TECHNICAL REPORT ISO/TR 10064-1:1992
TECHNICAL CORRIGENDUM 1

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Cylindrical gears — Code of inspection practice —

Part 1: **Inspection of corresponding flanks of gear teeth**

TECHNICAL CORRIGENDUM 1

Engrenages cylindriques — Code pratique de réception — Partie 1: Contrôle relatif aux flancs homologues de la denture

RECTIFICATIF TECHNIQUE 1

Technical Corrigendum 1 to Technical Report ISO/TR 10064-1:1992 was prepared by Technical Committee ISO/TC 60, *Gears*.

Title page, Foreword, page 1

Delete "Cylindrical gears" from the title of ISO/TR 10064-1, thereby changing it to the following:

> *Code of inspection practice — Part 1: Inspection of corresponding flanks of gear teeth*

This is to accommodate the coverage of gears other than cylindrical in new parts of ISO/TR 10064.

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TECHNICAL REPORT

ISO TR 10064-1

First edition
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Cylindrical gears — Code of inspection practice —

Part 1:
Inspection of corresponding flanks of gear teeth

Engrenages cylindriques — Code pratique de réception —

Partie 1: Contrôle relatif aux flancs homologues de la denture

Reference number
ISO/TR 10064-1:1992(E)

Foreword

ISO (the International Organization for Standardization) is a worldwide federation of national standards bodies (ISO member bodies). The work of preparing International Standards is normally carried out through ISO technical committees. Each member body interested in a subject for which a technical committee has been established 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. ISO collaborates closely with the International Electrotechnical Commission (IEC) on all matters of electrotechnical standardization.

The main task of technical committees is to prepare International Standards, but in exceptional circumstances a technical committee may propose the publication of a Technical Report of one of the following types:

— type 1, when the required support cannot be obtained for the publication of an International Standard, despite repeated efforts;

— type 2, when the subject is still under technical development or where for any other reason there is the future but not immediate possibility of an agreement on an International Standard;

— type 3, when a technical committee has collected data of a different kind from that which is normally published as an International Standard ("state of the art", for example).

Technical Reports of types 1 and 2 are subject to review within three years of publication, to decide whether they can be transformed into International Standards. Technical Reports of type 3 do not necessarily have to be reviewed until the data they provide are considered to be no longer valid or useful.

ISO/TR 10064-1, which is a Technical Report of type 3, was prepared by Technical Committee ISO/TC 60, *Gears*.

This Technical Report updates description of and advice on gear inspection methods.

ISO 10064 consists of the following parts, under the general title *Cylindrical gears — Code of inspection practice*:

— *Part 1: Inspection of corresponding flanks of gear teeth* [Technical Report]

— *Part 2: Inspection of radial composite deviations, runout and tooth thickness allowance*

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INTRODUCTION

Together with definitions and values allowed for gear element deviations, the international standard ISO 1328-1975 also provided advice on appropriate inspection methods.

In the course of revising ISO 1328-1975, it was agreed that the description and advice on gear inspection methods should be brought up to date. Because of necessary enlargement and other considerations, it was decided that the relevant section should be published under separate cover as a Technical Report, Type 3, and that, together with this Technical Report, a system of documents as listed in clause 2 (References) should be established.

Cylindrical gears — Code of inspection practice —

Part 1:
Inspection of corresponding flanks of gear teeth

1. SCOPE

This part of the Technical Report constitutes a code of practice dealing with the inspection of corresponding flanks of cylindrical involute gears, i.e. with the measurement of pitch deviations, profile deviations, helix deviations and tangential composite deviations.

In providing advice on gear checking methods and the analysis of measurement results, it supplements the standard ISO 1328, part 1.

Most of the terms used are defined in ISO 1328 part 1, others are defined as they appear in the text and in clause 3.

2. REFERENCES

ISO 53:1954, Cylindrical gears for general and heavy engineering - Basic rack.

ISO 54:1977, Cylindrical gears for general engineering and heavy engineering - Modules and diametral pitches.

ISO 701:1976, International gear notation - Symbols for geometrical data.

ISO 1122-1:1983, Glossary of gear terms - Part 1: Geometrical definitions.

ISO 1328-1: -[1], Cylindrical gears - ISO system of accuracy - Part 1: Definitions and allowable values of deviations relevant to corresponding flanks of gear teeth.

ISO 1328-2: -[1], Cylindrical gears - ISO system of accuracy - Part 2: Definitions and allowable values of deviations relevant to radial composite allowance and backlash.

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1) To be published.

ISO 10063: - [1], Cylindrical gears - Flanks, undulation, surface roughness, shaft centre distance and parallelism of axes - Numerical values.

ISO/TR 10064-2: - [1], Cylindrical gears - Code of inspection practice - Part 2: Inspection of radial composite deviations, runout and tooth thickness allowance.

ISO/TR 10064-3: - [1], Cylindrical gears - Code of inspection practice - Part 3: Function groups, test groups and tolerance families.

3. SYMBOLS AND CORRESPONDING TERMS

3.1 Gear data

b	Facewidth
d	Reference diameter
d_b	Base diameter
m, m_n	Normal module
m_t	Transverse module
p_n	Normal pitch
p_t	Transverse pitch
p_b, p_{bn}	Normal base pitch
p_{bt}	Transverse base pitch
S	Number of pitches per sector
z	Number of teeth
α, α_n	Normal pressure angle
α_t	Transverse pressure angle
β	Helix angle
β_b	Base helix angle
ε_α	Transverse contact ratio
ε_β	Overlap ratio
ε_γ	Total contact ratio

1) To be published.

3.2 Gear deviations

Symbols used for deviations of individual element measurements from specified values are composed of lower case letters "f" with subscripts whereas symbols used for "cumulative" or "total" deviations, which represent combinations of several individual element deviations, are composed of capital letters "F" also with subscripts. It is necessary to qualify some deviations with an algebraic sign. A deviation is positive when e.g. a dimension is larger than optimum and negative when smaller than optimum.

f_{db} 1)	Base diameter difference
f_{dbm} 1)	Mean base diameter difference
f_e (f_{eL}, f_{eR})	Eccentricity between gear axis and axis of gear teeth (or of corresponding flanks, respectively)
$f_{f\alpha}$	Profile form deviation
$f_{f\beta}$	Helix form deviation
$f_{H\alpha}$ 1)	Profile slope deviation
$f_{H\alpha m}$ 1)	Mean profile slope deviation
$f_{H\beta}$ 1)	Helix slope deviation
$f_{H\beta m}$ 1)	Mean helix slope deviation
f_i'	Tooth-to-tooth tangential composite deviation (with master gear)
f_l'	Long period component of tangential composite deviation
f_s'	Short period component of tangential composite deviation

1) These deviations can be + (plus) or - (minus)

f'		Tooth-to-tooth transmission deviation (product gear pair)
f_{pb}	1)	Base pitch deviation
f_{pbm}	1)	Mean base pitch deviation
f_{pbt}		Transverse base pitch deviation
f_{pS}	1)	Pitch sector deviation
f_{pt}	1)	Single pitch deviation
$f_{w\beta}$		Undulation height (along helix)
f_{α}	1)	Pressure angle deviation (normal)
$f_{\alpha m}$	1)	Mean pressure angle deviation
f_{β}	1)	Helix angle deviation
$f_{\beta m}$	1)	Mean helix angle deviation
F_p		Total cumulative pitch deviation
F_{pk}	1)	Cumulative pitch deviation
F_{pkS}	1)	Cumulative pitch sector deviation
F_{pS}		Total cumulative pitch sector deviation
F_i'		Total tangential composite deviation (with master gear)
F'		Total transmission deviation (product gear pair)
F_{α}		Total profile deviation
F_{β}		Total helix deviation

1) These deviations can be + (plus) or - (minus)

3.3 Gear inspection terms

$d_{b\ eff}$	Effective base diameter
k	Number of successive pitches
l	Left hand helix
r	Right hand helix
C_a	Tip relief
C_f	Root relief
C_α	Profile barrelling
C_β	Tooth crowning
C_I (C_{II})	End relief at reference (non-reference) face
L	Left flank
L_{AE}	Active length
L_{AF}	Usable length
L_E	Base tangent length to start of active profile
L_α	Profile evaluation range
L_β	Helix evaluation range
N...	Number of a tooth, number of a pitch
R	Right flank
λ_β	Wave length of undulation (in direction of helix)

$\lambda_{\beta x}$	Axial wavelength of undulation
ξ	Involute roll angle
I	Reference face
II	Non-reference face

4. EXTENT OF GEAR INSPECTION

Inspection of the various gear tooth elements requires several measuring operations. It is necessary to ensure that for all measurements involving rotation of the gear, the in-service axis of the gear coincides with the axis of rotation during the measuring process.

It may not be economical or necessary to measure all gear tooth element deviations such as those of single pitch, cumulative pitch, profile, helix, tangential and radial composite deviation, runout, surface roughness etc., for some of the elements concerned may not significantly influence the function of the gear under consideration. Furthermore, some measurements can often be substituted for others, for example the tangential composite check might replace pitch checking or the radial composite check might replace runout inspection. In order to take account of these aspects, recommended test groups and tolerance families relative to the function of gears are included in ISO / TR 10064, part 3. However, it is emphasised that curtailment of quality control measures is subject to agreement between purchaser and supplier.

5. IDENTIFICATION OF DEVIATION POSITION

It is convenient to identify deviations associated with measurements of gear teeth by specific reference to individual right flanks, left flanks, pitches or the groups of these.

In the following, conventions are described which enable positive determination of the location of deviations.

5.1 Right or left flank

It is convenient to choose one face of the gear as reference face and to mark it with the letter "I". The other non-reference face might be termed face "II".

For an observer looking at the reference face, so that the tooth is seen with its crest uppermost, the right flank is on the right and the left flank is on the left.

Right and left flanks are denoted by the letters "R" and "L" respectively.

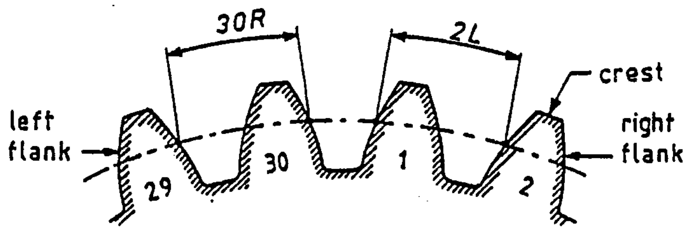


Fig. 1 Notation and numbering for external gear
30 R = pitch Nr. 30, right flank
2 L = pitch Nr. 2, left flank

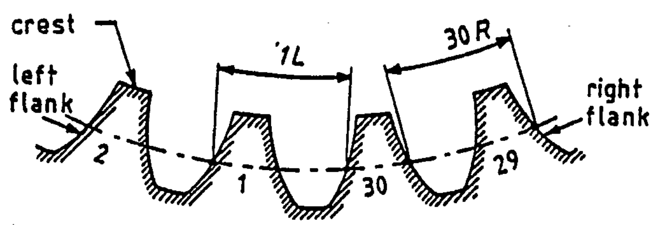


Fig. 2 Notation and numbering for internal gear
1 L = pitch Nr. 1, left flank
30 R = pitch Nr. 30, right flank

5.2 Right hand or left hand helical gears

The helix of an external or internal helical gear is referred to as being right hand or left hand. The hand of helix is denoted by the letters "r" and "l" respectively.

The helix is right hand (left hand) as, when looking from one face, the transverse profiles show successive clockwise (anticlockwise) displacement with increasing distance from an observer.

5.3 Numbering of teeth and flanks

Looking at the reference face of a gear, the teeth are numbered sequentially in the clockwise direction. The tooth number is followed by the letter R or L, indicating whether it is a right or a left flank.
Example: "Flank 29 L".

5.4 Numbering of pitches

The numbering of individual pitches is related to tooth numbering as follows: pitch number "N" lies between the corresponding flanks of teeth numbers "N-1" and "N"; with a letter R or L it is indicated whether the pitch lies between right or left flanks. For example "Pitch 2 L", (see Fig. 1)

5.5 Number of pitches "k"

The subscript "k" of a deviation symbol denotes the number of consecutive pitches to which the deviation applies.

In practice, a number is substituted for "k", for example F_{p3} indicates that a given cumulative pitch deviation refers to three pitches.

5.6 Checking recommendations
Measurements are normally carried out at approximately mid tooth depth and/or mid facewidth, as appropriate.If the facewidth is larger than 250 mm, two additional profile measurements, each approximately 15% of the facewidth distant from either end of the facewidth, is advisable. Profile and helix deviations should be measured over three or more equally spaced, corresponding flanks.

In order to ensure accuracy of measurements, inspection apparatus should be calibrated periodically against approved standards.

6. THE CHECKING OF SINGLE AND CUMULATIVE PITCH DEVIATIONS

6.1 General

Checking of pitch deviations implies measurement of the actual (angular) values or comparator checks between corresponding flanks of teeth around the circumference of a gear.

In contrast to the checking of normal, transverse and cumulative pitch deviations, base pitch deviations are checked in base tangent planes and are therefore independent of the gear axis.

