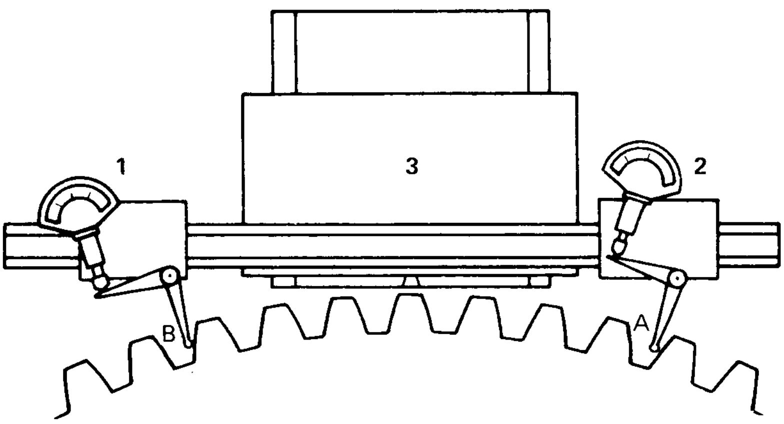
2.2.1.3.3 Inspection of large wheels — Inspection by sectors (Span measurement). In the case of wheels with a large number of teeth, it is not desirable to determine the displacement chart by summation of the individual errors : each of the assessments of individual errors may indeed be affected by a small error due to the inspection apparatus and its operator.

To determine cumulative pitch errors, the use of "span measurement" is therefore recommended. By means of a suitable apparatus the division error is not determined on each circular pitch, but on successive sectors containing a certain number of pitches (figure 6) : this number must be sufficiently large, and if possible should be a submultiple of the number of teeth in the wheel being inspected. The two styli A and B of the instrument shall be in contact with similar flanks and at each reading, one of the styli shall occupy the position of the other at the previous reading.

The instrument must be properly placed in relation to the wheel, mounted on a fixed support outside the controlled wheel which can be rotated to come into the different control positions.

The cumulative error on each of the sectors is the algebraic difference between the reading taken for this sector and the mean value of all the readings.

NOTE — Cumulative errors should not be determined from base pitch individual errors.



1 : checking
2 : positioning
3 : fixed support

FIGURE 6

2.2.1.4 Case of helical teeth. As far as possible, inspection of circular pitch individual errors should be carried out in the same plane perpendicular to the gear axis.

If inspection is carried out in a direction perpendicular to the direction of the teeth, the values of the tolerances given in 4.4.1.3 should be multiplied by the cosine of the helix angle.

2.2.1.5 Case of double helical teeth. In the case of double helical teeth, it is necessary to avoid having too great a difference between the cumulative errors on the two wings of the helix for arcs of the same length occupying the same angular position on the wheel, as this might result in bad load distribution on the two wings of the helix, with the risk of axial displacements and vibration.

2.2.2 *Eccentricity – Radial run-out*

2.2.2.1 The error of concentricity, or **eccentricity**, of a wheel is the deviation between the geometrical axis of the teeth and the reference axis (i.e. the hub).

It is not possible to determine this error in isolation, but its influence is nevertheless recorded when checking errors affecting the regularity of the drive (division, profile, etc.) : for example, a certain eccentricity can introduce a circular displacement curve of a sinusoidal nature of a total amplitude equal to twice this eccentricity. It is therefore generally agreed that the determination of eccentricity be replaced by a practical inspection conventionally termed **radial run-out inspection.**

2.2.2.2 The practical determination of **radial run-out** is carried out in the following way : the total amplitude of the variation of penetration of a measuring device (ball or roller introduced into consecutive tooth spaces, or rider placed on consecutive teeth (figure 7), is measured for one complete rotation of the wheel being checked. The actual radial run-out would be equal to twice the eccentricity if there were no tooth error.

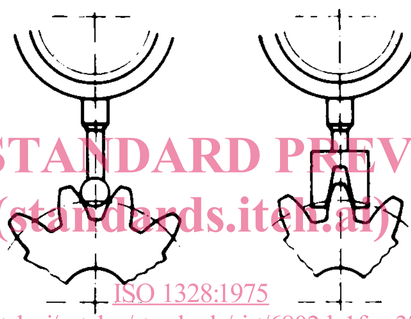


FIGURE 7

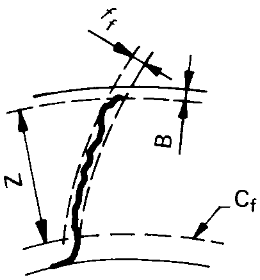
In practice, radial run-out is influenced by tooth errors on both series of flanks, and also possibly by the method of manufacture. The tolerances indicated in 4.4.2 are valid for commercial gear pairs for transmitting movement, whatever the method of manufacture. For special gear pairs (radar, master gears, etc.) lower values are sometimes necessary.

