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BASICS OF INTERCHANGEABILITY

Summary lectures

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O–75

Наведено основні норми взаємозамінності та єдиної системи допусків та посадок. Подано норми взаємозамінності в застосуванні до різьбових, шпонкових і шліцьових з'єднань, підшипників кочення та зубчастих передач. Викладено основні положення вимірювань і контролю калібрами, розрахунку розмірних ланцюгів і статистичних методів контролю якості продукції. Розглянуто визначення відхилень і допуски форми та розташування поверхонь, шорсткість і хвилястість поверхонь.

Для студентів механічних спеціальностей, які вивчають дисципліну «Основи взаємозамінності».

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Basic norms of interchangeability and unified system of tolerances and fits are submitted. Interchangeability norms in applications to threaded, key and spline joins, frictionless bearings and gearings are considered. Basic concepts for measurements and check with gauges, calculations of dimension chains and statistical methods for products quality control are submitted. Determination of deviations and tolerances for surfaces form and position, surfaces roughness and waviness are considered.

For students of mechanical engineering specialties studying discipline "Basics of Interchangeability".

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Figures 66. Tables 17. Bibliogr.: 5 references

INTRODUCTION

Interchangeability of parts and units of modern machines and mechanisms greatly reduces costs in production cycle and repair works with ensuring high reliability and long service life. Standardisation is a basis for interchangeability. Standardisation establishes unified norms for elements of machines that allow producing them in different countries and factories with assembling machines at any other production area. Technical metrology is a basic component for construction of quality control system.

Discipline “Basics of interchangeability” is a natural and obligatory stage in education of engineering specialists. The discipline applies knowledge of other subjects (mathematics, physics, etc.) and is a basis for subjects of more specialized level directly connected with designing of machines and development of manufacturing processes. Knowledge of this discipline is also necessary for proper performance of design and manufacturing documentation.

1 INTERCHANGEABILITY, QUALITY AND STANDARDISATION BY ISO

1.1 Basic concepts and definitions

Interchangeability (complete) is a property of *independently* produced parts or assembly units to provide *state of operability* and *reliability* of mechanisms and machines under conditions of *fittingless assembling* or at repair.

State of operability (***operability***) is a state of an item, at which values of all actual parameters correspond to the requirements of technical specifications or normative technical documentation.

Reliability is a property of an item to keep in time the values of all parameters in specified limits, that is, to ensure state of operability with time.

Concepts of interchangeability are fundamental ones for designing. They are also provided at manufacture and used at exploitation of items.

Parameters of interchangeability:

- Geometric (dimension, shape, relative position of surfaces and others);
- Physical and mechanical (specific gravity, hardness, strength and others);
- Chemical (chemical composition, admixtures);
- Electrical.

Types of interchangeability:

- complete (functional);
- incomplete (restricted).

Complete interchangeability is an interchangeability, at which all types of parameters are ensured with the accuracy that allows to perform fittingless assembling (or replacement at repair) of any independently produced parts to obtain finished items. That is, parts can be manufactured independently in several shops (factories, towns, countries), and be assembled into assembly units or items in other factories.

Basic advantages of products produced under ***conditions of complete interchangeability*** are:

1) Development works for creation of new items are easier, faster, and cheaper, because basic elements are standardised (threads, splines, toothed gearing, etc.);

2) Manufacture of items is easier and cheaper (accuracy of blanks is specified, improved inspection methods, easier assembling and others);

3) Exploitation is cheaper (shortening of repair period and its high quality).

Along with complete interchangeability the ***restricted*** (***incomplete***) ***interchangeability*** is permitted. Its types are:

1) Group interchangeability (selective assembling);

2) Assembling on the basis of probability calculations;

3) Assembling with adjusting of dimensions or positions of separate parts;

4) Assembling with fitting of one of several assembling parts.

There is also difference between *external* and *internal interchangeability*.