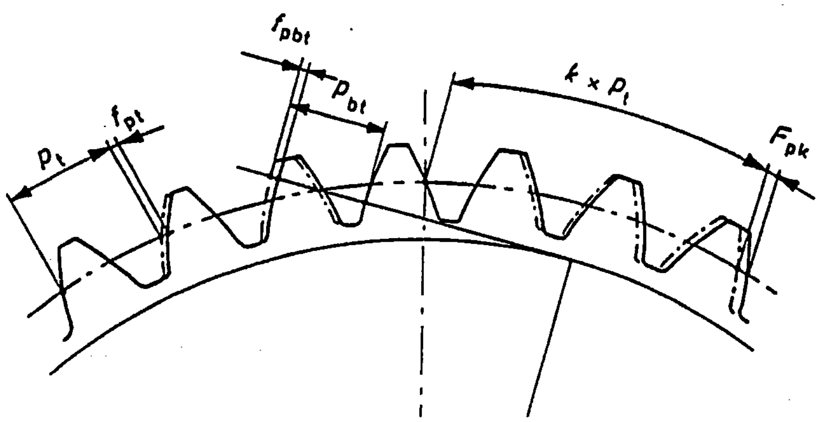


Fig. 3 Pitch (p_t), pitch deviation (f_{pt}), transverse base pitch (p_{bt}), transverse base pitch deviation (f_{pbt}), cumulative pitch ($k \times p_t$, in the Fig. k=3), cumulative pitch deviation (F_{pk},in the Fig. k=3)

N	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18
A	25	23	26	24	19	19	22	19	20	18	23	21	19	21	24	25	27	21
B	22.00																	
C	+3	+1	+4	+2	-3	-3	0	-3	-2	-4	+1	-1	-3	-1	+2	+3	+5	-1
D	+3	+4	+8	+10	+7	+4	+4	+1	-1	-5	-4	-5	-8	-9	-7	-4	+1	0

Fig. 4 Sample table with hypothetical deviation values obtained by single pitch checking with a comparator. In practice, integer values are seldom encountered.

N = pitch number

A = Values obtained with a pitch comparator (two probes), without reference to a defined absolute value

B = Arithmetic mean of all values A

C = Pitch deviations f_{pt}, expressed as the difference between individual values and mean value B

D = Cumulative pitch deviations, acquired by consecutive addition of f_{pt} (C) values, in the Fig. referred to the flank between the pitches 18 and 1, corresponding to the descriptions in Fig. 4 and Fig. 5.

When angular pitch measurement (one probe) is applied, values D are ascertained by subtracting the theoretical angle from the measured angle at each position, then multiplying the differences (in radians) by the radial distance to points of probe/flank contact. Values C are then obtained by subtracting value D of flank number N-1 from value D of flank number N.

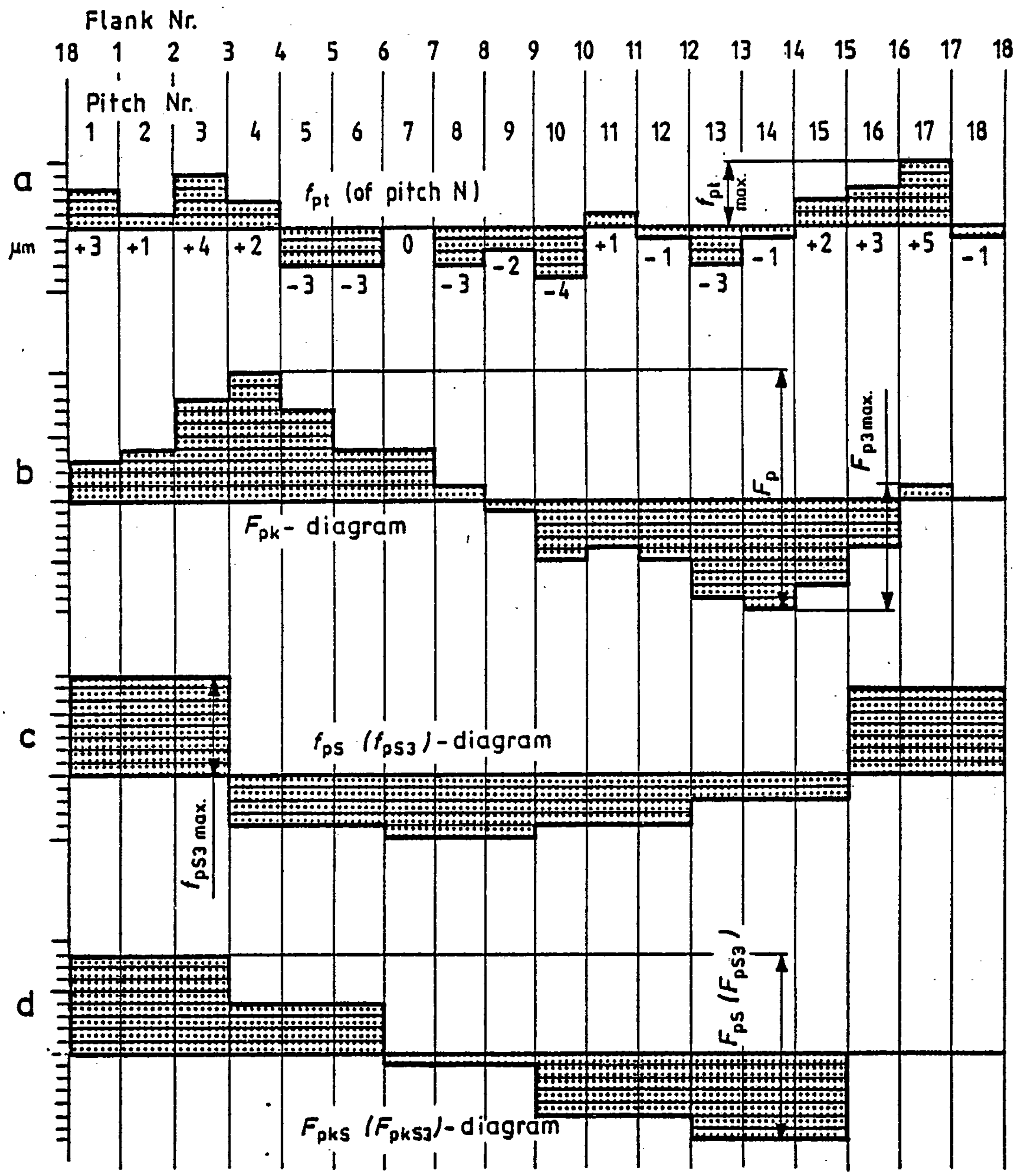


Fig. 5 A diagrammatic representation of pitch deviations on the sample gear of Fig. 4 (z = 18)

a Single pitch deviation f_{pt}; f_{pt} max. = + 5 µm, at pitch 17

b Cumulative pitch deviations F_{pk}, in the Fig. referred to flank 18, F_{pk} max.= total cumulative pitch deviation F_p = 19 µm, between flank 4 and flank 14.
F_{p3} max.= 10 µm, between flanks 14 and 17.

c Pitch sector deviations f_{pS}, measured over sectors of S = 3 pitches each.
f_{pS3} max. = 8µm, between flanks 18 and 3

d Cumulative pitch sector deviations F_{pkS}, in the Fig. referred to flank 18, derived from pitch measurement by sectors (c). Total cumulative pitch sector deviation F_{pS} = F_{pS3} = 15 µm, between flanks 3 and 15.

In general, for large number of teeth, the difference between F_p and F_{pS} becomes negligible.

6.2 The checking of single pitch accuracy

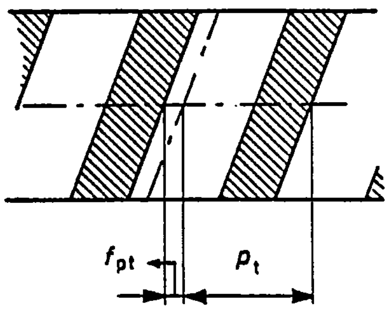


Fig. 6 Transverse pitch p_t and single pitch deviation f_{pt}

For checking pitch accuracy, commonly used devices are either a comparator, provided with two probes, or an angular dividing apparatus having a single measuring probe.

Inspection practices relevant to these processes are described in clauses 6.2.1 and 6.2.2 respectively.

Coordinate measuring machines without a rotating table can also be used for measurements of pitch and pitch deviations by applying appropriate relative motions which generally correspond to the principle described in clause 6.2.2.

6.2.1 Single pitch checking with a pitch comparator (two probes)

The two probes are to be positioned at the same radial distance from the gear axis and in the same transverse plane. The direction of the probe displacement should be tangential to the measuring circle.

Since the exact value of the radial distance is difficult to ascertain, such comparators are seldom used to verify true values of transverse pitches. Thus the most suitable use of such instruments is for the determination of pitch deviation.

Some pitch comparators are equipped with slides which advance the probes to a constant radial depth, normally to approximately mid tooth depth (Fig. 7). The gear under inspection turns slowly, either continuously or intermittently around its axis, and the probes on the slide are moved to and from the gauging position.

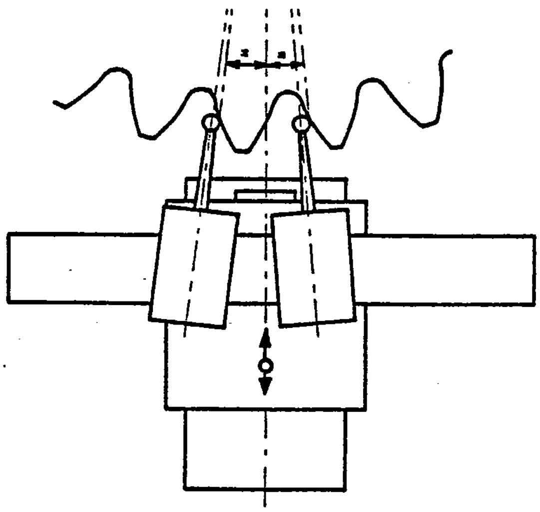


Fig. 7 Pitch checking with a pitch comparator

6.2.2 Pitch checking applying the angular indexing method (one probe)

This process involves the use of an angular indexing apparatus. The degree of its precision must be consistent with the gear diameter.

The measuring head is moved radially to and from a predetermined gauging position at which for each flank, the positional deviation from the theoretical position, is measured. Every value recorded represents the positional deviation of the relevant flank with respect to the selected reference or zero flank. A chart of recorded values thus shows cumulative pitch deviations (F_{pk}) around the gear circumference.

Each single pitch deviation is determined by means of subtracting the positional deviation of flank number N-1 from that of flank number N. Minus values are to be indicated as appropriate.

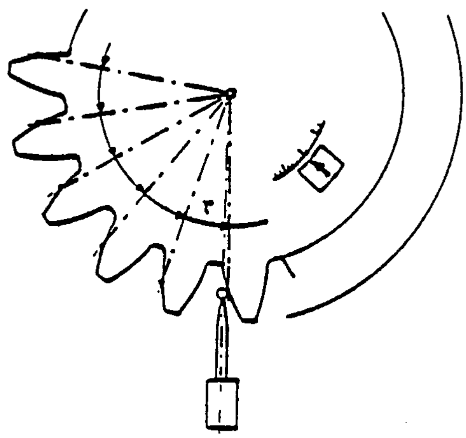


Fig. 8 Pitch checking applying the angular indexing method

6.3 The checking of normal pitch accuracy with a pitch comparator

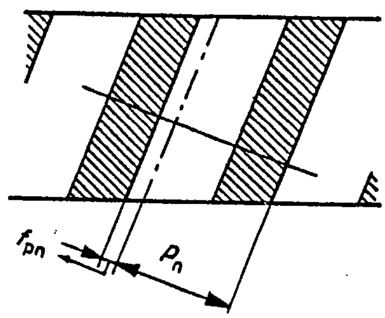


Fig. 9 Normal pitch p_n and normal pitch deviation f_{pn} (normal section)

Normal pitch deviation measurements should only be substituted for transverse pitch deviation measurements, when no suitable instrument other than a portable comparator, suitable only for checking "normal" pitch deviations, is available. With an instrument such as that illustrated in Fig. 10, the tip cylinder of the gear is used for positioning and it must be adequately concentric with the gear axis. Other comparators which can be used for the same purpose have different means for positioning and do not use the tip surface as a location surface.

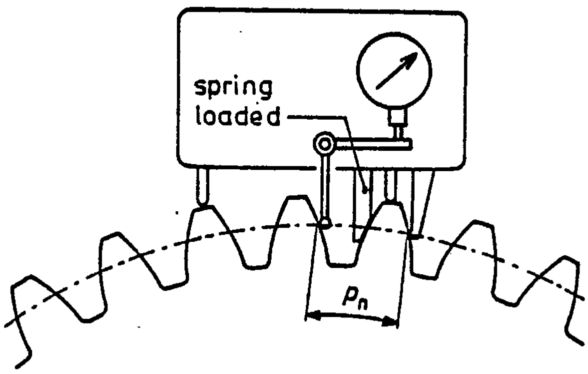


Fig. 10 Portable pitch comparator for checking normal pitch deviation, presented on a spur gear

Because the limits of tolerances set out in the standard ISO 1328, part 1, refer to transverse pitch, the results of normal pitch deviation measurements are to be converted to transverse values before comparison with tolerance values is made.

