

## Tolerances for Cylindrical Gear Teeth

Bases

**DIN**  
**3961**

Toleranzen für Stirnradverzahnungen; Grundlagen

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**1 General****1.1 Symbols and designations**

$a$	Centre distance	A, B, C	Test groups
$a_d$	Reference centre distance	$C_b$	Tooth trace relief with longitudinal crowning
$a''$	Two-flank working distance	$F_t$	Total profile deviation
$b$	Facewidth	$F'_i$	Single-flank working deviation
$d$	Reference diameter	$F''_i$	Two-flank working deviation
$f_f$	Profile form deviation	$F_p$	Total pitch deviation
$f'_i$	Single-flank working error	$F_{pk}$	Pitch-span deviation over $k$ pitches
$f''_i$	Two-flank working error	$F_{pz/8}$	Pitch-span deviation over $1/8$ of gear periphery
$f_p$	Individual pitch deviation	$F_r$	Concentricity deviation
$f_{pe}$	Normal base pitch deviation	$F_E$	Generator total deviation
$f_u$	Pitch error	$F_\beta$	Tooth trace total deviation
$f_{H\alpha}$	Profile angle deviation	G, L, N, T	Function groups
$f_{H\beta}$	Tooth trace angle deviation	$L$	Arc length on reference circle
$f_{\beta t}$	Tooth trace form deviation	$R_s$	Tooth thickness fluctuation
$f_{\beta w}$	Tooth trace waviness	$R_t$	Peak-to-valley height
$m$	Module	$R_{Md}$	Fluctuation of diametral two-ball or two-roller dimension
$m_n$	Normal module	$R_w$	Base tangent length fluctuation
$m_t$	Transverse module	TRA	Bearing surface
$s_n$	Normal tooth thickness on the reference cylinder	$T_s$	Tooth thickness tolerance
$s_t$	Transverse tooth thickness on the reference cylinder	$\alpha$	Pressure angle
$z$	Number of teeth	$\alpha_n$	Normal pressure angle
$z_1$	Number of teeth of pinion	$\alpha_t$	Transverse pressure angle
$z_2$	Number of teeth of gear	$\alpha_{wt}$	Working pressure angle
$A^*$	Allowance factor	$\alpha''$	Pressure angle with two-flank working test with master gear
		$\beta$	Helix angle
		$\varphi$	Tolerance increment from gear tooth quality to gear tooth quality

Continued on page 2 to 10  
 Explanations on pages 10 to 12

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## 1.2 Gear tooth tolerances, system of gear fits

To make sure at all times that, apart from the interchangeability of the gears of gear transmission units which is governed by the mating dimensions, the respective functional requirements such as quiet running, true-angle transmission, lubrication facilities, loading capability, are upheld, it is essential for the deviations of certain parameters of the gear teeth as well as the mounting dimensions in the gear housing to be held within certain limits.

The system of gear fits resembles the system for cylindrical fits. For each of the gears of a gear pair all form deviations, pitch deviations etc. lie within the dimensions of two notional, error-free, concentric gears, the tooth centres of which do not need to coincide with those of the working gears, and the tooth thicknesses of which differ from the nominal dimension by the amount of the upper allowance in the case of one of the gears, and by the amount of the lower allowance in the case of the other gear. The backlash results not only from appropriately chosen allowances of tooth thickness, but also from the allowance of the centre distance and other individual deviations. The magnitude and scaling of the tooth thickness allowances are thus in a given ratio to the magnitude and scaling of the centre distance allowances.

Just as with ISO cylindrical fits there are two systems of fits, the hole basis and the shaft basis, so also are there two systems of gear fits, namely the centre distance basis and the tooth thickness basis. With the former, only one tolerance zone of the centre distance is used and the magnitude of the clearance is established by different tolerance zones of the tooth thickness. With the tooth thickness basis system the opposite applies.

The sizes of the gear tooth tolerances are developed on the same principles as the tolerances for cylindrical fits. Thus, just as the latter are designed for accuracies which exceeded the requirements existing at the time of their origin, and are still in advance of their time, so also are the gear tooth tolerances arranged such that in all probability more exacting requirements and improved measuring methods available in the future are catered for.

DIN 3962 Part 1 to Part 3 contain the tolerances for deviations of individual parameters of cylindrical gear teeth.

DIN 3963 contains the tolerances for working deviations.

DIN 3964 contains the centre distance allowances and shaft position tolerances for housings for cylindrical gear transmissions.

DIN 3967 contains the system of gear fits with data on backlash, tooth thickness allowances and tooth thickness tolerances.

The basis specified for the system of gear fits is that of the centre distance. For this purpose it is necessary so to arrange the centre distance allowances that when using gear transmissions with external and internal teeth the centre distance allowances are indicated in the same way, and in gear transmissions with many pairs of gears no unacceptably large deviations from the nominal dimension arise in the distance from the first to the last transmission shaft. The limiting allowances are thus disposed symmetrically relative to the zero line. The kind of gear tooth fit is determined through the choice of

the upper allowance and the tolerance on tooth thickness in DIN 3967 and through the choice of zone for the centre distance allowances in DIN 3964.

## 1.3 Scope

DIN 3962 Part 1 to Part 3 and DIN 3963 contain tolerances for gear teeth on cylindrical gears of module 1 to 70 mm and with reference circle diameters  $d$  up to 10 000 mm, graded in 12 gear tooth qualities. The tolerances apply regardless of the pressure angle. For checking tooth thickness and determining backlash, the working pressure angles and helix angles have to be taken into account, see DIN 3967. For  $d$  exceeding 10 000 mm tolerances can be agreed which can be calculated by using the formulae given in Section 3.5.1 and in Section 3.5.2.