External interchangeability is interchangeability of finished items, mainly, deliverables (aggregates, instruments, frictionless bearings, etc.), from which interchangeability is required by service indexes and geometric parameters of *joint surfaces*. *External interchangeability is always complete*. For example, when replacing the failed electric motor, a new motor is mounted on the same place, on the same joint surfaces, with the same dimensions. This is interchangeability by joint dimensions or *dimensional interchangeability*. New motor should be interchangeable also by operation indexes – by power, rotational velocity and other indexes. This is *parametric interchangeability*.

Internal interchangeability is interchangeability of separate parts, assembly units and mechanism *inside of each item*. For example, complete interchangeability by joint diameters of bearing rings and restricted interchangeability by rolling elements. That is, *internal interchangeability* can be *complete* and *incomplete*.

Interchangeability is inconceivable without *accuracy*.

Accuracy is a degree of conformity of real part (assembly unit) to design accuracy specified by a designer or by normative technical documentation.

Accuracy is an opposite characteristic of a tolerance. It is impossible to produce a part with exact dimension, for example, of 20 mm, because it means an infinite level of accuracy – 20.000000000000000000… For real parts the dimension is specified by two limit dimensions – maximum and minimum limit dimensions (e.g., 20.1 and 19.9 mm). Performed dimension of a part should be between these two specified limits. The larger difference between them, the larger permissible tolerance (specified error) for production is. And it is evident that the higher accuracy of specified parameter (smaller tolerance for a dimension), the larger production expenses are (Fig. 1.1). Therefore, the assigned *level of accuracy* should be always thoroughly *substantiated*.

Quality is a set of properties and characteristics of products or services, which gives them an ability to satisfy the specified or prospective needs.

Products or ***service*** is a result of activity or process. They may be material or nonmaterial, that is, intellectual.

Quality level is a relative characteristic of products based on comparison of quality indexes of estimated products with basic values of corresponding indexes.

Problems of products quality and standardisation are regulated by International Organization for Standardization (ISO), which includes more than 100 member-states producing approximately 90 % world industrial products.

Quality system is a complex of interrelated elements, including organisational structure, responsibilities, procedures, processes and resources necessary for realization of quality control. Quality system is created at a factory as a tool for realisation and assurance of definite policy and achievement of permanent objectives in the field of quality. Quality system should cover all stages of a product life cycle (Fig. 1.2).

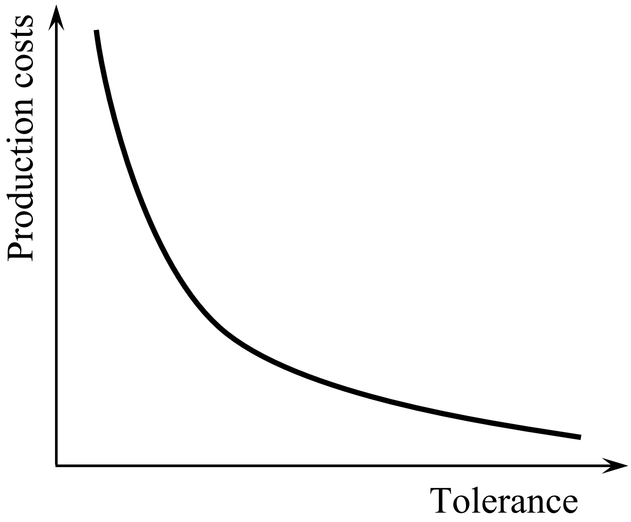


Figure 1.1 – Influence of accuracy on the costs for manufacture of products

Quality loop (spiral of quality) is a schematic model of mutually dependent types of activities that influence a quality of products or services at the sequent stages – from research of needs to estimation of their satisfaction (ref. Fig. 1.2).

Quality assurance is all planned and systematically realised activities in the framework of quality system, as well as, if necessary, other confirmed activities, which are needed for creation of satisfactory confidence that a factory will fulfil quality requirements.

Quality control is methods and types of activities of on-line character used for performance of quality requirements.

Quality improvement is actions made everywhere at a factory with a goal of improvements of efficiency and results of activities and processes for obtaining benefits for both a factory and consumers of products.