The relationship is as follows:

$$f_{pt} = \frac{f_{pn}}{\cos\beta}$$

Alternatively, tolerance values can be multiplied by $\cos\beta$ in which case fewer calculations are likely to be necessary.

Normal pitch deviation measurements should not be summed to determine cumulative pitch deviation.

6.4 The measurement of base pitch p_b and of base pitch deviations f_{pb}

The transverse base pitch of a gear is equal to the length of the common normal to the transverse profiles of two consecutive corresponding tooth flanks. It is also the length of arc of the base circle between the origins of the involute profiles of consecutive corresponding flanks (Fig. 11).

$$p_{bt} = d_b \times \frac{\pi}{z}$$

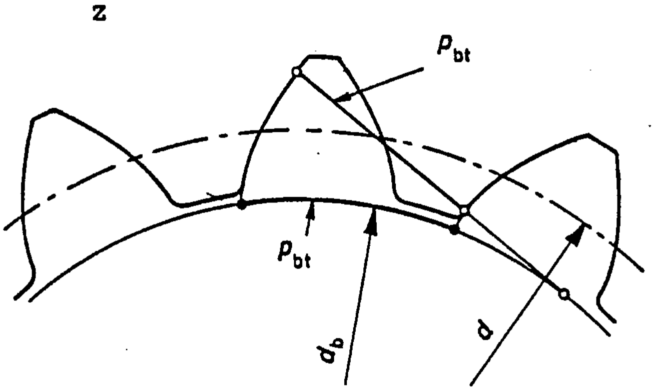


Fig. 11 Transverse base pitch p_{bt}

The normal and the transverse base pitches are related in accordance with the following equation:

$$p_{bn} = p_{bt} \times \cos\beta_b$$

Effective load sharing between the teeth of mating gears requires adequate control of base pitch accuracy of both elements. This is particularly important when gears of both elements are required to be interchangeable. In such cases an important measurement objective is determination of the value of the mean base pitch for comparison with the mean base pitches of other gears in the range.

The theoretical value of normal base pitch is a function of normal module and normal pressure angle, thus:

$$p_{bn} = m \times \pi \times \cos\alpha_n$$

Usually a portable comparator is used for the measurement of normal base pitch deviations. The principle of such an instrument is illustrated in Fig. 12. With the aid of a suitable gauge, the base pitch comparator can be calibrated to measure directly the deviations deviations from a theoretical base pitch.

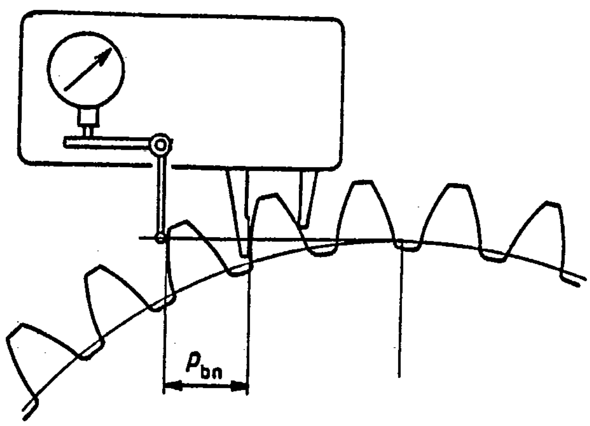


Fig. 12 Portable instrument for measuring base pitch, presented on a spur gear

When measuring base pitch, it must be ensured that the points of contact with the comparator probes do not lie in zones with profile or helix modifications.

When suitable profile checking equipment is not available, measured values of base pitch deviation can serve as a base from which the value of the pressure angle deviation f_α can be derived. Because measured values of base pitch deviation are influenced by pitch deviations and profile form deviations, this procedure will only serve a useful purpose when the two latter deviations are quite small.

In any calculations for the derivation of approximate mean values of pressure angle or other deviations, a mean value of base pitch is used.

The mean normal base pitch deviation f_{pbm}, the mean base diameter difference f_{dbm}, the mean pressure angle deviation $f_{\alpha m}$ and the effective base diameter $d_{b\ eff}$ are related as follows:

$$f_{dbm} = \frac{f_{pbm} \times z}{\pi \times \cos\beta_b}; \qquad f_{\alpha m} \approx -\frac{f_{pbm}}{m \times \sin\alpha \times \pi}$$

$$d_{b\ eff} = d_b + \frac{f_{pbm} \times z}{\pi \times \cos\beta_b}$$

6.5 Determination of cumulative pitch deviations F_{pk} and F_p

Cumulative pitch deviations can be determined by means of the algebraic summation of any specified number of individual measured values of single transverse-pitch deviations (see Fig. 5b). The individual values of single pitch deviation are determined in accordance with clause 6.2.1.

The angular indexing method, described in clause 6.2.2, provides directly the values of cumulative pitch deviation.

6.5.1 Determination of the total cumulative pitch deviation F_p

By definition, the "total cumulative pitch deviation" is the maximum cumulative pitch deviation of any sector of the corresponding flanks of a gear. Its value is equal to the distance measured at the appropriate scale between the highest and lowest points of the curve of cumulative pitch deviation. See Fig. 5b.

6.5.2 Cumulative pitch checking over sectors

When the comparator single pitch checking method is applied to gears with large number of teeth, accumulation of large numbers of measurement inaccuracies can result in substantial inaccuracies of values obtained by the summation process. One source of inaccuracy is failure to ensure that the trailing probe always contacts the point occupied by the leading probe during the preceding measurement.

By checking sectors of pitches, the possible frequency of the last mentioned inaccuracies will be reduced and it is recommended that measurement of sector deviations is adopted for gears having more than 60 teeth.
Fig. 13 illustrates the principle of measuring the deviation of a sector of 4 pitches, including e.g. pitches numbers 1 to 4. The next sector to be measured would include pitches numbers 5 to 8, when the trailing probe which is seen on the right comes into contact with that point on the flank of tooth number 4 which was

previously occupied by the leading probe, as seen on the left of the figure. The precautions described in clause 6.2.1 are equally necessary to measurements of pitch sector deviations.

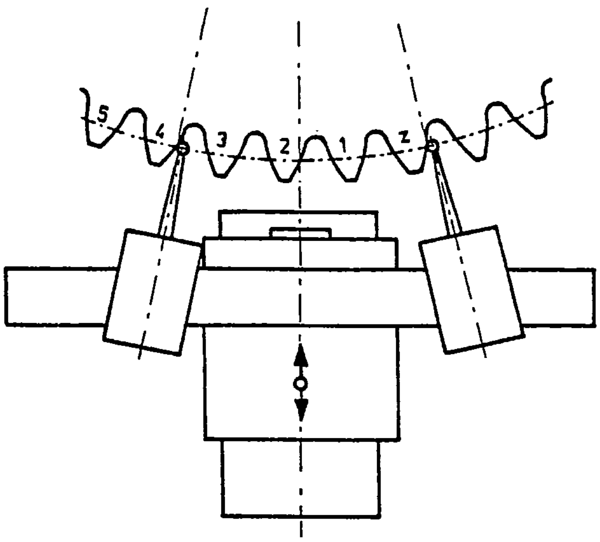


Fig. 13 Principle of pitch measurement by sectors

It is necessary to choose the number(S) of pitches per sector such that:

a) The length of the sector chord is suited to the capacity of the comparator to be used.
b) The number of values obtained will suffice for the plotting of an acceptable cumulative deviation curve.

Aids to the choice of suitable numbers of pitches, by formula and curves, are provided in Fig. 14.

If possible, z/S should be an integer.

When, however, the quotient z/S is not an integer, the number of sector deviation measurements should be equal to the next whole number larger than z/S, in which case the last sector will include some of those pitches already included in the first sector of pitches.

Example

z = 239, m = 8, chosen S = 5

Hence, the number of sectors (of readings) must be at least equal to 239 : 5 = 47,8. With 48 sectors (readings) there is an overlap of (5 x 48) - 239 = 1 pitch.

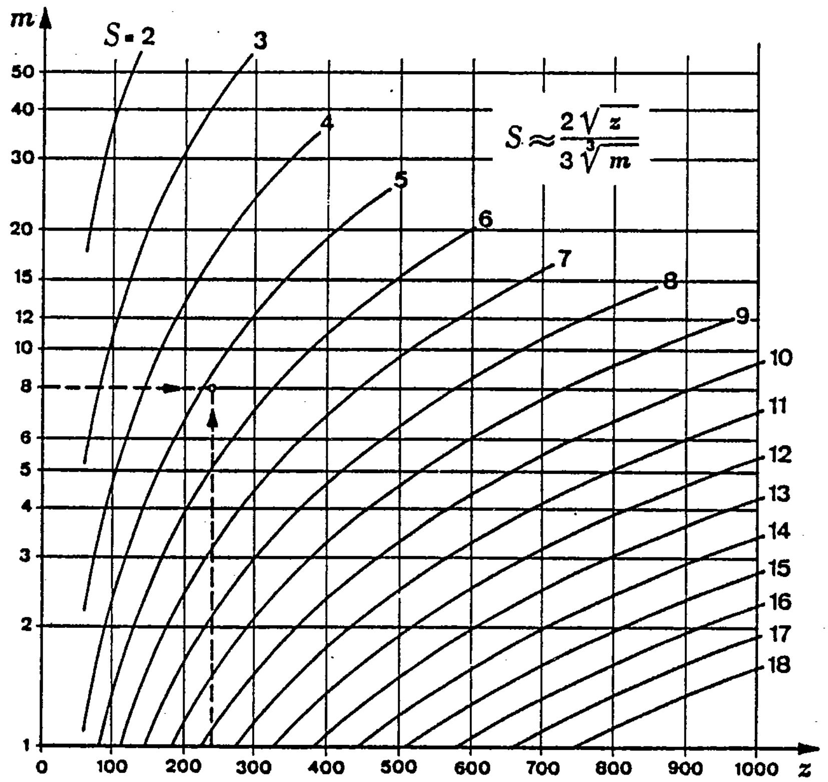


Fig. 14 Guide for choosing the number (S) of pitches per sector for pitch measurement by sectors

6.5.3 Evaluation of pitch sector checking results

It is important to recognize, that the total cumulative pitch deviation is not always revealed in a curve based on algebraic summation of pitch sector deviations. This is because the effect of any extreme single pitch deviations lying within sectors, which would otherwise influence the value of the total cumulative pitch deviation, can be compensated within the sector.

Thus if any values come close to exceeding the limits of specified tolerances, single pitch deviations in minimum, maximum and overlap zones should be blended into the cumulative pitch sector deviation curve in order to determine the total cumulative pitch deviation more accurately.

It will be found convenient, to substitute numerical values in the subscripts of the symbols F_{pS} and F_{pkS} representing cumulative pitch sector deviation. By this means, the relevant arc length and/or the number of pitches per sector can be indicated. For example, F_{p24S4} indicates the cumulative pitch sector deviation over an arc of k=24 pitches, based on measurements over sectors of 4 pitches.

6.5.4 Significance of cumulative pitch deviation F_{pk}

If cumulative pitch deviations over relatively small numbers of pitches are too large, in service conditions substantial acceleration forces will be generated. This holds especially true for high-speed gears, where these dynamic loads can be considerable. Hence the need for cumulative pitch tolerances over small numbers of pitches.

Fig. 15 shows cumulative pitch deviation diagrams for two gears. The total cumulative pitch deviations shown by each curve are similar, but maximum cumulative pitch deviations over small numbers of pitches are markedly different, as seen in the sectors "k" in curves "a" and "b". Depending on specified tolerances, the deviation F_{p4} in curve "a" of F_{pk} could be tolerable, whereas that in curve "b" may be unacceptable.

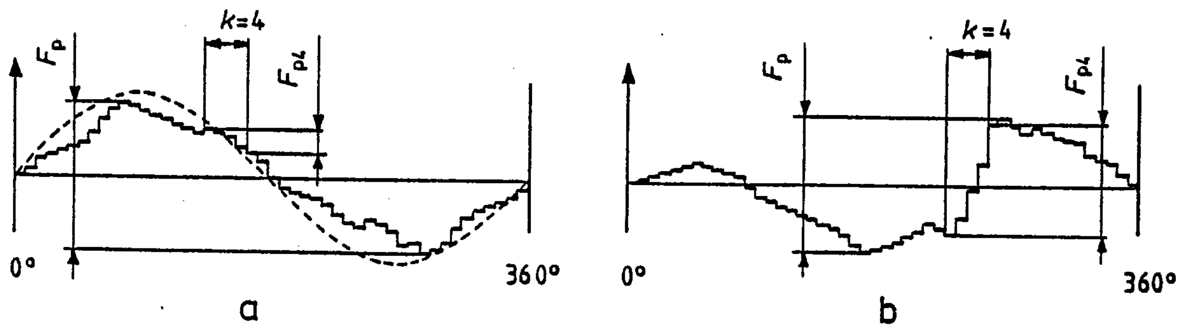


Fig. 15 Cumulative pitch deviation diagrams

The maximum cumulative pitch deviation F_{pk} over a specified number of k pitches can be derived from the F_{pk} diagram, by setting off from each flank of the gear in turn, the arc length ($k \times p_t$). In practice, the maximum value can be found by observation of a small number of sectors.