The dimensions of the balls, rollers or riders are chosen so that their contact points with the teeth are located approximately at half height of the teeth.

2.2.3 *Total profile error*

2.2.3.1 The **total profile error** is the distance, measured along their common normal, between two reference profiles which enclose the actual profile (figure 8).

The total profile error is the resultant of the **base diameter error** and the **shape error** of the profile. The zone of inspection will be limited towards the root of the tooth by the working root circle, i.e. to the zone of effective contact with the mating profile of the other gear of the gear pair. If this wheel is not known with certainty, it will suffice if a rack is assumed. The zone of inspection will be limited towards the tooth tip by the beginning of the chamfer.



f_f : total profile error

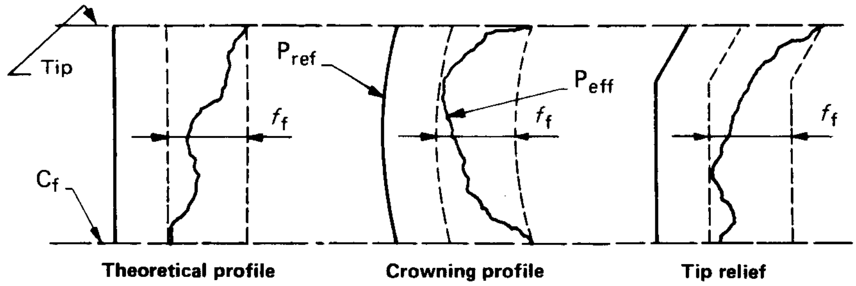
Z : zone of inspection

B : chamfer

C_f : working root circle

FIGURE 8

2.2.3.2 Obviously an intentional alteration to the profile, such as crowning or, more simply, tip or root relief, should not be regarded as a profile error : the reference profile will not necessarily be the design involute; figure 9 gives examples of charts obtained on conventional profile-checking instruments.



f_f : total profile error

C_f : working root circle

P_{ref} : reference profile

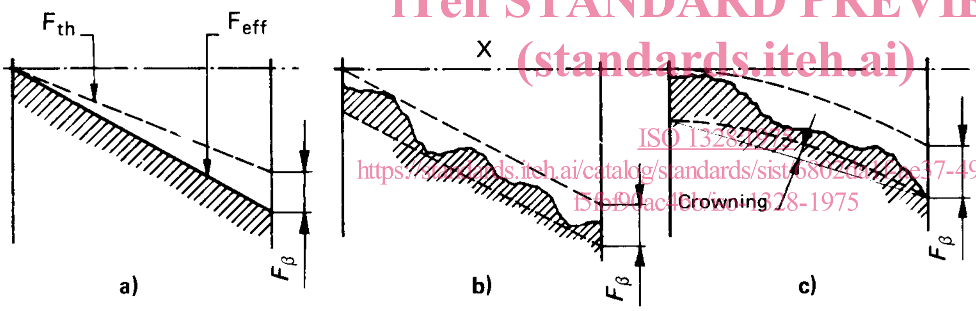
P_{eff} : actual profile

FIGURE 9

2.2.4 *Total alignment error (or distortion)*

2.2.4.1 The **total error of alignment** or **distortion** is the resultant of the **deviation** of the tooth trace on a cylinder coaxial to the pitch cylinder, and of the **longitudinal shape error.**

Misalignment is determined by enclosing the actual trace of a flank between two reference traces. It is determined over the total effective width of the teeth and in a plane perpendicular to the axis (figure 10); the tolerances given in 4.4.4 are relative to this method of determination; if another method of determination is necessitated by the checking equipment used, it will suffice to make the relevant adjustment of tolerances.



F_β : distortion

X : axis

F_{th} : theoretical flank trace

F_{eff} : actual flank trace

FIGURE 10

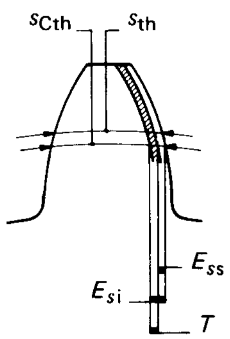
2.2.4.2 Figure 10 a) is relative to teeth with a theoretical reference trace, and with only one deviation error.