ISO develops standards of recommendation character (including ones for machine-building industry), which are the basis for national standards obligatory for performance. For instance, ДСТУ (DSTU) is National Standard of Ukraine; ГОСТ (GOST) is international standard for Commonwealth of Independent States (CIS). Branch standards ОСТ (OST) and factory standards СТП (STP) are developed on the basis of national standards. Each lower standard in this hierarchy must not contradict to paragraphs of a higher rank standard.

Standardisation is an activity, the essence of which is finding the solutions for repeated tasks in spheres of science, engineering and economics, directed to obtaining the optimal level regulation in a certain brunch.

Standardisation is a planned activity to establish obligatory rules, norms and requirements, performance of which ensures economically optimal quality of products, increase of labour productivity and effective use of materials and capital equipment at observation of safety regulations.

Standardisation is an "engineering laws". It is also a basis for interchangeability.

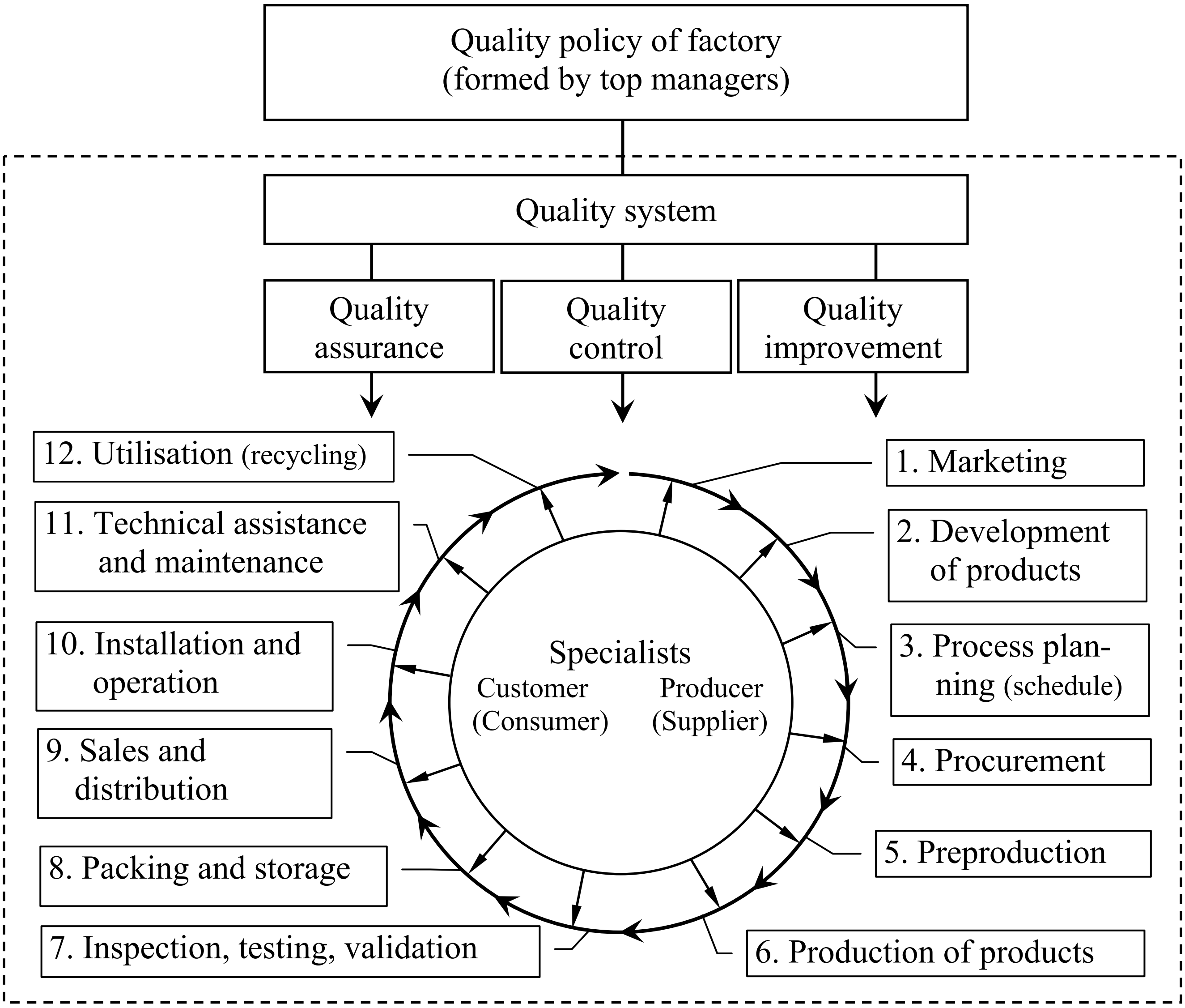


Figure 1.2 – Diagram of quality management system

Standardisation resolves contradictions between all participants of process for creation of products (including designers, production and maintenance engineers) due to they use the same source – standard.

Each standard contains a legislated error of manufacture that should be observed during some period of standard action (revision of a standard is performed each 5 years).

Types of standardisation are: unification, typification, and unitization.

Unification is transformation of some objects to uniformity, or unified shape. It is rational reduction of quantity of elements of similar functional purpose, but without reduction of systems variety, in which they are applied.

At unification the variety of applied similar elements (holes diameters, thread dimensions, types of rolled products) is reduced. For instance, if calculated diameter is equal to 16.92 mm, then according to basic series of preferred numbers the diameter of 16 mm or 18 mm should be assigned.