Considering the values provided as examples in Fig. 5 with k=3, the maximum cumulative pitch deviation over three pitches equal to 10µm is represented by the sum of the single pitch deviations of pitches numbered 15, 16 and 17.

A convenient way of identifying the position of any value of F_{pk} is to list the relevant pitch numbers in parentheses; for instance, the above example would be indicated as follows:

$$F_{p3}\ (15 \ldots 17) = 10\mu m$$

6.6 Notes on pitch deviation measurement and evaluation of results

Single probes and comparator probes usually have spherical ends. Each probe axis should be aligned parallel to the radial line from the gear axis through the point of probe/flank contact (Fig. 7, 8 and 13)

For all measurements of pitch deviations other than base pitch deviations measured with a base pitch comparator, radial and axial values of runout should be so small that they can be neglected. If however, the axis of the gear to be inspected is offset from the axis of rotation of the inspection apparatus and the position of the gauging device is fixed relative to the latter axis, a sinusoidal component having double amplitude equal to twice the eccentricity will be added to the true curve of cumulative pitch deviation of the gear.

The sinusoidal curve due to the above mentioned offset (excentricity f_e), which forms part of the cumulative pitch deviation diagram Fig. 15a, is derived from only one set of corresponding flanks. Its amplitude may be, and its phase will be, different from that of the curve of radial runout derived from both left and right flanks in combination, which will have double amplitude equal to $2f_e$.

An amount of eccentricity derived from measurements of cumulative deviation or tangential composite deviation, referred to right flanks or left flanks, is preferably denoted by the symbol f_{eR}, respectively f_{eL}.

7. THE CHECKING OF PROFILE DEVIATIONS

By definition (see ISO 1328, part 1, clause 4.2), profile deviations are normal to tooth profiles in transverse planes. Nevertheless, deviation may be measured normal to tooth flank surfaces and such measured values are to be converted before comparing them with limits of tolerances, by dividing the values by $\cos\beta_b$.

7.1 The profile diagram

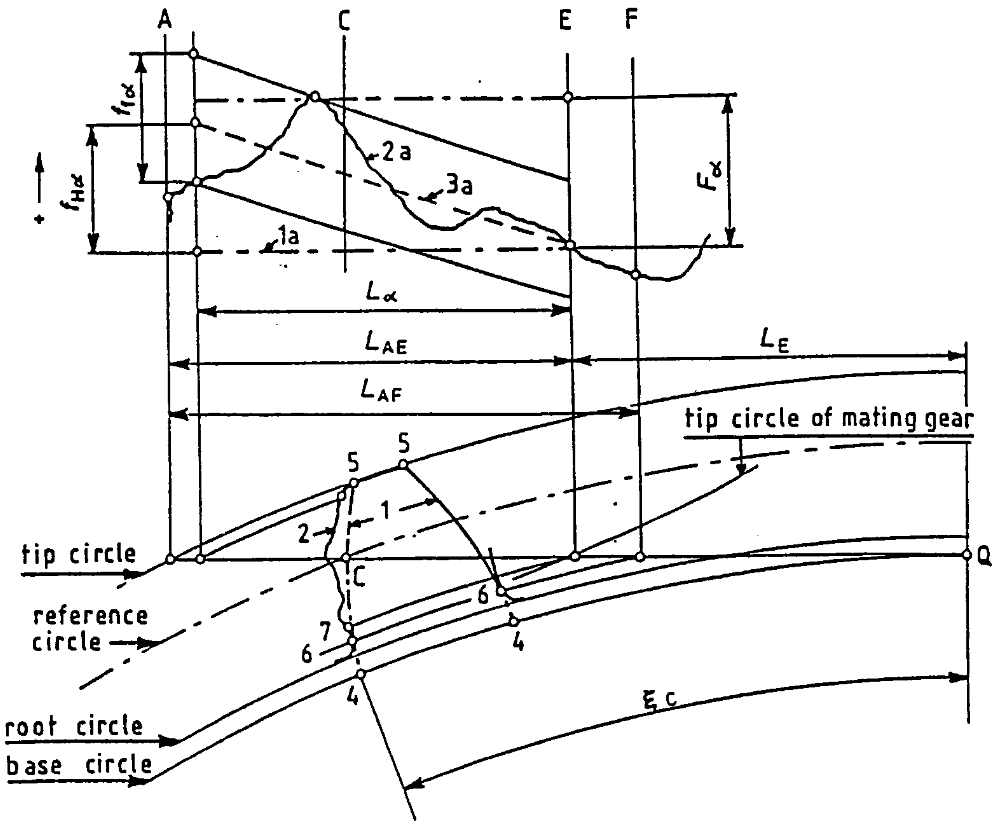
The profile diagram includes the profile trace, a curve traced on paper or other suitable medium, by gear tooth profile inspection equipment. Deviations of the curve from a straight line represent deviations of the profile from an involute curve generated from the base circle of the gear under inspection.

Profile modifications also appear as departures from the involute curve, but these are not considered to be deviations from the "design profile".

Any arbitrary point along the profile diagram can be related to a radius, a base tangent length and an involute roll angle.

Fig.16 shows a sample tooth profile and the relation to the corresponding profile trace, together with the appropriate terms. Details of terms, definitions and concepts concerning the profile trace, are provided in ISO 1328 part 1.

The profile evaluation range L_α is equal to the active length L_{AE} but shortened at the tip or chamfer point by 8%, in order to exclude from the evaluation unintentionally undersized tip zones which may result from the machining process and which do not impair gear performance. However, for assessment of total profile deviation (F_α) and profile form deviation ($f_{f\alpha}$), excess of material within the remaining zone of 8% which increases the amount of deviation must be taken into account. For deviations due to minus metal within that zone, tolerances are increased.



1 :Design profile	A :Tooth tip or start of chamfer
2 :Actual profile	C :Reference point
3 :Mean profile	E :Start of active profile
1a :Design profile trace	F :Start of usable profile
2a :Actual profile trace	L_{AF} :Usable length
3a :Mean profile trace	L_{AE} :Active length
4 :Origin of involute	L_α :Evaluation range
5 :Tip point	L_E :Base tangent length to start of active profile
5-6 :Usable profile	F_α :Total profile deviation
5-7 :Active profile	$f_{f\alpha}$:Profile form deviation
C-Q :Base tangent length to point C	$f_{H\alpha}$:Profile slope deviation
ξ_C :Involute roll angle to point C	
Q :Start of roll (point of tangency of transverse base tangent)	

Fig. 16 Tooth profile and profile diagram

7.2 Evaluation of profile diagrams

For purposes of gear quality classification, it is necessary to check only "Total profile deviation"F_{α}. See ISO 1328 part 1.

However, for some purposes it can be useful to determine the "profile slope deviation" $f_{H\alpha}$ and the "profile form deviation" $f_{f\alpha}$. For this it is necessary to superpose the "mean profile trace" onto the diagram as shown in Fig. 16, also in the Fig 2, 3 and 4 in ISO 1328 part 1, clause 4.2. Guidance for allowable values of $f_{f\alpha}$ and $f_{H\alpha}$ is given in the annex B of ISO 1328, part 1.

When profile deviations are measured normal to tooth flanks and have not been converted by the inspection apparatus to transverse values, the results are to be divided by $\cos\beta_b$ to convert them to the corresponding values normal to transverse profiles. The values so obtained can then be compared with specified limits of tolerances which refer to deviations measured normal to transverse profiles.

7.3 Algebraic signs of $f_{H\alpha}$ and f_α

The profile slope deviation is termed positive and the corresponding pressure angle deviation is termed negative when the mean profile trace rises towards the tooth-tip end A of the diagram, as shown in figure 16. In Fig. 17, both positive and negative slopes, caused by eccentricity of mounting on the gear generating machine are shown.

If the slopes seen in the profile diagrams of mating gears are equal and have the same sign, the deviations are mutually compensating. This applies to both external and internal gears.

7.4 Pressure angle deviation f_α

A profile rising towards its tooth tip end implies that the pressure angle is too small.

The pressure angle deviation f_α can be derived from the profile slope deviation using the following equations

in radians: $$f_\alpha = - \frac{f_{H\alpha}}{L_\alpha \times \tan\alpha_t \times 10^3}$$

in arc seconds: $$f_\alpha = - \frac{206,26 \times f_{H\alpha}}{L_\alpha \times \tan\alpha_t}$$

$f_{H\alpha}$ in µm, L_α in mm

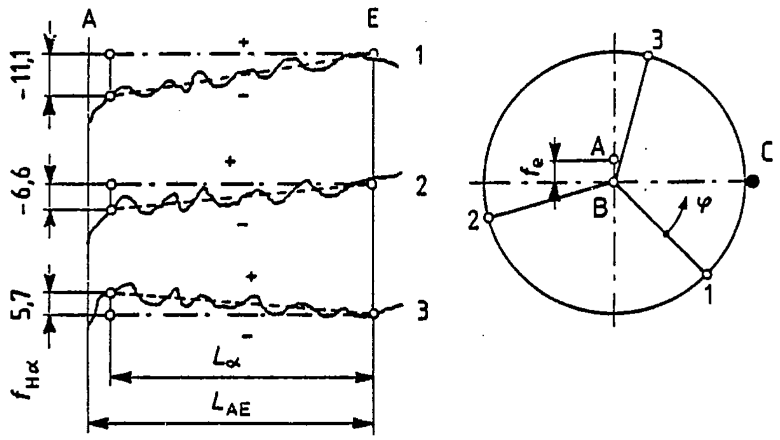
For both external and internal gears:

when $f_{H\alpha} > 0$, then $f_{db} > 0$ and $f_\alpha < 0$.

7.5 Mean profile slope deviation $f_{H\alpha m}$

Slope deviations of individual profiles can be caused by eccentricities due to inaccuracies of manufacturing or inspection set-up, however, such deviations will vary around the gear. In the mean value of profile slope deviations related to corresponding flanks, such variations are cancelled out.

The effect of eccentricity on profile slope, and the determination of mean profile deviation, are illustrated in Fig. 17.



$$f_{H\alpha m} = \frac{1}{3}(-11{,}1 - 6{,}6 + 5{,}7) = -4\ \mu m$$

A = The axis of rotation of the machine tool relative to that of the gear.

B = The axis of rotation of the inspection apparatus and that of the gear.

C = The position of tool or profile measuring probe

1,2,3 = Positions of the profiles from which the traces were obtained (at 45°, 165° 285°) and relevant profile traces

Fig. 17 Mean profile slope deviation $f_{H\alpha m}$

It is necessary to calculate the average value of the slope deviations of the profiles of corresponding flanks of gear teeth as a step towards an eventual decision on what steps are to be taken by way of correction of machine tool settings or other suitable action.

For all practical purposes, it is usually sufficient to calculate the arithmetic mean of the profile slope deviation of a limited number of equispaced corresponding flanks around the gear circumference.

A suitable mean value can be obtained from the profile traces of corresponding flanks of two diametrically opposite teeth. However, if profile slope deviations vary around the gear, this will not always be disclosed unless traces of the profiles of at least three equispaced corresponding flank are obtained.

7.6 Base diameter difference f_{db}, mean base diameter difference f_{dbm} and effective base diameter $d_{b\ eff}$

The base diameter difference $f_{db} = d_{b\ eff} - d_b$ is directly related to the profile slope deviation $f_{H\alpha}$. The relationship is as follows:

$$f_{db} = f_{H\alpha} \times \frac{d_b}{L_\alpha}$$

Thus, when the "mean profile slope deviation" $f_{H\alpha m}$ is determined (see clause 7.5), the mean base diameter difference and the effective base diameter can be derived from the following equations:

$$f_{dbm} = f_{H\alpha m} \times \frac{d_b}{L_\alpha}$$

$$d_{b\ eff} = d_b \left(1 + \frac{f_{H\alpha m}}{L_\alpha} \right)$$

7.7 Profile tolerance field

A convenient inspection procedure is to check whether or not the profile trace can be enclosed in the specified tolerance field.

Many of the tolerance fields specified have forms roughly resembling the letter "K" (Fig 18), hence the well known term "K-chart".

The use of such a chart is illustrated in Fig. 18a in which the profile trace lies within the tolerance field, whereas in Fig. 18b the profile trace does not.

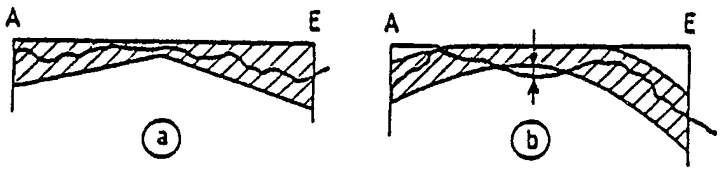


Fig. 18 Inspection of profile accuracy by the tolerance field method

If need be, a combination of the two types of evaluating the profile accuracy (with standard tolerances referred to a quality grade and with the tolerance field method) can be applied, as shown in the example in Fig. 19.