Figure 10 b) is relative to teeth with a theoretical reference trace, and with deviation and a longitudinal shape error.

Figure 10 c) is the general case, where the reference trace has an intentional longitudinal correction.

2.2.5 *Thickness tolerances*

2.2.5.1 Taking the theoretical thickness as the nominal thickness, then the **thickness tolerance** is defined by the **upper deviation** and the **lower deviation** (figure 11). For helical teeth, the values relate to the theoretical "normal" thickness.



s_{Cth} : theoretical tooth thickness (on the checking circle)

s_{th} : theoretical tooth thickness (on the reference circle)

$$= \frac{\pi m}{2} \pm 2\,xm\ \mathrm{tg}\alpha$$

with m = module
x = addendum modification coefficient
α = pressure angle

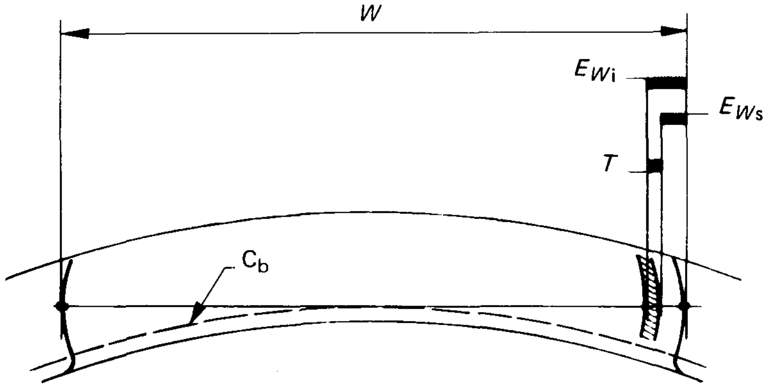
E_{ss} : upper deviation

E_{si} : lower deviation

T : tolerance

FIGURE 11

2.2.5.2 As thickness is often checked by measuring the "base tangent length" over a number of teeth (figure 12), it suffices to define the **distance tolerance** by its **upper deviation** and its **lower deviation**.



W	:	theoretical distance
E_{Ws}	:	upper deviation
E_{Wi}	:	lower deviation
T	:	tolerance
C_b	:	base circle

FIGURE 12

2.2.5.3 Deviations are not necessarily a result of the quality of the teeth : considerable deviations may be necessary for certain types of precision teeth designed to operate with considerable backlash between flanks.

2.2.5.4 On the other hand, in a given gear pair, the deviations depend more directly on the minimum normal backlash necessary to ensure correct functioning of the mating parts.[1)]

2.2.6 *Radial composite error* (double flank composite error)

2.2.6.1 A quick and practical overall check for teeth consists in engaging the gear being inspected with a master gear (made with sufficient accuracy to allow its errors in relation to those of the gear being inspected to be ignored) (figure 13). The error recorded results from a combination of all the individual errors of the teeth.

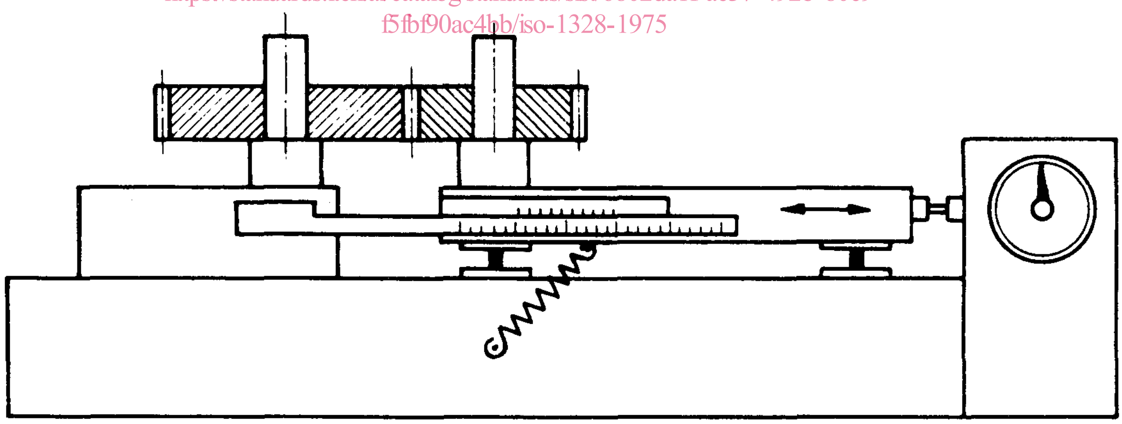


FIGURE 13

For checking the radial composite error, the gear being inspected and the master gear are arranged on an apparatus so designed that one of its arbors can move and is attached to a spring so that there can be a constant radial load in the matching of the two gears with each other. Variations in the centre distance are generally recorded on a chart as Cartesian co-ordinates (figure 14 a)) or polar co-ordinates (figure 14 b)).