Typification is development and establishment of typical design or manufacturing solutions for several items, which have similar characteristics. For example, branch typification of manufacturing processes.

Unitization is a kind of standardisation, at which machine, equipment or production tools are composed from unificated aggregates (assembly units). When recomposing, the purpose of aggregates can be changed in a small degree. For example, multi-head machines, modular jigs and others.

Technical specifications are normative engineering documentation for a specific item, material or other product.

Technical requirements (spec) are established set of requirements to products, their manufacture, inspection, acceptance, etc.

1.2 System of tolerances and fits. Basic definitions

International ***Unified System of Tolerances and Fits*** (USTF) is a family of series of limits and fits, constructed in a relationship on a basis of experience, theoretical and experimental investigations and issued in the form of standards. USTF is applied for slick joints of cylindrical and flat parallel surfaces (Fig. 1.3).

According to ISO and USTF each internal element is called a “hole” (female surface), and each outside element is called a “shaft” (male surface). Parameters relating to a hole are designated by capital letters, and those relating to a shaft – by small letters (ref. Fig. 1.3).

1.2.1 Dimensions

Dimension is a numerical value of linear parameter (diameter, length, etc.) in selected units of measurement (millimetre, micrometer, etc.).

Limit dimensions are two permissible limits of size (maximum and minimum limit dimensions), between which the actual size of quality part should be or it can be equal to one of them.

Nominal dimension (***D***, ***d***, ***L***, ***l***, ***b***, ***c***) is a size, which is specified in a drawing on the basis of engineering calculations, design experience, etc., relative to which the devia-