In the case of racks, the tolerances for their teeth shall not exceed those for the teeth of the mating gear. If the manufacturer has no knowledge of the mating gear, he may make the rack length equal to the mating gear circumference.

## 2 Other relevant Standards

- DIN 3960 Definitions and parameters for cylindrical gears and cylindrical gear pairs with involute teeth
- DIN 3962 Part 1 Tolerances for cylindrical gear teeth; tolerances for deviations of individual parameters
- DIN 3962 Part 2 Tolerances for cylindrical gear teeth; tolerances for tooth trace deviations
- DIN 3962 Part 3 Tolerances for cylindrical gear teeth; tolerances for pitch-span deviations
- DIN 3963 Tolerances for cylindrical gear teeth; tolerances for working deviations
- DIN 3964 Centre distance allowances and shaft position tolerances of housings for cylindrical gear transmissions
- DIN 3967 System of gear fits, backlash, tooth thickness allowances and tooth thickness tolerances; bases, calculation of tooth thickness allowances, conversion of allowances for the different measuring methods

## 3 Structure of the tolerance system

### 3.1 Tolerance relationships

The DIN gear tooth tolerance system has been derived from long experience in gear production. The aim has been as far as possible so to coordinate the various tolerances with one another that in a given gear tooth quality they correspond approximately to the average relationships in which the deviations occur in production. The bases of the tolerance system are thus production-oriented; the way in which this system can also be used for tolerancing to suit different functional requirements is described in Section 4.

The numerical equations for the tolerances and the values calculated from them in DIN 3962 Part 1, DIN 3963, DIN 3964 and DIN 3967 have also been extrapolated to ranges in which they can no longer be meaningfully applied in all cases; for example, in gear tooth quality 12 it is only very rarely that there will be any point in making

a pitch test; the same also applies to the working test for gear tooth qualities 2 or 3.

These extrapolated values in the Standards therefore do not imply that, for example, their direct verification can be expected, see Section 5. They only make sure that the tolerances are in a reasonable relationship to the "normal" range if they should happen to be needed at some time; see also Section 6.3.

### 3.2 Individual parameters

DIN 3962 Part 1 to Part 3 contain tolerances for:

Profile form deviation	$f_f$
Profile angle deviation	$f_{H\alpha}$
Total profile deviation	$F_f$
Individual pitch deviation	$f_p$
Normal base pitch deviation	$f_{pe}$
Pitch error	$f_u$
Total pitch deviation	$F_p$
Pitch-span deviation over $k$ pitches	$F_{pk}$
Pitch-span deviation over $1/8$ of gear periphery ( $k = \frac{z}{8}$ )	$F_{p z/8}$
Concentricity deviation	$F_r$
Tooth thickness fluctuation	$R_s$
Tooth trace total deviation	$F_\beta$
Tooth trace angle deviation	$f_{H\beta}$
Tooth trace form deviation	$f_{\beta f}$

For definitions see DIN 3960.

### 3.3 Working deviations

In DIN 3963 tolerances are presented for the following working deviations:

Two-flank working deviation	$F_i''$
Two-flank working error	$f_i''$
Single-flank working deviation	$F_i'$
Single-flank working error	$f_i'$

For application of the working test see also Code VDI/VDE 2608.

For definitions see DIN 3960.

### 3.4 Gear tooth qualities

The DIN gear tooth tolerance system contains 12 gear tooth qualities. The finer qualities are intended for master gears and special requirements. They are so graduated that the tolerances of each quality amount to  $\varphi$  times the tolerances of the preceding quality. The increments  $\varphi$  are stated in Section 3.5.

The limiting values of the reference diameter and module ranges are taken from the preferred number series R 20.

### 3.5 Calculation of tolerances

The tolerances are calculated according to formulae exhibiting different kinds of dependence on the gear tooth data (reference circle diameter, module, number of teeth and facewidth). The numerical factors of the formulae reproduced below apply to the 5th gear tooth quality.

For calculating the tolerances from the formulae, the geometrical mean values of the ranges stated in the Tables are used for  $m_n$  and  $d$ .

The tolerances calculated in this way are rounded to preferred numbers. Hence the total number of values is comparatively small. In addition, knowledge of the preferred numbers can be used to calculate the next larger or smaller value from any known value.

$m, d, b, L$  in mm. Tolerances in  $\mu\text{m}$ .

#### 3.5.1 Individual deviations

a) Profile form deviation

$$f_f = 1,5 + 0,25 (m_n + 9 \cdot \sqrt{m_n})$$

$\varphi = 1.4$  up to gear tooth quality 9  
 $\varphi = 1.6$  from gear tooth quality 9

b) Profile angle deviation

$$f_{H\alpha} = 2,5 + 0,25 (m_n + 3 \cdot \sqrt{m_n})$$

$\varphi = 1.4$  up to gear tooth quality 9  
 $\varphi = 1.6$  from gear tooth quality 9

c) Total profile deviation

$$F_f = \sqrt{f_{H\alpha}^2 + f_f^2}$$

$\varphi = 1.4$  up to gear tooth quality 9  
 $\varphi = 1.6$  from gear tooth quality 9

d) Individual pitch deviation and normal base pitch deviation

$$f_p = f_{pe} = 4 + 0,315 (m_n + 0,25 \cdot \sqrt{d})$$

$\varphi = 1.4$  up to gear tooth quality 9  
 $\varphi = 1.6$  from gear tooth quality 9

e) Pitch error

$$f_u = 5 + 0,4 (m_n + 0,25 \cdot \sqrt{d})$$

$\varphi = 1.4$  up to gear tooth quality 9  
 $\varphi = 1.6$  from gear tooth quality 9

f) Total pitch deviation

$$F_p = 7,25 \cdot \frac{d^{1/3}}{z^{1/7}}$$

$\varphi = 1.4$  up to gear tooth quality 9  
 $\varphi = 1.6$  from gear tooth quality 9