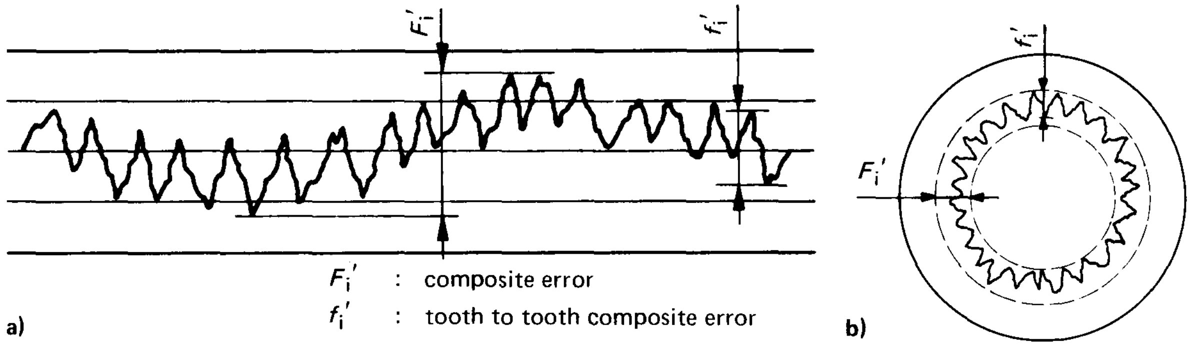
2.2.6.2 The **radial composite error** is the total amplitude of the chart.

2.2.6.3 Local errors, such as pitch, profile and alignment errors, produce a succession of small undulations along the chart record, generally one pitch apart. The **radial tooth-to-tooth composite error** is the maximum amplitude of these undulations (figures 14 a) and 14 b)).

1) Since studies must be undertaken for the determination of the deviation as a function of the permissible normal backlash, the data given in 4.4.5.2 and in table 8, which are at present valid — unless otherwise stated — for individual gears, may be modified or completed later.

2.2.7 *Tangential composite error* (single flank composite error)

2.2.7.1 The gear under inspection meshes with a master gear of adequate accuracy, at the designed operating centre distance, contact being made on one series of similar flanks. Owing to the existence of errors on the teeth, irregularity occurs in the positioning of the gear under inspection in relation to the theoretical position. Certain instruments make it possible to record this error in relation to the pitch circle of the gear under inspection. Chart recordings can be made in the form of Cartesian co-ordinates or polar co-ordinates (see figures 14 a) and 14 b)).



F_i' : composite error

f_i' : tooth to tooth composite error

FIGURE 14

2.2.7.2 The **tangential composite error** is the total amplitude of the chart.

2.2.7.3 The **tangential tooth-to-tooth composite error** is the maximum amplitude of the small undulations, often distant by 1 pitch, occurring along the chart record.

2.2.7.4 There is only a single value for the radial composite error for a given gear; on the other hand there is a tangential composite error for each series of flanks on the gear under inspection. For preference, the tangential composite error should be determined for the operating direction of the gear pair.

2.3 INSPECTION OF THE GEAR PAIR

2.3.1 *Centre distance error – Centre distance tolerance*

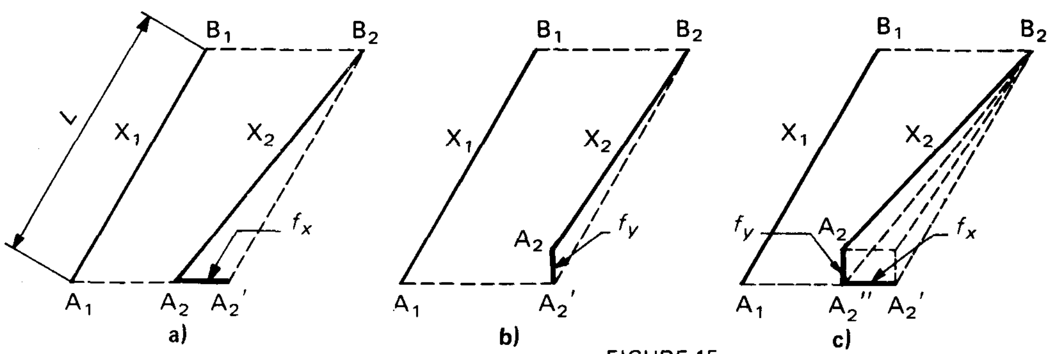
The **centre distance error** is the difference between the centre distance actually produced and the design operating centre distance, the centre distance being measured in a plane normal to the direction of the axes which cross the middle of the width of the gear teeth.