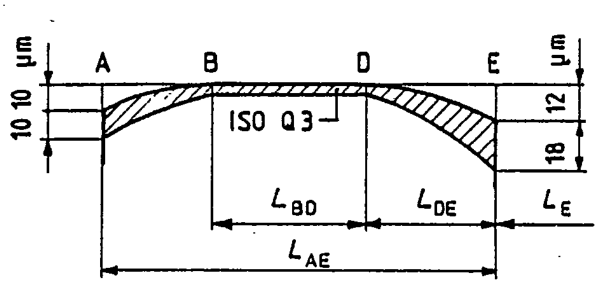


Fig. 19 Different tolerance systems for different profile zones

7.8 Profile barrelling C_α

For some fields of application, suitable profile modification involves tip and root relief of arcuate form which normally extends from about the middle of the evaluation range towards tips and roots of the gear teeth. See Fig. 20.

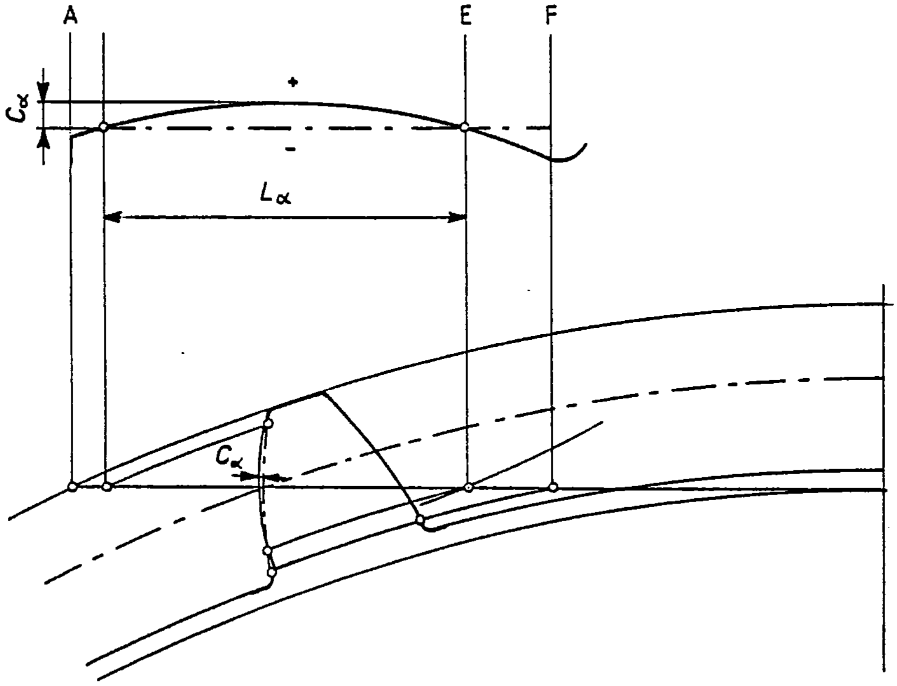


Fig. 20 Profile barrelling C_α

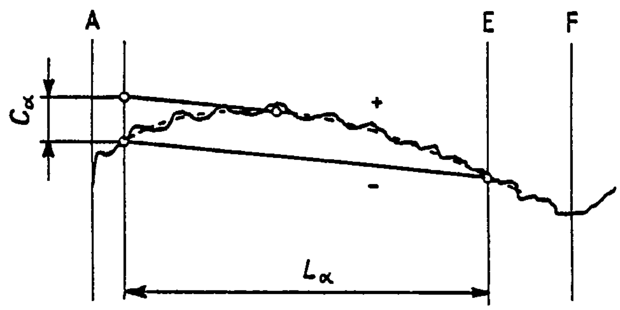


Fig. 21 Determination of profile barrelling C_α

The amount by which the height of the curvature of the involute is increased can be determined as described in the following.

In the diagram, a straight line is drawn through the points of intersection of the profile trace or its mean trace with the ends of the evaluation range, as shown in Fig. 21. The distance between this line and a parallel to it which is tangent to the mean curve, measured in direction of recorded deviations, is equal to the amount of profile barrelling (C_α).

In profile diagrams generated from intentionally barrelled teeth, design and mean profile traces are usually parabolic curves.

8. THE CHECKING OF HELIX DEVIATIONS

By definition, helix deviations are the amounts measured in the direction of transverse base tangents, by which actual helices deviate from design helices. If deviations are measured normal to tooth flanks, they are to be divided by $\cos\beta_b$ to convert the values to transverse quantities before comparisons with limits of tolerances are made.

8.1 The helix diagram

The helix diagram includes the helix trace, a curve generated on paper or other suitable medium by helix checking equipment. Deviations of the curve from a straight line represent, in magnified form, deviations of the actual helix from an unmodified helix.

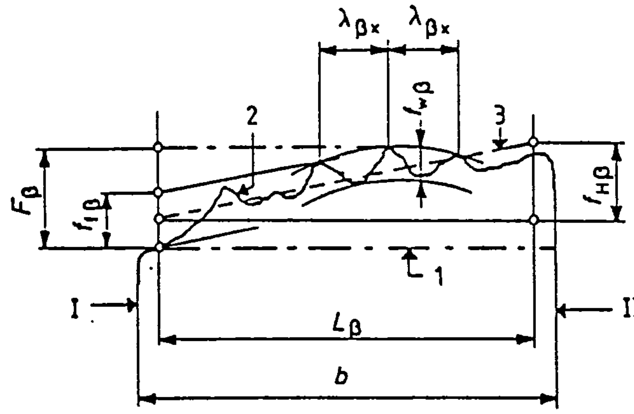
Helix modifications introduced by the designer also appear as departures from the straight line, but they are not considered to be deviations from the "design helix". See ISO 1328 part 1, clause 4.3.1.4.

Sometimes trace lengths are magnified representations of small facewidths, or reduced representation of large facewidths. See also "length of trace" ISO 1328 part 1, clause 4.3.1.2.

Relevance to right hand and left hand helices can be indicated by means of the letters "r" respectively "l" used either as symbols or as subscripts.

In Fig. 22, a typical example of a helix diagram shows the helix deviations of a tooth flank of which the design helix is an unmodified helix. Had the "design helix" been crowned, end relieved or otherwise modified, traces representing it would be appropriately formed curves.

Details of terms, definitions and concepts concerning the helix trace are provided in ISO-1328, part 1.



1	: Design helix trace	F_β	: Total helix deviation
2	: Actual helix trace	$f_{f\beta}$	: Helix form deviation
3	: Mean helix trace	$f_{H\beta}$	: Helix slope deviation
b	: Facewidth or distance between chamfers	$\lambda_{\beta x}$	: Axial wavelength of undulation
L_β	: Helix evaluation range	$f_{w\beta}$	: Undulation height
I	: Reference face	II	: Non-reference face

Fig. 22 Helix diagram

The helix evaluation range L_β is equal to the length of trace, reduced at each end by 5% of length of trace, but not by more than one module (1 * m). This reduction is made in order to ensure that unintentional, slight end reliefs caused by some machining conditions, are not normally included in the assessment of the deviation magnitudes intended for comparison with stringent tolerances. For assessment of the total helix deviation (F_β) and the helix form deviation ($f_{f\beta}$), excess of material within the end zones of 5% which increases the amount of deviation must be taken into account. For deviations due to minus metal within these end zones, tolerances are increased.

8.2 Evaluation of helix diagrams

For purpose of gear quality classification, it is necessary to check only "Total helix deviation" F_β. See ISO 1328 part 1.

However, for some purposes it can be useful to determine the "helix slope deviation" $f_{H\beta}$ and the "helix form deviation" $f_{f\beta}$. For this it is necessary to superpose the representative "mean helix trace" onto the diagram as shown in Fig. 22. Recommendations for tolerable values of $f_{f\beta}$ and $f_{H\beta}$ are given in the annex B to ISO 1328, part 1.

When helix deviations are measured normal to tooth flanks and have not been converted by the inspection apparatus to transverse values, the results are to be divided by $\cos\beta_b$ to convert them to the corresponding values normal to transverse profiles. The values so obtained can then be compared with specified limits of tolerances which refer to deviations measured normal to transverse profiles.

8.3 Determination of the helix slope deviation by axial pitch checking

Where helix diagrams cannot be obtained, for example on very large gears which cannot be checked on measuring machines, the "helix slope deviation" $f_{H\beta}$ can be determined from measurements taken with an axial pitch instrument.

Instruments of this type essentially comprise a precision level and two ball styli. The styli are adjusted so that the spacing of the balls is approximately equal to an integer number of axial pitches. The balls are placed in gear tooth spaces with the line of their centres roughly parallel to the gear axis. The level is then adjusted to zero and any relative tilt at other positions around the gear is determined. The tilt determinations, together with the distance between the styli can be related mathematically to the mean helix slope deviation of the flanks. The method is not of high precision.

If the measurements are taken at three or more equally spaced positions around the gear, the effect of transverse pitch deviations on the result tend to cancel, and an approximate mean "helix slope deviation" that is independent of the gear axis can be calculated.

Furthermore, provided neither gear flank has severe profile deviations and the gear flanks are not crowned, the mean "helix slope deviation" for the left and right flanks can be determined.

Measurements can be taken with the gear in any attitude without affecting their validity.

For this method to be used requires that the gears have a facewidth greater than one axial pitch.

8.4 Algebraic signs of $f_{H\beta}$ and f_β

The helix slope deviation $f_{H\beta}$ and the helix angle deviation f_β are to be completed with an algebraic sign.

Deviations are deemed to be positive ($f_{H\beta} > 0$ and $f_\beta > 0$) when helix angles are larger, and negative when helix angles are smaller, than the design helix angle.

The helix deviations of spur gears if other than zero are indicated by the subscripts "r" and "l", instead of an algebraic sign, implying deviations in the sense of right or left hand helices respectively.

If the helix deviations $f_{H\beta}$ and f_β of a gear and its mating gear flanks are equal in magnitude and algebraic sign, the deviations are mutually compensating.

8.5 Mean helix slope deviation $f_{H\beta m}$ and mean helix angle deviation $f_{\beta m}$

If during manufacture a gear was mounted with its axis offset from the axis of rotation of the gear generating machine or if these two axes were crossed, slope deviations of the gear tooth helices would vary around the gear. See Fig. 23.

Even when the deviations are within specified limits of tolerance, attention should always be drawn to this fault because of its possible contribution to gear vibrations in service and in order that steps might be taken to avoid future occurence of the same.

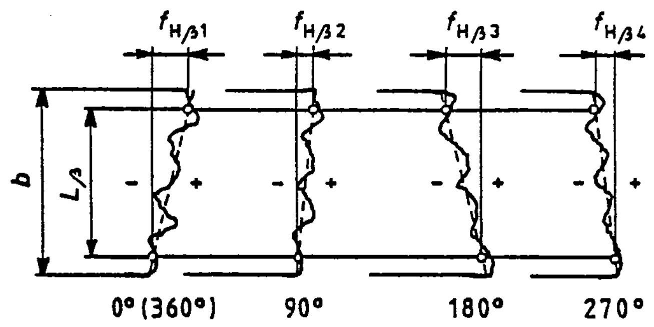


Fig. 23: Traces generated from four tooth flanks spaced equally arround gear, illustrating helix slope deviations associated with eccentricity or wobble.

For some purposes, e.g. for correction of machine tool setting or adaptation to mating gear, it is necessary to determine the mean helix deviation by calculating as follows the average of several deviations measured on three or more flanks of teeth spaced equally around the gear.

$$f_{H\beta m} = \frac{1}{n} (f_{H\beta 1} + f_{H\beta 2} + \dots + f_{H\beta n})$$

A suitable mean value can be obtained from the helix diagrams of corresponding flanks of two diametrically opposite teeth. However, if the helix slope deviations vary around the gear, this will not always be disclosed unless traces of at least three equispaced flanks are obtained.

8.6 Helix tolerance field

A convenient way of checking helix accuracy, is to verify whether or not the trace can be enclosed in the specified tolerance field.
Details of this method are to all intents and purposes the same as those described for the "profile tolerance field", see clause 7.7.

8.7 Tooth crowning C_β

In a diagram the helix trace of an unmodified tooth flank would be represented by a more or less straight line whilst the corresponding trace of a crowned tooth flank would be an arcuate curve. In helix diagrams generated from intentionally crowned teeth flanks, design and mean helix traces are usually parabolic curves.

The procedure for evaluation of profile barrelling C_α which is similar in form and which is described in clause 7.8, is equally applicable to evaluation of gear tooth crowning C_β.

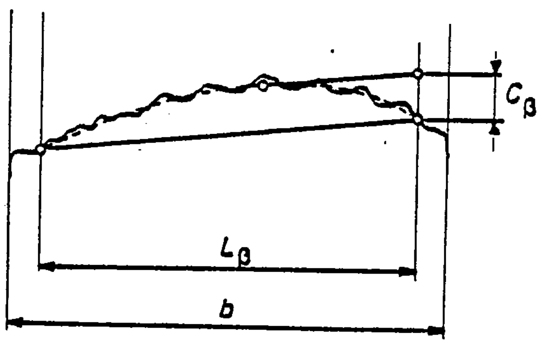


Fig. 24 Tooth crowning C_β

8.8 Undulations

Undulations are helix form deviations having constant wavelength and almost constant height. Perturbations of gear production machine transmission elements are their most common cause, especially those of:

a) the cutter saddle feedscrew drive, and
b) the worm of the indexing wormgear drive.

The wavelength of undulations caused by a), measured in direction of helix, is equal to the pitch of the feedscrew divided by cosβ.