g) Pitch-span deviation

$$F_{pk} = 6,25 \cdot m_n^{1/7} \cdot \frac{L^{1/3}}{d^{1/7}}$$

$\varphi = 1.4$  up to gear tooth quality 9  
 $\varphi = 1.6$  from gear tooth quality 9

h) Concentricity deviation

$$F_r = 1,68 + 2,18 \cdot \sqrt{m_n} + (2,3 + 1,2 \log m_n) \cdot d^{1/4}$$

$\varphi = 1,4$

i) Tooth thickness fluctuation

$$R_s = 1 + 1,28 \sqrt{m_n} + (1,33 + 0,7 \log m_n) \cdot d^{1/4}$$

$\varphi = 1,4$

k) Tooth trace total deviation

$$F_\beta = 0,8 \cdot \sqrt{b} + 4$$

$\varphi = 1.25$  up to gear tooth quality 6  
 $\varphi = 1.4$  gear tooth quality 6 to 8  
 $\varphi = 1.6$  gear tooth quality 8 to 12

l) Tooth trace angle deviation

$$f_{H\beta} = 4,16 \cdot b^{0,14}$$

$\varphi = 1.32$  up to gear tooth quality 6  
 $\varphi = 1.4$  gear tooth quality 6 to 8  
 $\varphi = 1.55$  gear tooth quality 8 to 12

m) Tooth trace form deviation

$$f_{\beta t} = \sqrt{F_{\beta}^2 - f_{H\beta}^2}$$

### 3.5.2 Working deviations

a) Two-flank working deviation

$$F_i'' = 2 + 2,57 \cdot \sqrt{m_n} + (3,12 + 0,432 \log m_n) \cdot d^{1/4}$$

$$\varphi = 1,4$$

b) Two-flank working error

$$f_i'' = 1,8 \cdot \sqrt{m_n} + 1,6 \cdot d^{1/4} - 1$$

$$\varphi = 1,4$$

c) Single-flank working deviation

$$F_i' = 0,8 \cdot (F_p + F_t)$$

$$\varphi = 1,4 \text{ up to gear tooth quality 9}$$

$$\varphi = 1,6 \text{ from gear tooth quality 9}$$

d) Single-flank working error

$$f_i' = 0,7 \cdot (f_p + F_t)$$

$$\varphi = 1,4 \text{ up to gear tooth quality 9}$$

$$\varphi = 1,6 \text{ from gear tooth quality 9}$$

### 3.6 Tolerances for the diameter of the tip circle (tip cylinder)

Because of the module-dependent bottom clearance and the effect of the tip circle diameter on the transverse contact ratio of the gear teeth, the tolerance  $\pm 0,05 \cdot m$  is prescribed for the diameter of the tip cylinder in the case of non-overcut tooth tips, except where agreed to the contrary.

In the case of overcut tooth tips, which are mainly used on gears for light precision engineering, the diameter of the tip cylinder is employed for indirect checking of the tooth thickness. It is smaller than the nominal dimension by 2.747 times the tooth thickness allowance when  $\alpha = 20^\circ$  and by 3.732 times the tooth thickness allowance when  $\alpha = 15^\circ$ . For further details see DIN 58 405 Part 2, May 1972 edition, Section 2.1.

## 4 Tolerancing

### 4.1 Tolerance families

The tolerances chosen must be correct from the functional and production viewpoint. The more exacting the func-

tional requirements made on a gear, the better must be the gear tooth quality produced, but on the other hand the more differentiated must be the tolerancing in order to avoid unnecessary production cost. Hence it is possible for fundamentally different gear tooth qualities to be prescribed for the individual parameters. Such a linking is termed a tolerance family.

However it is not desirable to form tolerance families in which a different gear tooth quality is fixed for each critical parameter. The service properties required of a gear can be classified in the following function groups:

- G: uniformity of transmission of motion
- L: quiet running and dynamic load-carrying capacity
- T: static load-carrying capacity
- N: no information on function

These depend largely – if not exclusively – on specific geometrical requirements, see Table 1. When tolerance families with gear tooth qualities are formed it is therefore desirable to choose the same gear tooth qualities each time for the tolerances of a function group. In this way tolerance families with not more than three different gear tooth qualities can arise. If a parameter occurs in several function groups, it is the group with the finer gear tooth quality which prevails for this parameter.

### 4.2 Application of tolerance families

It will often happen that for a particular gear it is desirable or practicable to demand only a general service property without taking special functional requirements into account (function group N).

In these cases only a single gear tooth quality is prescribed, e. g. quality 8. (The points to be observed and checked in these and the following cases are described in Section 5). In other cases fulfilment of the requirements of a specific function group will be stipulated. In such cases the letter symbol of the function group must be quoted along with the required gear tooth quality. Examples are given in Table 2.

Basically, tolerances shall be agreed only for such function groups whose deviations really matter for the application concerned. As a rule, not more than two function groups should be agreed.

Table 1. Function groups of deviations

Function group		Important deviations *)
G	Uniformity of transmission of motion	$F_i' f_i F_p F_i'' F_t f_i''$
L	Quiet running and dynamic load-carrying capacity	$f_i' f_p (f_{pe}) f_i'' F_t f_{H\beta} F_p (F_t)$
T	Static load-carrying capacity	$f_{pe} f_{H\beta} TRA$
N	No information on function	$F_i' f_{H\beta} F_t f_i''$

\*) Apart from these deviations there are, of course, also other influencing variables on which the service properties depend, e. g. the quietness of running also depends on speed and loading, the load-carrying capacity on surface quality, the material and its condition. It is therefore quite likely to be necessary to state other requirements, including non-geometrical ones, e. g. regarding particular hardness values or specific sound pressure levels under given operating conditions.