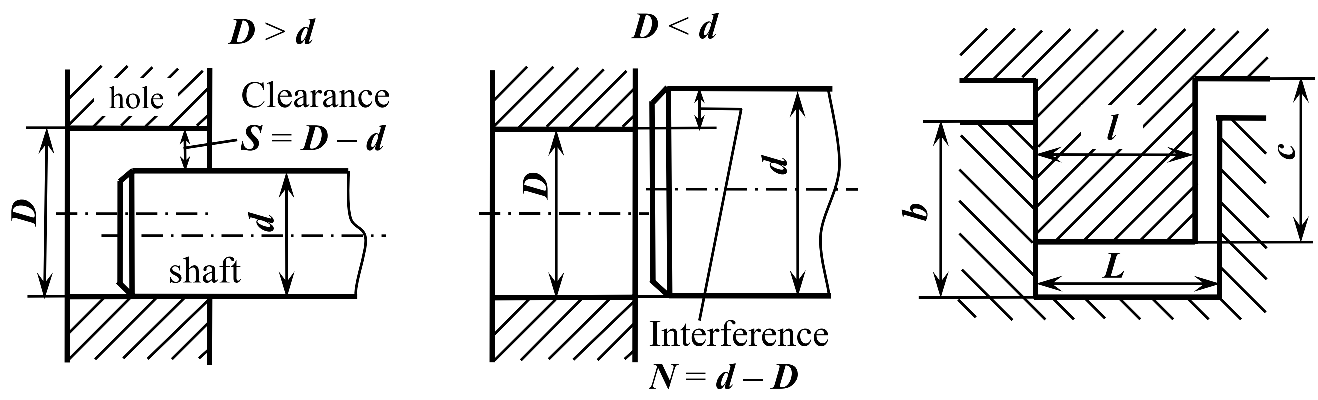


Figure 1.3 – Female and male dimensions of parts: ***D***, ***L*** – hole-type dimensions; ***d***, ***l*** – shaft-type dimensions; ***b***, ***c*** – neither shaft nor hole type dimensions

tions are determined (Fig. 1.4). In a drawing nominal dimension is specified in millimetres (mm). Nominal dimension and tolerance do not determine dimensional characteristic of a part, because position of tolerance is not specified yet. It is a basic deviation that determines position of tolerance relative to nominal dimension (zero line). Nominal dimension can coincide with maximum limit dimension or minimum limit dimension, or its value can be between two limit dimensions, or it can be more than the maximum and less than the minimum limit dimensions.

At graphic presentation tolerance bands of dimensions are located on the one side of diameters for convenience of plotting a diagram (ref. Fig. 1.4,c).

Dimensions, deviations and tolerances of hole and shaft shown in Fig. 1.4 are:
- Nominal hole diameter $\boldsymbol{D}$;
- Upper limit deviation of hole diameter $\boldsymbol{ES}$;
- Lower limit deviation of hole diameter $\boldsymbol{EI}$;
- Maximum hole diameter $\boldsymbol{D_{max} = D + ES}$;
- Minimum hole diameter $\boldsymbol{D_{min} = D + EI}$;
- Tolerance of hole diameter $\boldsymbol{T_D = D_{max} - D_{min} = ES - EI}$;
- Nominal shaft diameter $\boldsymbol{d}$;
- Upper limit deviation of shaft diameter $\boldsymbol{es}$;
- Lower limit deviation of shaft diameter $\boldsymbol{ei}$;