Of undulations due to cause b) the wavelength is:

$$\lambda_\beta = \frac{d \times \pi}{z_M \times \sin_\beta}$$

The number of undulations generated as a result of b), projected into a transverse plane, are equal to the number of teeth z_M of the master indexing wormwheel. These can be sources of objectionable pure-tone components of noise spectra, at frequencies corresponding to the rotational speed (revolutions) of the affected gear multiplied by z_M.

The method of application of the undulation measuring attachment of a helix checking apparatus is shown in the diagram in Fig. 25. This is discussed in the following.

When undulations due to the causes a) or b) mentioned above are to be measured, the appropriate wavelength is calculated and the spherical location feet of the attachment are set at an odd number of wavelengths distant from each other.

The amount of the undulations are indicated by a probe situated midway between the feet as the latter are slid along the helix.

It can be seen in the figure that the displacement of the probe when a peak and next a trough are sensed by the probe, is equal to twice the height of the undulation as shown in Fig. 25. This feature enhances the sensitivity of the apparatus which also plots the results in the form of a diagram.

It should be noted that the undulations would not be indicated if the feet were spaced at a distance equal to an even number of wavelengths as shown in Fig. 25 with $s = 4\lambda_\beta$.

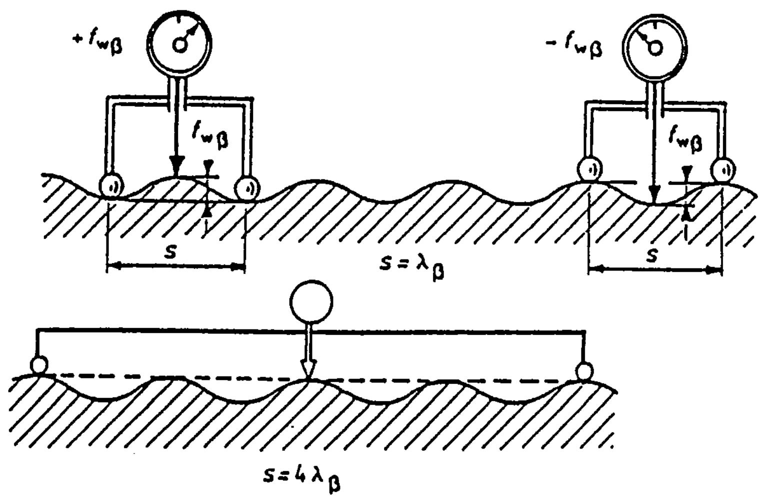


Fig. 25 Principle of undulation inspection

9. CHECKING OF TANGENTIAL COMPOSITE DEVIATIONS

9.1. General

For purposes of verifying tangential composite deviations, two gears one of which may be a master, are rotatably mounted in mesh at an appropriate centre distance. The gears are then rotated whilst contact occurs on only one set of corresponding flanks until a complete diagram is generated.

During the tangential composite deviation check, gear tooth flank contact is maintained under very light load and low angular velocities. Thus the generated records reflect the combined influence of the deviations of tooth elements of both gears of the pair (i.e. profile, helix and pitch).

The following combinations can be checked:

a) A product gear meshed with a master gear.
b) A pair of mating product gears.
c) A train of gears with more than two gears in mesh.

Concerning case a), a suitable record is generated during one turn of the product gear.
Attention must be paid to the fact that the accuracy of the master gear influences the checking results. Inaccuracies of the master gear are usually neglected, if its accuracy is by at least 4 quality grades higher than that of the product gear. If the quality of the master gear is less than four grades better than that of the gear to be inspected, inaccuracies of the master gear are to be taken into account.

The total tangential composite deviation (F_i') is the maximum difference between the effective and the theoretical circumferential displacement (at the reference circle) of the gear under inspection which is turned through one complete revolution.

The tooth-to-tooth tangential composite deviation (f_i') is the value of the tangential composite deviation over a displacement of one pitch.

Concerning case b), the generated deviations (F' and f') involving two product gears are termed "transmission deviations of a gear pair". In order to fully explore the complete spectrum of the deviations, it is necessary to continue rotation until both product gears have made the number of turns equal to the number of teeth of its mating gear divided by the highest common factor of the respective numbers of teeth of the two meshing gears (z_1 and z_2). The numbers of revolutions determined in this way correspond to the complete meshing period of the gear pair. The diagram generated reflects the components of element deviations of the teeth of both gears of the pair. If deviations of the teeth of the individual gears are to be identified, the data must be suitably processed, see clause 9.3.3.2.

Tangential composite deviations of heavily loaded gears can also be similarly checked when a suitable rig is available. Under such circumstances, recorded deviations are influenced by load induced tooth deformations, by mesh stiffness variations and, depending on the speed of rotation, by impact effects as well as by imperfections of tooth geometry. ISO 1328, part 1, does not apply to this kind of inspection.

The third case c) is the assessment of the kinematics of a gear transmission. Such inspection is not considered to be within the field of application of ISO 1328.

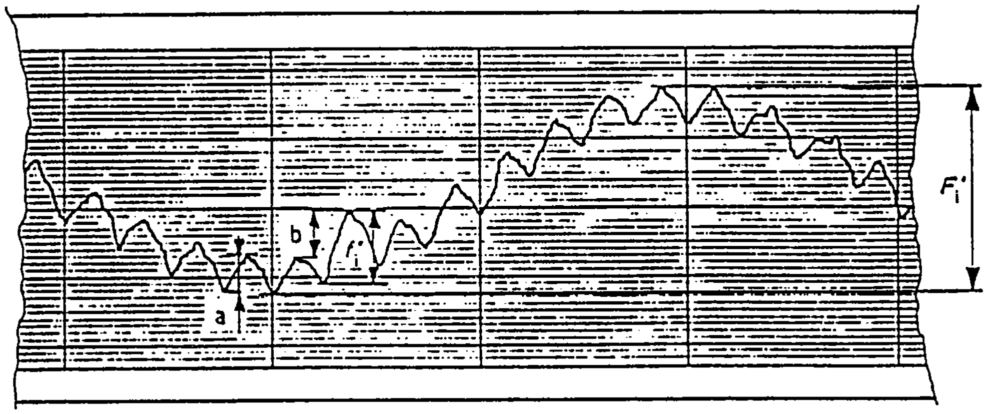
9.2 Checking product-gear / master-gear pairs

9.2.1 Spur gears

Recorded diagrams of tangential composite deviations generally include short period components corresponding to successive cycles of tooth engagement, superposed on long period components associated with complete revolutions of each of the meshing gears.

The diagram in Fig. 26 represents the record of tangential composite deviations generated during one revolution of a pinion having sixteen teeth when meshed with a master gear.

In the figure the "total tangential composite deviation" F_i', the maximum "tooth-to-tooth tangential composite deviation" f_i' as well as the profile component "a" and the single pitch component "b" are indicated.



f_i' = tooth-to-tooth tangential composite deviation (maximum value)

F_i' = total tangential composite deviation

a = deviation largely influenced by the profile deviation

b = single pitch component

Fig. 26 : Tangential composite deviation diagram of a spur gear

9.2.1.1 <u>Influence of profile deviations of spur gears</u>

When using a master gear in the checking of tangential composite deviations, the assumption that the master gear is perfectly accurate implies that the generated tangential composite deviation diagram represents only the combined deviations of the tooth elements of the product gear.

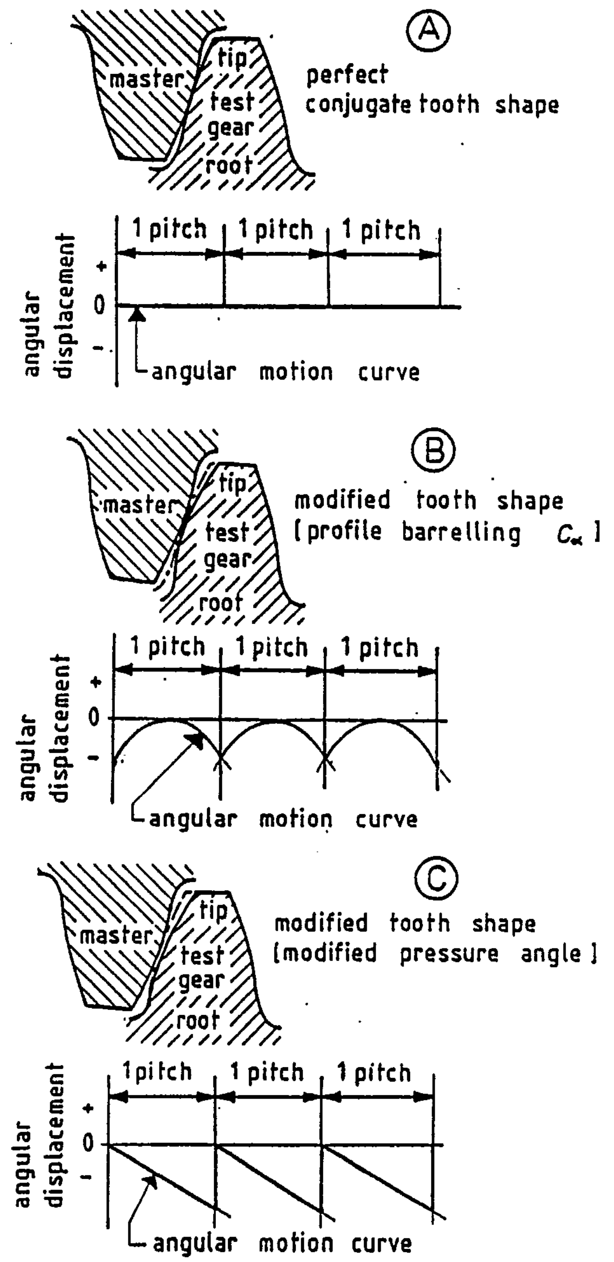


Fig. 27: Influence of profile deviations of spur gears

Fig. 27 shows schematically, tangential composite deviation recordings of three consecutive cycles of tooth engagement of a master gear and product gear, corresponding to each of three different tooth profiles, the first of which is unmodified and faultless, the second being progressively modified from mid-depth towards each limit of the active profile and the third with "slope deviation".

Fig. 27A shows the straight line diagram which would be generated by a test gear and master gear both of which have faultfree unmodified teeth.

In Fig. 27B, the record indicates the influence of tip and root relief in form of a profile barrelling over the whole profile (C_α). From the start of the tooth engagement cycle with first contact at the tooth tip of the driven product gear, the deviation value increases progressively to zero as contact nears mid-tooth depth, then changes to a progressively decreasing trend as contact approaches the end of the tooth engagement cycle.

In Fig. 27C, the triangular components of the diagram show progressive tangential composite deviation from zero to a negative value as contact moves from the product gear tooth tip towards the start of active tooth profile. At this point, contact abruptly transfers to the following tooth with the introduction of an equally abrupt positive tangential composite deviation.

It must be borne in mind that generated diagrams of tangential composite deviations do not merely reflect influences of profile deviations revealed by checks on a few teeth, but may be influenced by contact involved in any prominences on the working surfaces of the teeth of the product gear.

9.2.1.2 Influences of pitch deviations of spur gears

In the event of a pitch deviation at pitch N: When during the course of rotation contact is transferred from tooth number N-1 to tooth number N, a local tangential component will be introduced which will show on the tangential composite deviation diagram as a displacement of one of the profile-generated components of the diagram.

The schematic diagram in Fig. 28 illustrates the influence of single pitch deviations on the tangential composite deviation diagram.

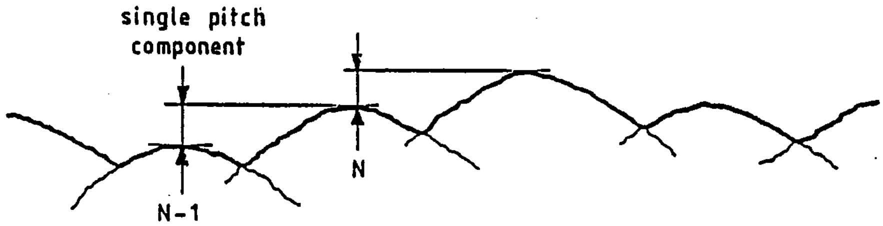


Fig. 28: Influence of single pitch deviation of spur gears

Single pitch deviations have cumulative effect on the tangential composite displacement arc as they pass through the mesh. Their influence is clearly visible on the tangential composite deviation diagram thus enabling values of cumulative pitch deviations, (e.g. when k=2, k=3 and etc.) to be determined as the ordinates of tangents to the apices at appropriate numbers of pitches apart. The principle is illustrated in Fig. 29, in which influences of single pitch deviation, combined single pitch and profile deviation and also approximate total cumulative pitch deviation are indicated.