The purpose of this free linking is maximum economy in manufacture and preventing gears from being declared as rejected when they exhibit considerable inaccuracies only in regard to secondary parameters. Where requirements of like kind exist for specific types of gear (e. g. change gears of machine tools, camshaft drive gears for motor vehicle engines), specific tolerance families and codes can be specified for this purpose in appropriate Standards.

Table 2. Examples of tolerance families

Indication	Meaning
G 8, L 7	Parameters for uniformity of transmission of motion (G) in gear tooth quality 8, parameters for quiet running (L) in gear tooth quality 7. Nothing prescribed for the parameters for static load-carrying capacity (T).
L 4	Parameters for quiet running and dynamic load-carrying capacity (L) in gear tooth quality 4. Nothing prescribed for other groups.
G 6, L 6, T 7	Parameters for uniformity of transmission of motion (G) and quiet running (L) each in gear tooth quality 6, static load-carrying capacity (T) in gear tooth quality 7.

## 5 Quality testing and test groups

In a similar manner to tolerancing, the testing must also correspond with the prevailing requirements and conditions. The more exacting the quality stipulation is, the greater is the amount of test effort required for the reliable determination of measured values.

On the other hand, that which is not specified should also not be tested. When tolerancing is by function groups, only the parameters important in this connection should be tested each time. Nor is it absolutely necessary to test all the parameters belonging to the function group, but instead advantage can be taken of the fact that they are in some cases closely related to one another. Thus, for example, a profile form test and a tooth trace test can be replaced approximately by the contact pattern test, or vice versa. Similar considerations apply to concentricity testing on the one hand and the two-flank working test on the other, etc.

A given function group can therefore be tested at will by way of different measurement combinations. A combina-

tion of this kind is termed a test group. In Table 3 three test groups A, B and C are assigned to each function group. The test group in the first column is the one best suited functionally, but requires measuring instruments which are not always available. In the second column is the test group recommended for the normal case, which may possibly test the function by a somewhat less direct method, but by using measuring instruments which are mostly available. The third test group is possibly more complicated in use than the second because some of the criteria are still further resolved or tested by even more indirect methods, so that in some cases the number of measuring operations required increases.

Through the provision of these test groups it is possible to match acceptance tests to the circumstances of different plants in which different measuring methods and instruments are often used.

The test procedures indicated by symbols in Table 3 are explained in Table 4, which also gives information on which of the parameters determined in a test procedure must be within tolerance.

Unless agreed to the contrary, the choice of test group according to Table 3 rests with the manufacturer. If, in this case, the parameters of the test group selected by the manufacturer are within tolerance, then the gear tooth quality is deemed to be complied with, notwithstanding the fact that parameters of other test groups (of the same function group) exceed the tolerances of the stipulated gear tooth quality.

It is also permissible to verify the stipulated gear tooth quality by parameters which according to Table 3 are assigned to a test group of a finer gear tooth quality.

If the customer requires testing in a specific test group, this must be agreed already at the time of ordering.

**Example:** A gear is required in function group G (uniformity of transmission of motion) in the 6th gear tooth quality, and this is to be verified exclusively by testing the parameters according to test group B. This is indicated by stating: gear tooth quality G 6 B. If no letter symbol is stated for the test group (A, B or C), it is left to the manufacturer to select the group.

Apart from these test groups, it may also be necessary or desirable to state detailed test codes. To avoid difficulties in the acceptance of gears in such cases, all details should be agreed as necessary between manufacturer and customer, e. g. which test is to be carried out at acceptance, which instruments (type and make) are to be used for this purpose, whether those of the manufacturer or those of the customer. It is also often expedient to state the working flanks. With regard to compliance with the tolerances, the corresponding Standards apply.

The reference temperature is 20 °C. Plastics gears are to be tested at a relative humidity of 65 %.

Table 3. Test groups for function groups and gear tooth qualities

Function group	G Uniformity of transmission of motion			L Quiet running and dynamic load-carrying capacity			T Static load-carrying capacity			N No information on function		
	A	B	C	A	B	C	A	B	C	A	B	C
Gear tooth quality 1												
2	$F'_i R_s$			$F'_i F_\beta$	$F_p F_t F_\beta$		$F'_i F_\beta$		$f_{pe} TRA$	$F'_i R_s$	$F_p R_s / F_t F_\beta$	$f_{pe} R_s / F_t F_\beta$
3	$F'_i R_s$			$F'_i F_\beta$	$F_p F_t F_\beta$		$F'_i F_\beta$		$f_{pe} TRA$	$F'_i R_s$	$F_p R_s / F_t F_\beta$	$f_{pe} R_s / F_t F_\beta$
4	$F'_i R_s$	$F_p F_t F_\beta$		$F'_i F_\beta$	$F_p F_t F_\beta$	$f_{pe} F_t TRA$	$f_{pe} F_t F_\beta$		$F''_i TRA$	$F'_i R_s$	$F''_i F_t F_\beta$	$f_{pe} R_s / TRA$
5	$F'_i R_s$	$F_p F_t F_\beta$	$f_{pe} F_t F_\beta$	$F'_i F_\beta$	$F_p F_t F_\beta$	$f_{pe} R_s TRA$	$f_{pe} F_t F_\beta$		$F''_i TRA$	$F'_i R_s$	$F''_i F_t F_\beta$	$f_{pe} R_s / TRA$
6	$F'_i R_s$	$F_p F_t F_\beta$	$f_{pe} F_t F_\beta$	$F_p F_t F_\beta$	$F''_i F_t F_\beta$	$f_{pe} R_s TRA$	$f_{pe} F_t F_\beta$		$F''_i TRA$	$F'_i R_s$	$F''_i F_t F_\beta$	$f_{pe} R_s / TRA$
7	$F_p F_t F_\beta$	$f_{pe} F_t F_\beta$		$F_p F_t F_\beta$	$F''_i F_t F_\beta$	$f_{pe} R_s$	$f_{pe} TRA$		$f_p TRA$	$F''_i F_t F_\beta$	$F''_i TRA$	$f_{pe} R_s$
8	$F_p F_t F_\beta$	$f_{pe} F_t F_\beta$		$F_p F_t F_\beta$	$F''_i F_t F_\beta$	$f_{pe} R_s$	$f_{pe} TRA$		$f_p TRA$	$F''_i F_t F_\beta$	$F''_i TRA$	$f_{pe} R_s$
9					$F''_i F_t F_\beta$	$f_{pe} R_s$	$f_{pe} TRA$		$f_p TRA$	$F''_i TRA$	$F''_i$	$f_{pe}$
10					$F''_i F_t F_\beta$	$f_{pe} R_s$	$f_{pe} TRA$		$f_p$	$F''_i TRA$	$F''_i$	$f_{pe}$
11							$f_{pe}$		$f_p$	$F''_i$	$f_{pe}$	$R_s$
12										$F''_i$	$f_{pe}$	$R_s$