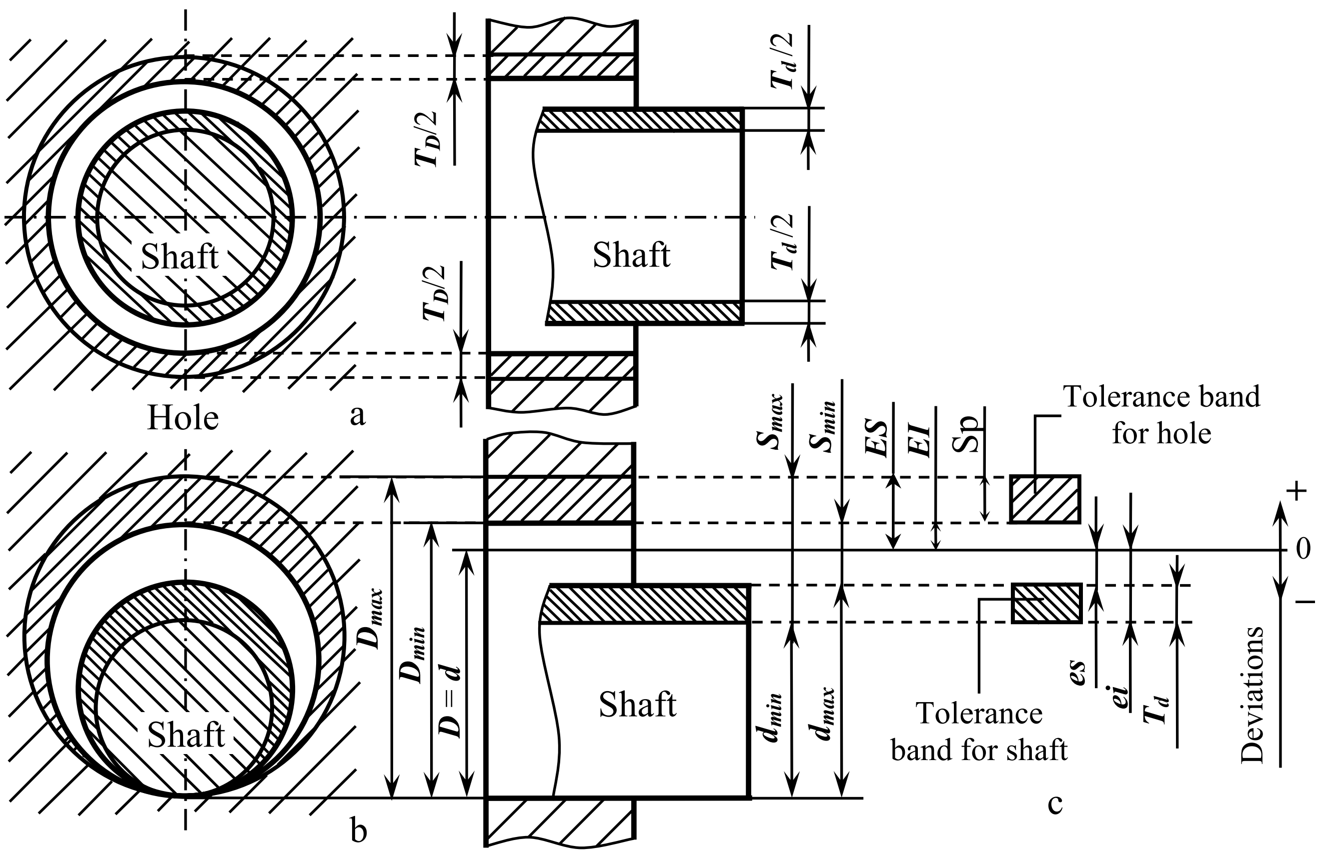


Figure 1.4 – Diagrams of real locations of tolerances in hole and shaft (a), one-side location of tolerances (b), limit dimensions, deviations and tolerance bands of hole and shaft (c)

- Maximum shaft diameter $\boldsymbol{d_{max}} = \boldsymbol{d} + \boldsymbol{es}$;
- Minimum shaft diameter $\boldsymbol{d_{min}} = \boldsymbol{d} + \boldsymbol{ei}$;
- Tolerance of shaft diameter $\boldsymbol{T_d} = \boldsymbol{d_{max}} - \boldsymbol{d_{min}} = \boldsymbol{es} - \boldsymbol{ei}$.

1.2.2 Deviations

Analysis of diagrams for joints and engineering calculations are very convenient with application of deviations (Fig. 1.5). Deviations are located relative to zero line. Position of zero line is determined by nominal dimension.

Upper deviation (ref. Fig. 1.5) is an algebraic difference between maximum limit of dimension and nominal dimension. It is designated: for hole by ***ES***, for shaft by ***es***:

$$\boldsymbol{ES} = \boldsymbol{D_{max}} - \boldsymbol{D}; \quad \boldsymbol{es} = \boldsymbol{d_{max}} - \boldsymbol{d}.$$

Lower deviation (ref. Fig. 1.4) is an algebraic difference between minimum limit of dimension and nominal dimension. It is designated: for hole by ***EI***, for shaft by ***ei***:

$$\boldsymbol{EI} = \boldsymbol{D_{min}} - \boldsymbol{D}; \quad \boldsymbol{ei} = \boldsymbol{d_{min}} - \boldsymbol{d}.$$

Sometime, for uniformity of formulas, the deviations of hole and shaft are designated by $\Delta\boldsymbol{S}$ (upper deviation of hole and shaft) and $\Delta\boldsymbol{I}$ (lower deviation of hole and shaft).