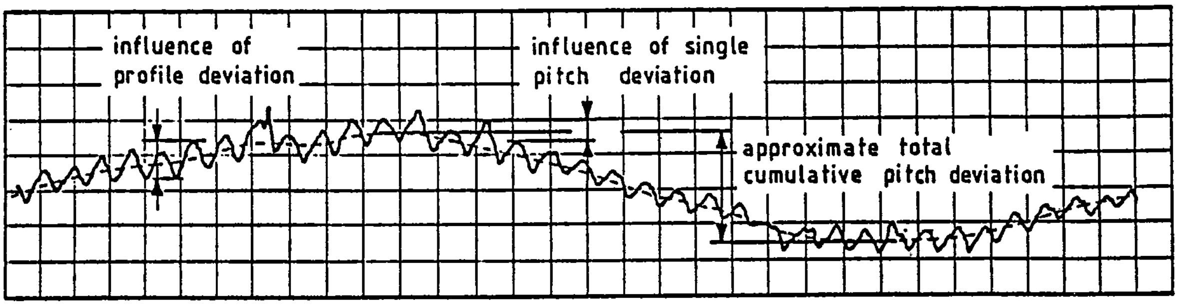


Fig. 29: Components of a tangential composite deviation diagram of a spur gear

9.2.1.3 <u>Influence of helix deviations of spur gears</u>

A helix deviation which is constant in magnitude and sign, i.e. is common to every tooth of a gear, results in consistent localised bearing in the mesh. The tangential composite deviations are not thereby influenced substantially.

When helix deviations vary in magnitude and sign (i.e. direction) around a product gear, the tangential composite deviations may be affected. A change of magnitude of helix deviation may influence the tangential composite deviation.

If under these latter circumstances profile deviations are different at opposite ends of the mesh, then the profile (tooth-to-tooth) components of the tangential composite deviation diagram will also be affected.

9.2.1.4 Influence of contact ratio of spur gears

A tangential composite deviation diagram generated from a "master-gear"/"product-gear" combination is composed of successive curves representing for the most part the profile deviations, as shown in Fig. 30 The relationship between the phases of "two-pair"/ "single-pair"/two-pair" tooth engagement and the tangential composite deviation diagram during a complete cycle of tooth engagement is clearly illustrated. It can easily be recognised that the maximum length of the single-pair tooth contact path is realised when the contact ratio ε_α is equal to one. As the contact ratio increases, this length reduces and when the contact ratio is equal to or greater than two, there is no single-pair tooth contact at all.

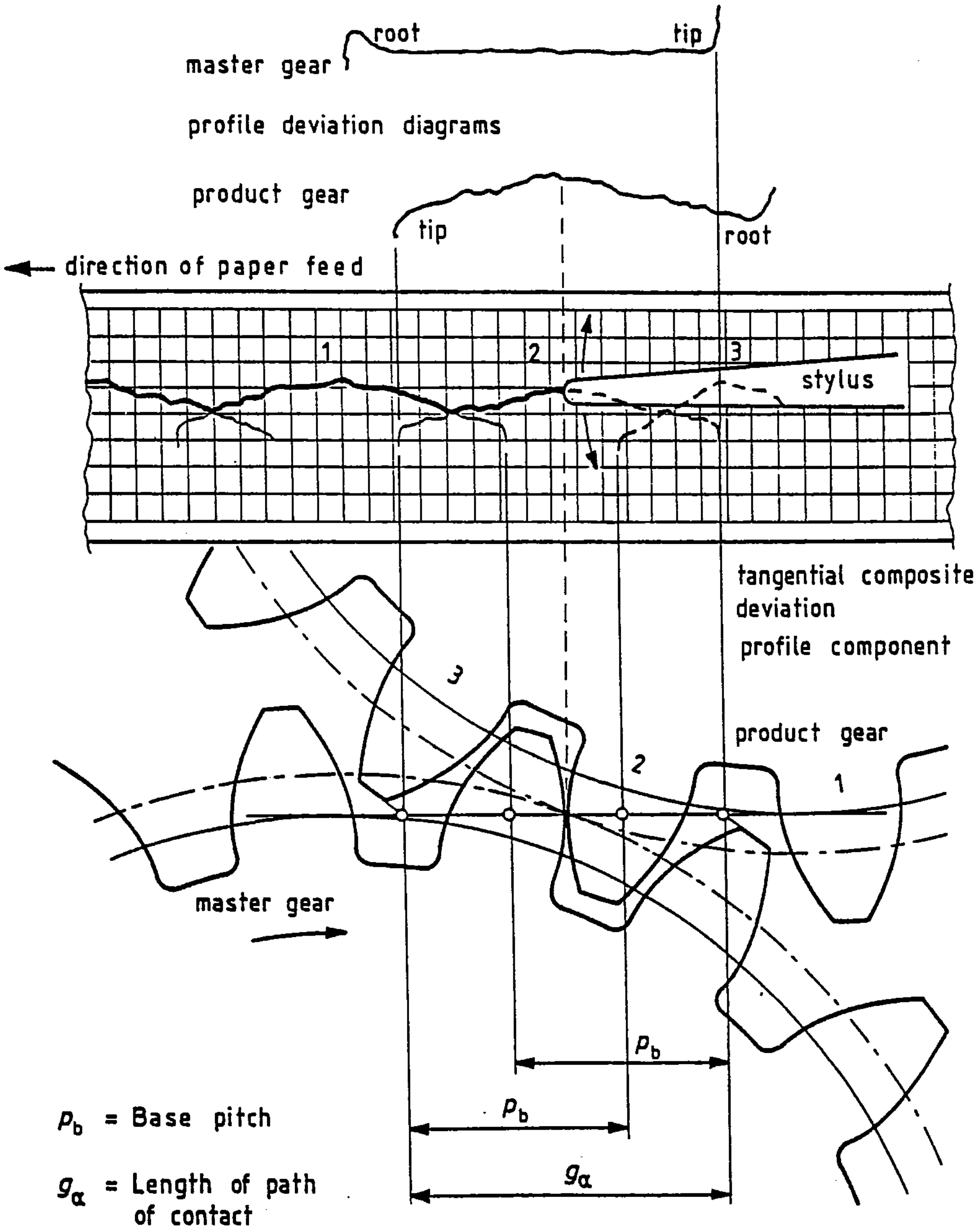


Fig. 30: Effect of contact transfer on the profile component in a tangential composite deviation diagram (spur gears)

In order to derive the maximum amount of useful data, the teeth of the master gear should be made as deep as is consistent with adequate tooth tip width. This enables checks to be made at extended centre distance such that the contact ratio is unity and other checks to be made with the centre distance so adjusted that the "in-service" working flanks are fully explored.

9.2.2 Helical gears

When the total contact ratio ε_γ is less than 2,0, the meshing conditions for helical gears are similar to those of spur gears of which the contact ratio ε_α is less than 2,0, in which case, all of the above comments concerning spur gears, apply equally to such helical gears.

When the total contact ratio ε_γ of helical gears exceeds two, which is normally the case, the short period components which represents profile irregularities are smoothed to some extent because in general, simultaneous contact takes place on two or more tooth pairs.

Diagrams in Fig. 31 with the two cases"A" (generated from helical gears) and "B" (from spur gears) illustrate the difference between the ways in which the influence of the overlapping teeth of the two types combine.

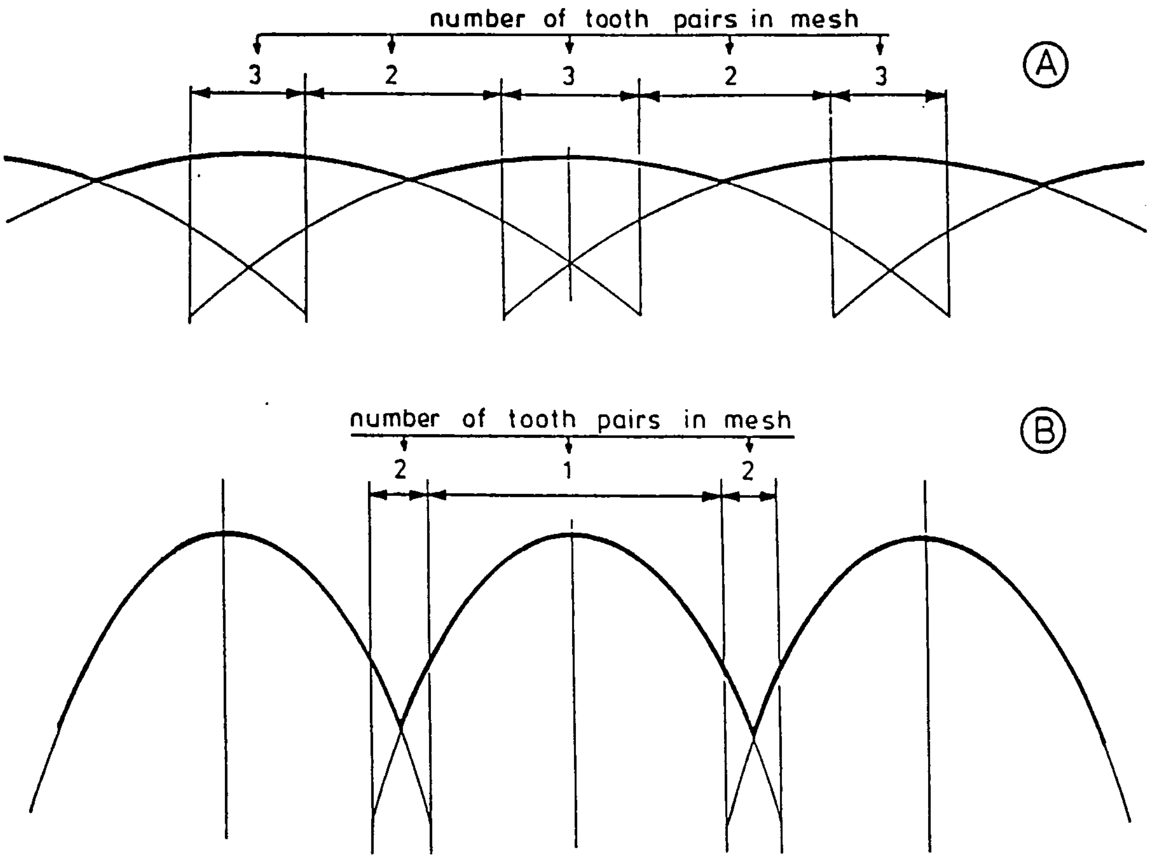


Fig. 31: Influence of overlap ratio

It is important that caution is excercised when assessing the results of tangential composite deviation checks, because these can be very different from expected results such as those derived from consideration of theoretical contact ratio and an assumption that contact is perfect over the tooth profiles and facewidth of helical gears.

Tangential composite deviation can be influenced by modification of tooth profile and of helix (tip relief, crowning etc.) introduced to accomodate possible deformations of shafts, housings and teeth under load.

If under full load the tooth bearing is uniformly distributed over the working surfaces of the teeth, such is not likely to be the case under the light load conditions of tangential composite deviation checks, when the tooth bearing may be localised. Given such circumstances, the contact ratio during the checking operation is much less than elementary theory would suggest.

9.3 Examples of application

9.3.1 Identification and location of defects

The checking of tangential composite deviations facilitates the identification and location of inaccuracies which may degrade the quality of a transmission. For example, as indicated in the diagram in Fig. 32, the presence of a defective tooth can readily be seen. Furthermore it is sometimes possible to carry out corrective measures in situ, in which case the effectiveness of the adjustments can be verified without delay.

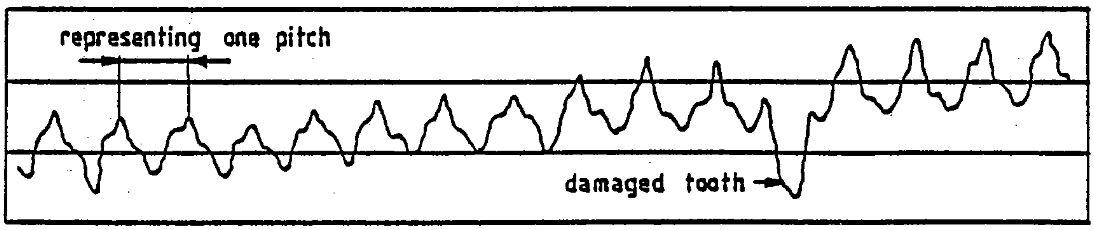


Fig. 32: Part of tangential composite deviation diagram.
Interpretation example

9.3.2 Selective meshing of gears

In some exceptional cases, involving gears with equal numbers of teeth or other integer ratios and which also are not required to be interchangeable, special steps can be taken to ensure that optimum performance is realised. Such gears can be meshed to best advantage by remeshing the gears with a phase shift of ninety degrees in order first of all to find the quadrant in which tangential composite deviations are smallest. Following this the process is repeated by remeshing the gears with phase shifts less than ninety degrees in order to find the optimum meshing phase.

In Fig. 33 diagrams are shown which were generated from a pair of gears at the different phases of mesh position indicated.