If no two-flank working test including the centre distance test is undertaken, a concentricity test and a tooth thickness test (as a chordal, base tangent length or ball/roller measurement) are always necessary in addition to the tests named.

Table 4. Scope of the test procedures denoted by symbols in Table 3

Symbol	Test procedure	Required to be in tolerance
$F'_i$	Single-flank working test	$F'_i$ and $f'_i$
$f'_i$	Single-flank working test	$f'_i$
$F''_i$	Two-flank working test	$F''_i$ , $f''_i$ , $a''$ (if $T_s$ is not tested)
$R_s$	Test of tooth thickness fluctuation	$R_s$
$TRA$	Contact pattern test	$TRA$
$F_\beta$	Tooth trace test	$F_\beta$ , $f_{H\beta}$ , $f_{\beta t}$
$F_t$	Profile test	$F_t$ , $f_{H\alpha}$ , $f_t$
$F_p$	Cumulative pitch test	$F_p$ , $F_{p z/8}$
$f_p$	Individual pitch test	$f_p$ , $f_u$
$f_{pe}$	Normal base pitch test	$f_{pe}$

## 6 Additional provisions and conditions

### 6.1 Profile form

In some cases it is necessary qualitatively to alter or even restrict the latitude allowed to the profile form deviation on the basis of the gear tooth quality, as for example in Fig. 1. The variation of the profile form must lie within the fields indicated. If a profile pattern of this kind is required additionally to the gear tooth quality, this shall be separately agreed. At the same time it must also be borne in mind that the single-flank and two-flank working error and the normal base pitch deviation may automatically become larger than with the true involute form, see VDI/VDE 2608 and 2613.

### 6.2 Tooth trace

The tolerances for  $F_\beta$ ,  $f_{H\beta}$  and  $f_{\beta t}$  stated in formula or numerical terms stand in a well-balanced relationship to the other tolerances from considerations of interchangeable manufacture. Where relatively large facewidths are involved, however (in some cases even when  $b > 50$  mm) these tolerances are sometimes too large, particularly for  $f_{H\beta}$  and  $F_\beta$ , when it is desired to achieve a good contact pattern. In such cases the tooth trace tolerances need to be made smaller than the other tolerances (see Sections 4.1 and 4.2) or else interchangeable manufacture has to

be abandoned in favour of pairwise gear matching. The difference ( $f_{H\beta 1} - f_{H\beta 2}$ ) in this case should be smaller or at most equal to the tolerance value of the Standard. In particular, the flanks should always be set back at the ends.

No tolerances for the tooth trace waviness  $f_{\beta w}$  are laid down in the Standards. For special applications, Standards are applied in which tolerances for  $f_{\beta w}$ , e. g. for turbine gear transmissions BS 1807 and BS 1488. For the generator total deviation  $F_E$  (see DIN 3960) it is generally possible to accept the same values as for  $f_{H\beta}$ .

In similar fashion to Fig. 1 for the profile form, so also for the tooth trace can a specific configuration, termed crowning, also be prescribed.

Fig. 2 shows how, with prescribed crowning  $C_b$  taken into account, the values for  $F_\beta$ ,  $f_{H\beta}$  and  $f_{\beta t}$  are derived from the tooth trace test pattern. In the evaluation of the test pattern the width has to be reduced by 10% at each tooth end, assuming a minus deviation.

Crowning imparts an improvement in functional terms, i. e. the effect of the tooth trace angle deviation  $f_{H\beta}$  is partly compensated. This means that in certain circumstances it is possible with crowned gear teeth to select a coarser gear tooth quality than with non-crowned gears, see VDI/VDE 2612.

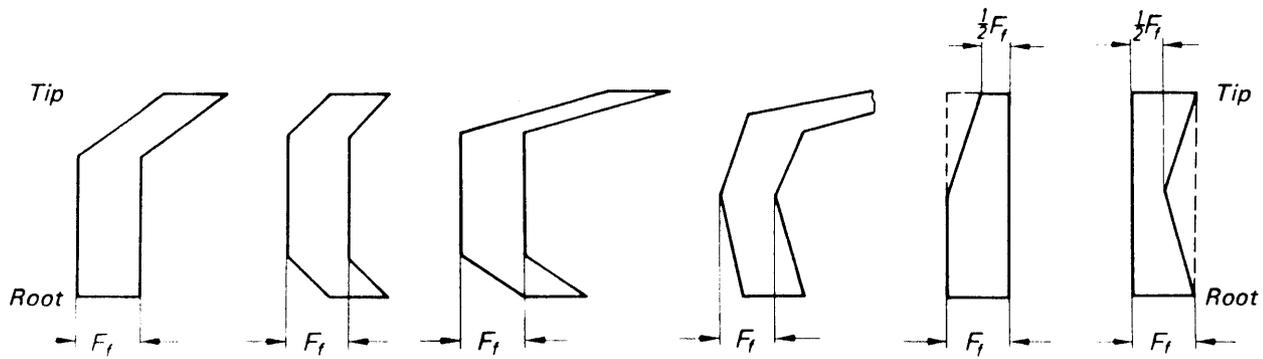


Figure 1. Altered pattern of permissible range of profile deviations (examples)