Mean deviation is an algebraic difference between mean dimension and nominal dimension. It is designated: for hole by ***EM***, for shaft by ***em***:

$$\boldsymbol{EM} = \boldsymbol{D_m} - \boldsymbol{D}; \quad \boldsymbol{em} = \boldsymbol{d_m} - \boldsymbol{d},$$

where

$$\boldsymbol{D_m} = \frac{\boldsymbol{D_{max}} + \boldsymbol{D_{min}}}{2}; \qquad \boldsymbol{d_m} = \frac{\boldsymbol{d_{max}} + \boldsymbol{d_{min}}}{2}.$$

Mean deviation can also be defined as an average of upper and lower deviations:

$$\boldsymbol{EM} = \frac{\boldsymbol{ES} + \boldsymbol{EI}}{2};$$

$$\boldsymbol{em} = \frac{\boldsymbol{es} + \boldsymbol{ei}}{2}.$$

Mean deviations for hole and shaft in general formulas can be designated with unified symbol $\Delta\boldsymbol{m}$ (mean deviation of hole and shaft).

Basic deviation is one of two limit deviations (the upper or lower) nearest to zero line.

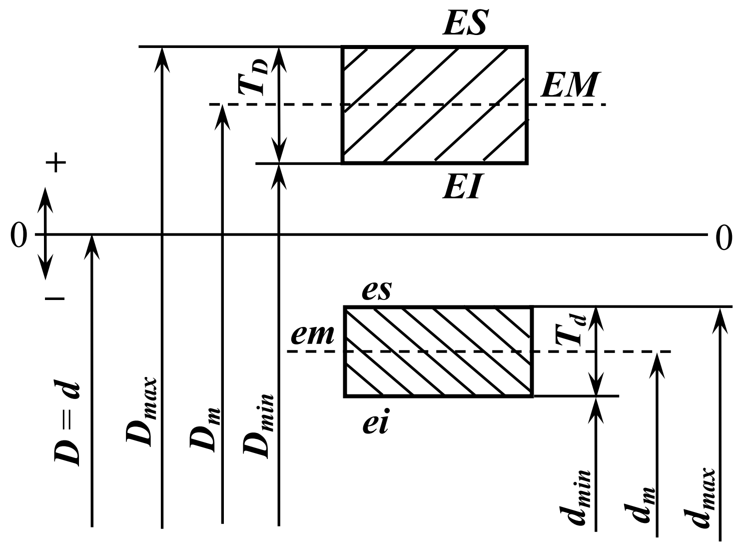


Figure 1.5 – Typical diagram for deviations and tolerance bands of hole and shaft

Deviations are specified:

- In micrometers (μm) in standards, diagrams, calculations of clearance (***S***) and interference (***N***) values;
- In millimetres (mm) in drawings;
- *Obligatory with a sign* ("+" or "–" except 0), because they can be positive or negative depending on their position relative to zero line (nominal dimension).

1.2.3 Tolerance bands

At graphic presentations the tolerance bands of dimensions are located relative to zero line (ref. Fig. 1.4, 1.5). Their positions are determined by positions (values) of upper and lower deviations: for hole ***ES***, ***EI***; for shaft ***es***, ***ei***.

Tolerance band differs from *tolerance* (T_D , T_d) by that it determines not only value, but also position of the tolerance relative to nominal dimension.

1.2.4 Fits

Fit is a joint of parts determined by difference of their dimensions *before assembling*. The required *fit* is ensured by relative displacement of tolerance bands of hole and shaft.

Types of fits are: *clearance*, *interference*, and *transition*, that is, movable and fixed fits (Fig. 1.6).

Clearance (***S***) is a difference of hole and shaft dimensions, when hole size is larger than shaft size (ref. Fig. 1.6). Maximum S_{max} and minimum S_{min} values of clearance are