The **centre distance tolerance** lies symmetrically over the zero line corresponding to the design centre distance.

Certain particular applications require matching of teeth and sometimes a device for adjusting the centre distance. Sometimes they may require a unilateral centre distance tolerance.

2.3.2 *Parallelism of the axes* (for the gear shaft)

Let X_1 and X_2 be the two axes and L the distance, as great as possible, along which the parallelism will be checked between the extreme points A_1 and B_1, A_2 and B_2 which can be obtained on these axes (figure 15). Consider the plane passing through the axis X_1 and the end B_2 of the axis X_2 :



f_x : inclination

f_y : deviation

FIGURE 15

2.3.2.1 Figure 15 a) makes it possible to define an **inclination error** A_2A_2' related to a given length.

A_2A_2' lies in the plane $A_1B_1B_2$.

2.3.2.2 Figure 15 b) makes it possible to define a **deviation error** A_2A_2' related to a given length.

A_2A_2' is normal to the plane $A_1B_1B_2$.

2.3.2.3 Figure 15 c) expresses the general case where the out-of-parallelism of the two axes X_1 and X_2 is the result of the combination of an inclination error $A_2'A_2''$ and a deviation error A_2A_2''.

2.3.2.4 The inclination and deviation errors determined over the distance L will be related to the facewidth of the gear pair.

It should be noted that the deviation error is expressed as an element of substantially the same value, whereas the influence of the inclination error is less felt.

2.3.3 *Backlash*

2.3.3.1 The **circular backlash** is determined as follows. One of the two gears of the pair is locked, while the other, mounted at the prescribed centre distance, is rotated backwards and forwards as far as possible. The maximum displacement is recorded by, for example, a comparator the stylus of which is located near the reference cylinder and at a tangent to this cylinder (figure 16).

2.3.3.2 The **normal backlash** is the total clearance which can be determined for example with a feeler gauge inserted between the teeth on the line of contact (figure 16).

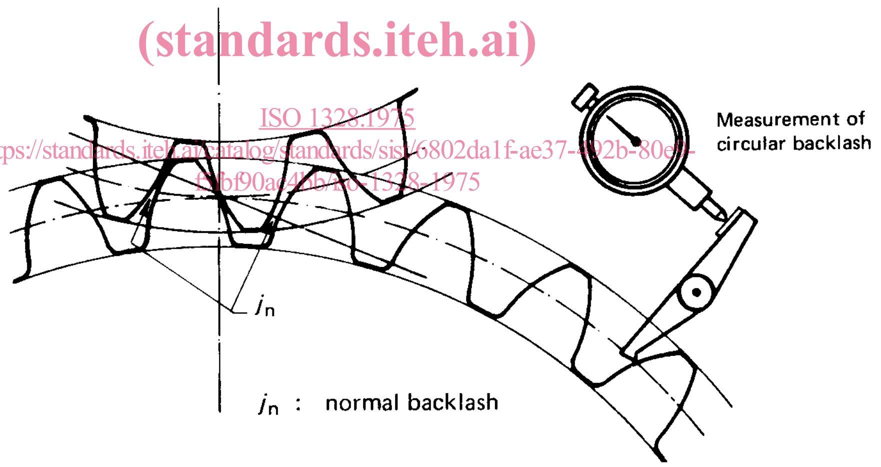


FIGURE 16

2.3.3.3 In the case of spur teeth, the normal backlash is equal to the circular backlash multiplied by the cosine of the pressure angle.

In the case of helical teeth, the normal backlash is approximately equal to the circular backlash multiplied by the cosine of the pitch inclination angle.

2.3.3.4 The **backlash tolerance** will be defined by the **upper deviation** and the **lower deviation.**

2.3.4 *Composite error of the gear pair* (double flank and single flank composite errors)

The radial composite error and the tangential composite error may be determined according to 2.2.6 and 2.2.7, or by matching the two gears used in the pair.

In the general case, it is accepted that the composite error of the gear pair may reach a value equal to the sum of the composite errors of each of its two gears.

In the case where the number of teeth in a wheel is a multiple of the number in a pinion, certain arrangements of these two parts in relation to one another may allow the composite error of the complete gear pair to be reduced.