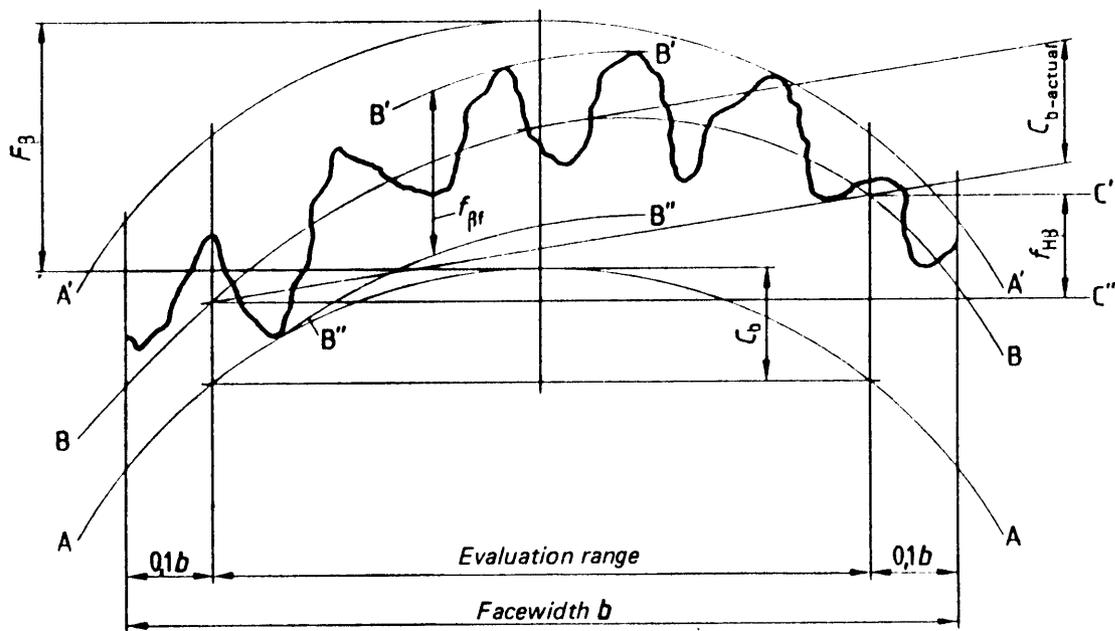


Figure 2. Tooth trace test pattern for narrowed tooth trace test range

- AA, A'A' Corrected base lines bounding the test pattern within the evaluation range.
- BB Averaging actual tooth flank
- B'B', B''B'' Actual tooth traces bounding the test pattern within the evaluation range.
- C'C', C''C'' Non-corrected nominal tooth traces through the points of intersection of the averaging actual tooth trace BB with the boundary lines of the evaluation range.
- $F_\beta$  Tooth trace total deviation; results from the distance apart of the corrected base lines AA and A'A' as measured at right angles to the chart feed.
- $f_{H\beta}$  Tooth trace angle deviation; results from the distance apart of the nominal tooth traces C'C' and C''C''.
- $f_{\beta t}$  Tooth trace form deviation; results from the distance apart of the actual tooth traces B'B' and B''B'' as measured at right angles to the chart feed.
- $C_b$  Nominal crowning
- $C_{b\text{ actual}}$  Actual crowning

**6.3 Working test for fine gear tooth qualities**

The permissible working deviations apply in principle to the testing of a working gear with an error-free standard (master gear). In practice it is necessary to limit the checking to determining that the deviations of the master gear compared with those of the test gear are sufficiently small (approximately in a 1 : 5 ratio). Hence it would not be meaningful to state permissible working deviations for the gear tooth quality range from 1 to 4, since in practice there are no adequately error-free standard gears available for this purpose, or gears with known error pattern. Nevertheless, a working test of two gears of approximately the same gear tooth quality is sometimes carried out. In this case, however, the test must be repeated with the flanks interchanged (left and right). If there deviations are  $F_{11}$  and  $F_{12}$  a measured value of

$$F_1 = \sqrt{F_{11}^2 + F_{12}^2}$$

will be expected according to the quadratic error propagation law. In a working test of two gears (master gears) having approximately the same gear tooth quality, therefore, only  $\sqrt{2} = 1.4$  times the tolerance value of this quality can be accepted. Hence for these applications the standardized quoting of permissible working deviations is still meaningful for fine gear tooth qualities.

A binding quality test according to this method is however only possible if the uncertainty of measurement can be kept small enough, cf. VDI/VDE 2608.

**6.4 Pitch-span deviations over discretionary pitch-spans**

In general the tolerancing and testing of pitch-spans over  $45^\circ$  and  $180^\circ$  of the gear circumference according to the values  $F_{pz/8}$  and  $F_p$  in DIN 3962 Part 1 is perfectly adequate. In some cases, however, and particularly where high-speed gears are involved, it may be necessary in certain circumstances to further restrict the pitch-span

deviations over discretionary pitch-spans. Their values are found from the formula or the diagram in DIN 3962 Part 3 and they require separate agreement.