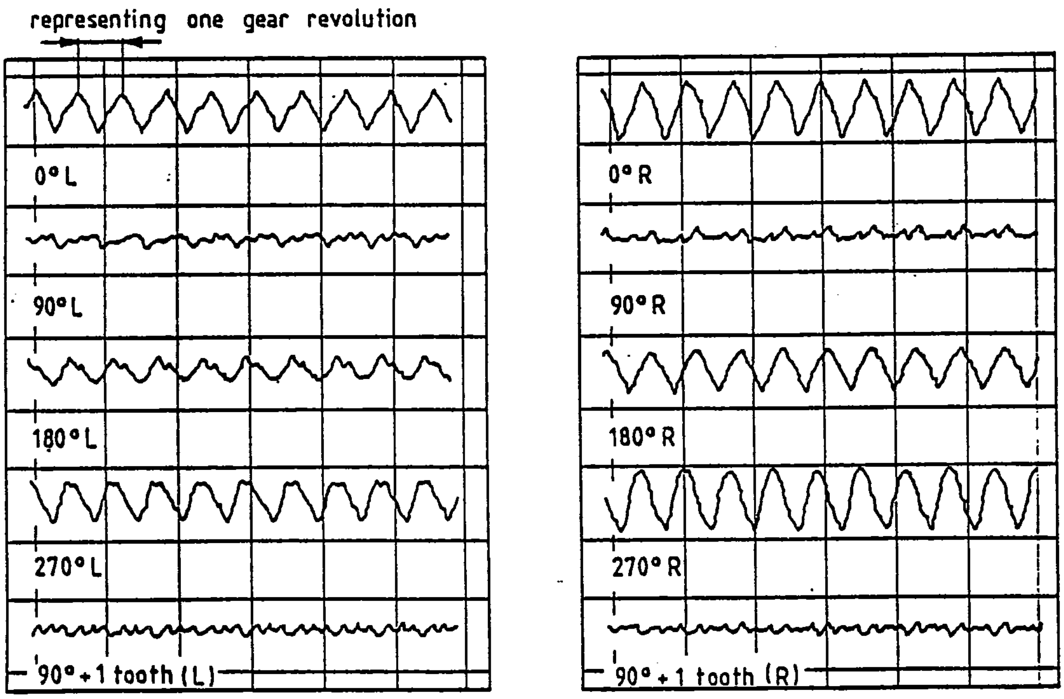


Fig. 33: Tangential composite deviation diagrams showing influence of mesh relocation

It is quite evident that the tangential composite deviation diagrams for the left flanks and right flanks are not the same; it may be necessary to choose an intermediate meshing position which provides the best compromise solution when a high degree of transmission accuracy is needed for both directions of rotation.

9.3.3 Interpretation of tangential composite deviation data

Information about interpretation of data presented in tangential composite deviation diagrams is provided in clause 9.2. When a tangential composite deviation diagram is generated using a master gear, only one revolution of the product gear is needed. If on the contrary two product gears are meshed, several revolutions of both may be needed to generate an adequate tangential composite deviation diagram.

Instrumentation which processes the data so as to separate and record the long and short period components of tangential composite deviation makes the identification and location of significant values relatively easy.

It is important to note that the tooth-to-tooth tangential composite deviation f_i' (Fig. 34A) is substantially attenuated when the long period component is filtered out. Thus the true, maximum deviation f_i' is not necessarily represented in the filtered, short period component curve (Fig. 34C).

9.3.3.1 Analysis of product gear / master gear test data

The total tangential composite deviation F_i' and the maximum tooth-to-tooth deviation f_i' can readily be identified from the full tangential composite deviation diagram. However, for identification of the maximum value of the long period component f_l' and the significant short period component f_s', it is convenient to process data signals through a filter system, using low pass filters to extract the former and high pass filters to extract the latter of those components.

Fig. 34A shows the diagram of an unfiltered tangential composite deviation signal, Fig. 34B and Fig. 34C show diagrams of the long period and the short period components of that signal when processed as described above.

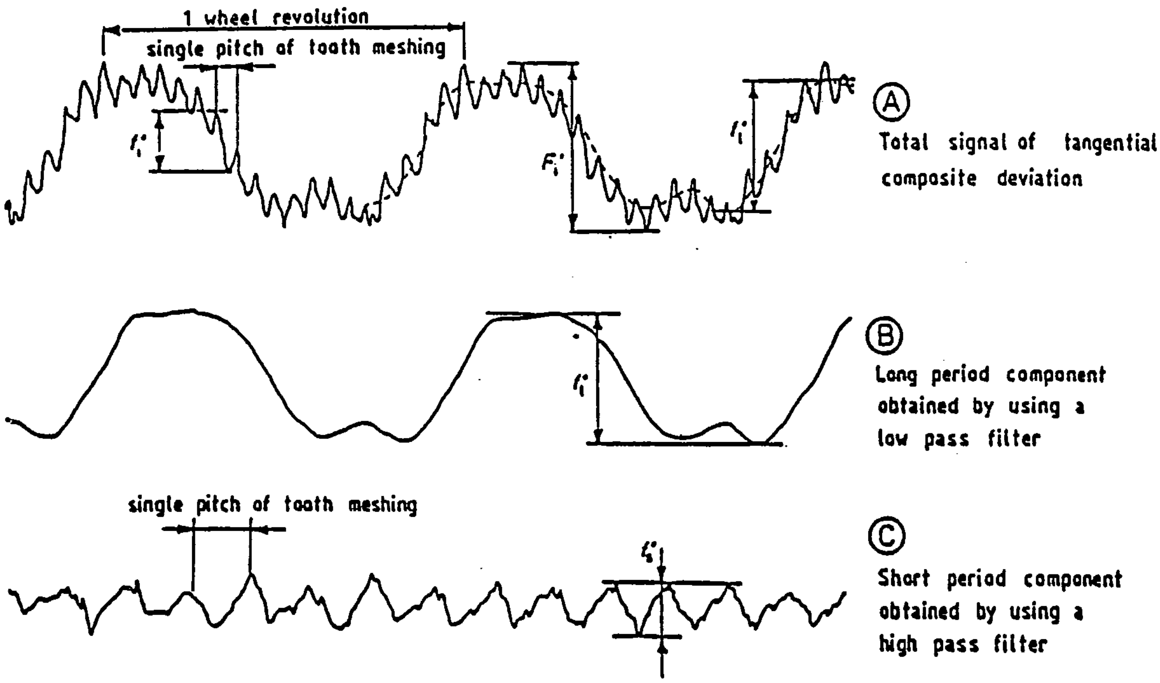


Fig. 34: Analysis of a tangential composite deviation diagram on long period and short period deviation components

9.3.3.2 Analysis of product gear pairs test data

Tangential composite deviation diagrams generated by meshing product gear pairs usually show a succession of periodic deviations corresponding to the cycles of tooth engagement as well as to the periods of rotation of the pinion and of the wheel.

By processing the full generated output signal of tangential composite deviations such as that represented in the diagram Fig. 35C, using very carefully chosen high pass, low pass and band pass filters, the components of the signal are separated.

The long period component generated by the pinion is represented in Fig. 35B. The short period component of tangential composite deviation is represented in Fig. 35A.

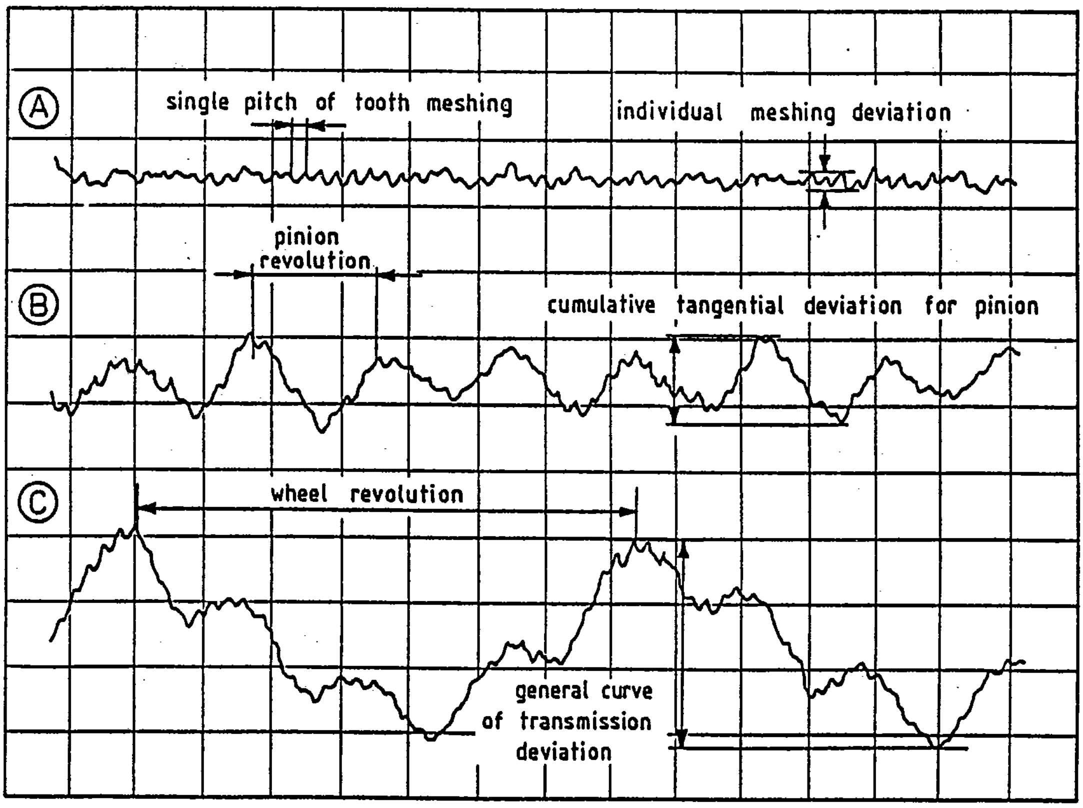


Fig. 35: Analysis of tangential composite deviation diagram. Result of signal filtration

9.3.3.3 Data analysis by the Fast Fourier Transform (FFT) method

Signals can be processed by connecting the output from the measuring equipment directly to a suitable frequency analyser for FFT analysis.

Diagrams in Fig. 36 represent a full tangential composite deviation diagram together with the results of a FFT analysis.

This form of analysis is very effective in that a great deal of information on the various defects of pinion and wheel is thereby provided, including long period and short period components of tangential composite deviation.

In order to execute a Fourier analysis under conditions which are likely to provide adequate and accurate results, it is necessary to provide a signal covering an integer number of revolutions of both gears.

The significant components of the FFT analysis shown in Fig. 36 are indicated against their harmonic numbers "n", relative to the rotational frequency of the larger gear. It must be borne in mind that gear noise and vibration spectra, may include significant components at one or more of the sub- and higher harmonics of the tooth meshing frequency.

In this example, the signal is covering 8 revolutions of the wheel which has 35 teeth, thus the total number of tooth engagement cycles is equal to 280.

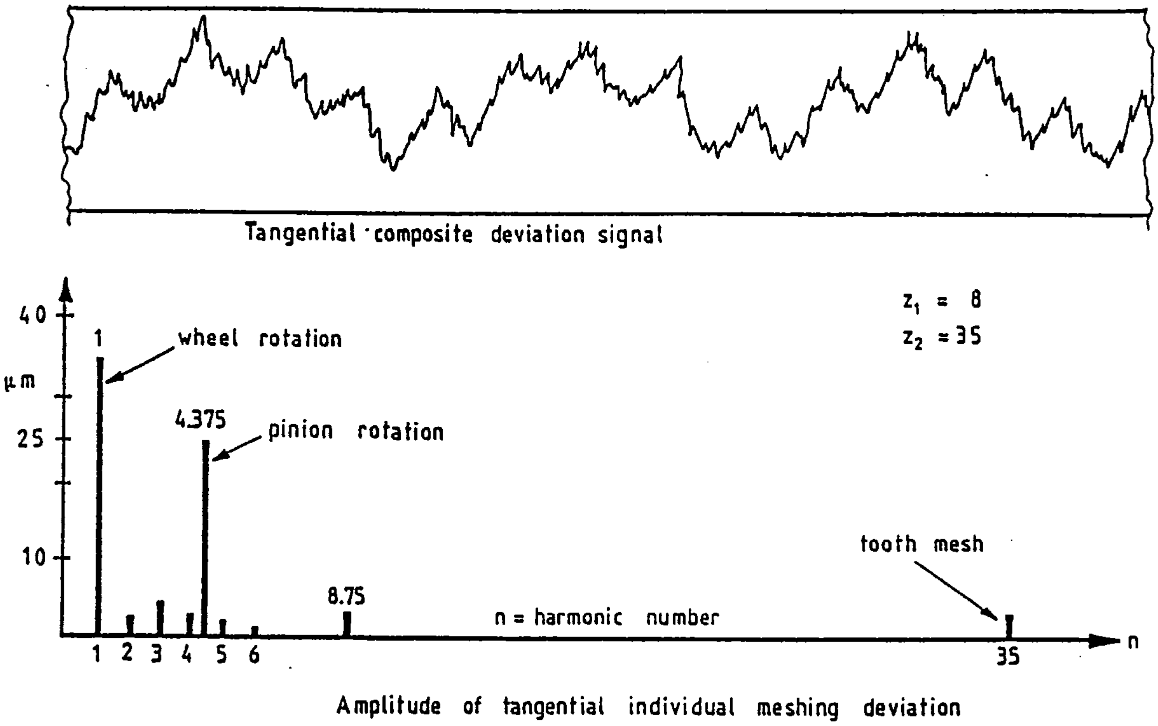


Fig. 36: Result of a Fourier analysis of a tangential composite deviation check performed on a pair of product gears

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Descriptors: gears, cylindrical gears, inspection by measurements, dimensional measurements, dimensional deviations.

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