A pitch-span measurement can be evaluated by plotting the positive and negative tolerance values on tracing paper versus the number of teeth  $z$ . The scales of  $F_p$  and  $z$  must agree with those of the test chart, and in this the  $z$  axis must be at right angles to the  $F_{pk}$  axis. If the zero point of the diagram is moved along the pitch-span curve, keeping the  $z$  axes parallel, the pitch-span curve must lie at all times within the two tolerance curves, Fig. 3.

Another method is to determine the pitch-span deviation  $F_{pk}$  and the associated span at the point of maximum increase in the test chart, Fig. 3 right. This measured value of  $F_p$  is then compared with the value permitted according to the diagram in DIN 3962 Part 3 for the same span.

**6.5 Individual pitch deviations and pitch errors**

If mavericks occur in individual pitch measurements, it is permissible to make measurements at different points on the flanks and to derive averages for each pitch. The resulting series of measurements is then used as the basis for assessing the individual pitch deviation and pitch error. No further mavericks are permissible in this.

When making pitch measurements with two spherical tracers it is important to ensure that the second tracer contacts the flanks each time at exactly the same spot as the first tracer in the preceding pitch.

Apart from the pitch deviation tolerances laid down in DIN 3962 Part 1 and Part 3 there are also other characteristics for assessing pitch, e. g. the arithmetic mean or root mean square values of the pitch deviations. They may form the subject of separate agreements; see also VDI/VDE 2605.

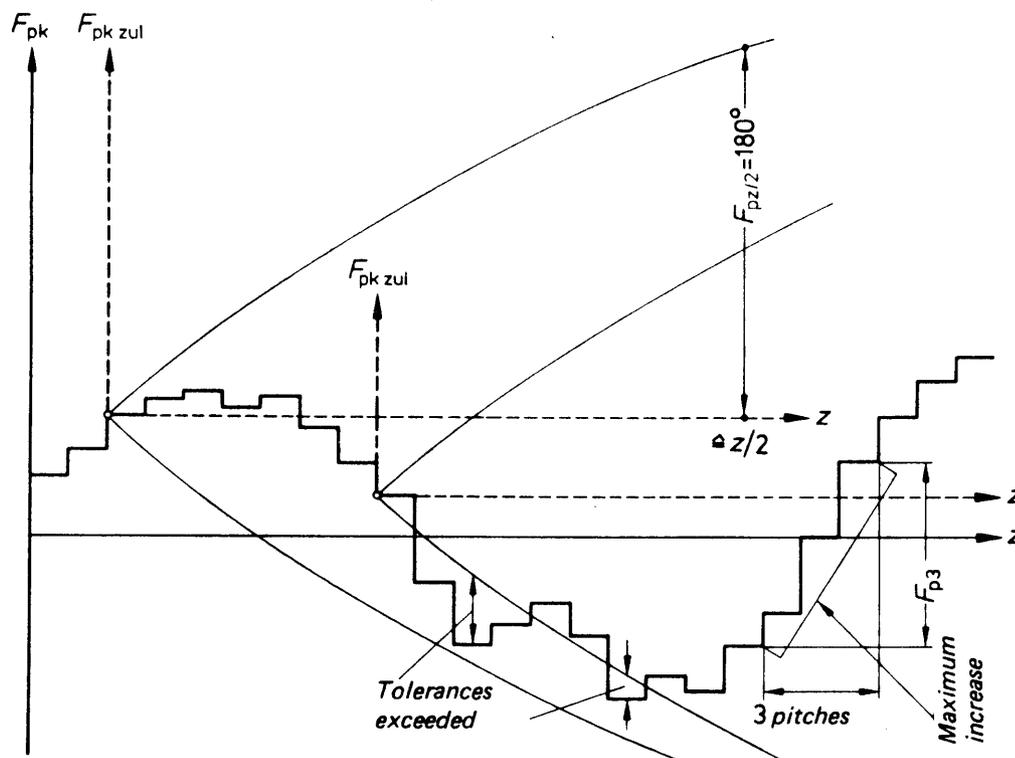


Figure 3. Evaluation of pitch-span measurement

*Other Standards and codes*

DIN 3999	Symbols for gear teeth
VDI/VDE 2605	Circular pitches and plane angles. Fundamental terms for angle dimensions, angle measurements, angle standards and their errors
VDI/VDE 2608	Single-flank and two-flank working test for straight and helical cylindrical gears with involute profile
VDI/VDE 2612	Profile and tooth trace test for cylindrical gears with involute profile, Part 1 Profile testing, Part 2 Tooth trace testing
VDI/VDE 2613	Pitch and concentricity tests on gear teeth
VDI/VDE 3336	Cutting cylindrical gears with involute profile, metal-removal processes

**Literature**

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Seifried, A.: Zahnrad-Teilungsfehler. Statistische Ermittlung der Zusammenhänge mit anderen Verzahnungsfehlern (Gear tooth pitch errors. Statistical determination of relationships with other gear tooth errors). VDI-Berichte No. 105 (1967) pp. 141–143.

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Kagerl, H. G.; Dulich, W.: Fertigungsgerechte Relationen Einflankenwälzabweichung/Einflankenwälzsprung – Einzelabweichungen (Production-oriented relationships, single-flank working deviation/single-flank working error – individual deviations). FERTIGUNGSTECHNIK UND BETRIEB 21 (1971) No. 8, pp. 455–458.

*Explanations*

This Standard is intended to contribute to the further development of the International Standard on tolerances for cylindrical gears with involute teeth ISO 1328 – 1975. It is recommended that comparative data derived from experience be collected for making available in due course to the responsible ISO committee.

The DIN gear tooth tolerance system in use so far, the principles of which have not altered, has for the most part proved itself in practice. Only with regard to a few points was a general redefining of tolerances necessary. This applies particularly to the permitted values for single-flank working deviations, tooth trace deviations and, to some extent, for pitch-span deviations.

Apart from this, it has been shown in practice that for the same module the profile deviations do not become larger with increasing diameter (increasing number of teeth). The profile tolerances therefore remain only module-dependent and no longer diameter-dependent.

The following survey in tabular form gives a general view of the alterations undertaken.

The definitions, designations and symbols have been redefined in accordance with DIN 3960, DIN 3998 and DIN 3999. Newly added are tolerances for total profile deviation, pitch-span deviation, tooth trace form deviation and tooth trace total deviation.

Information on tolerance families, quality testing and test groups has also been added.

The numerical specifying of tolerances derives from the equations in this Standard. The alterations are listed in the last column of the survey Table. Additional alterations of the tabulated numerical data in DIN 3962 Part 1 to Part 3 and DIN 3963 result from the fact that the module and diameter ranges have been regraded and from the fact that the values in the old Standards calculated from the tolerance equations had for the most part been rounded upwards, whereas in the new Standards they have been rounded up or down to the nearest preferred number.

The dimension range has been enlarged in the upward direction to module  $m = 70$  mm and diameter  $d = 10\,000$  mm. Regardless of whether all deviations are measurable, the numerical values in the tables of DIN 3962 Part 1 and DIN 3963 have been listed completely in the interests of future applications. The module range  $m < 1$  mm has been deleted, see in this connection DIN 58 405. The arrangement of the tolerance Standards has been altered so that the tooth trace deviations have been grouped with the individual deviations in DIN 3962 Part 2, whilst DIN 3963 contains only working deviations and DIN 3967 only the system of gear fits.

When the "Gear tooth tolerances" Sub-committee started revising the Standards on tolerances for cylindrical gear teeth it was faced with the question whether the tolerances contained in ISO 1328 – 1975 could be taken over complete. After a thorough examination the committee came to the conclusion that this would not be expedient for the purposes of German industry.

The principal objections to ISO 1328 – 1975 are its big tolerances for the two-flank working deviation, concentricity deviation and pitch-span deviation and the large amount of spread in their relationship to the other tolerances.

The Figures below present some examples of tolerances according to the new Standards and compare them with ISO 1328 – 1975 and AGMA 390.03. (For this purpose AGMA gear tooth quality 11 was assumed equal to DIN gear tooth quality 7 and AGMA gear tooth quality 9 equal to DIN gear tooth quality 9). The best agreement is found with the profile and tooth trace total deviations, whilst the biggest discrepancies occur in the two-flank working deviations and concentricity deviations. The latter however only to the middle gear tooth qualities. Because of the unequal increments in ISO, the agreement improves with the high and low gear tooth qualities.

**Survey of alterations in gear tooth tolerances to DIN 3961**  
(in connection with DIN 3960, DIN 3962 Part 1 to Part 3 and DIN 3963)

Symbol and designation		Alteration, redefinition		
New	Old	Conceptual content	Tolerance equations	
$f_t$ Profile form deviation	$f_t$ Flank form error	Unchanged	Now only module-dependent, no longer diameter-dependent at $\frac{d}{m} = z = 25 \dots 100$ approx. same as before	
$f_{H\alpha}$ Profile angle deviation	$F_g$ Base circle error	Unchanged $f_{H\alpha} \triangleq F_g$ in DIN 3962 Part 1 to Part 4, November 1952x editions		
$F_t$ Total profile deviation	—	Redefined according to ISO 1328 – 1975	Only module-dependent!	
$f_p$ Individual pitch deviation	$f_t$ Individual pitch error	Unchanged	Reduced 10 – 20 %	
$f_{pe}$ Normal base pitch deviation	$f_e$ Normal base pitch error	Unchanged	Reduced 10 – 20 %	
$f_u$ Pitch error	$f_u$ Pitch error	Unchanged	$f_u = 1.25 f_p$ instead of previously $f_u = f_p$	
$F_p$ Total pitch deviation	$F_t$ Cumulative pitch error	Made more precise: $F_p \triangleq$ Maximum	Reduced for large diameters!	
$F_{pk}$ Pitch-span deviation	—	Redefined	On basis of ISO 1328 – 1975 dependent on pitch-span	
$F_{p z/8}$ Pitch-span deviation over $\frac{1}{8}$ of periphery	—	Redefined		
$F_r$ Concentricity deviation	$f_r$ Concentricity error	Unchanged	$F_r \approx 0,84 F_i''$ instead of previously $F_r = F_i''$	
$R_s$ Tooth thickness fluctuation	$f_s$ Tooth thickness error	Unchanged	$R_s = 0,59 F_r \approx 0,52 F_i''$ instead of previously $f_s = 0,7 f_r = 0,7 F_i''$	
$F_i''$ Two-flank working deviation	$F_i''$ Tooth flank working error	Unchanged	Unchanged	
$f_i''$ Two-flank working error	$f_i''$ Two-flank working error	Unchanged	Somewhat increased	
$F_i'$ Single-flank working deviation	$F_i'$ Single-flank working error	Unchanged	Considerably increased, with: $F_i' = 0,8 (F_p + F_t)$ From gear tooth quality 9: $\varphi = 1,6$ instead of previously 1.4	
$f_i'$ Single-flank working error	$f_i'$ Single-flank working error	Unchanged	Considerably increased, with: $f_i' = 0,7 (f_p + F_t)$ From gear tooth quality 9: $\varphi = 1,6$ instead of previously 1.4	
$f_{\beta t}$ Tooth trace form deviation	—	Redefined	Coordinated with $f_{H\beta}$ and $F_\beta$	
$f_{H\beta}$ Tooth trace angle deviation	$f_\beta$ Flank alignment error	Unchanged	Increase diminishing with width $b$ (instead of proportional as hitherto)	
$F_\beta$ Tooth trace total deviation	—	Redefined	According to ISO 1328 – 1975	