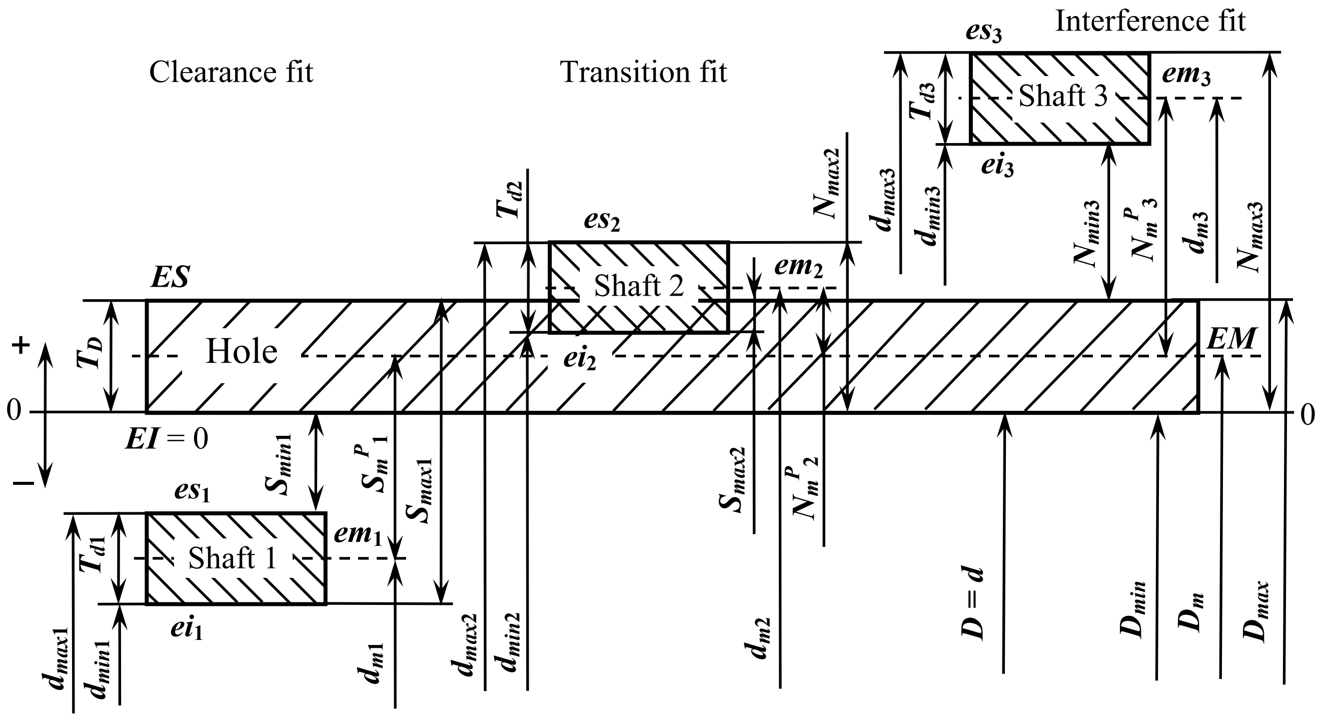


Figure 1.6 – Location of tolerance bands of hole and shafts for clearance, transition and interference fits

obtained from the formulas:

$$S_{max} = D_{max} - d_{min} = (D + ES) - (d + ei) = ES - ei;$$

$$S_{min} = D_{min} - d_{max} = (D + EI) - (d + es) = EI - es.$$

Interference (***N***) is a difference of hole and shaft dimensions, when hole size is smaller than shaft size (ref. Fig. 1.5). Maximum N_{max} and minimum N_{min} values of interference are obtained from the formulas:

$$N_{max} = d_{max} - D_{min} = (d + es) - (D + EI) = es - EI;$$

$$N_{min} = d_{min} - D_{max} = (d + ei) - (D + ES) = ei - ES.$$

Nominal dimension of fit is the same for hole and shaft, that is, nominal dimension of hole equals to nominal dimension of shaft ($D = d$).

Mean probable clearance S_m^P and ***mean probable interference*** N_m^P are calculated as a difference between mean diameters or mean deviations for hole and shaft from the formulas:

$$S_m^P = D_m - d_m = (D + EM) - (d + em) = EM - em;$$

$$N_m^P = d_m - D_m = (d + em) - (D + EM) = em - EM.$$

Fit tolerance is equal to sum of tolerances of hole and shaft creating a joint

$$T_\Sigma = T_D + T_d.$$

Operation requirements to fits are:

1. To *movable joints*: to create minimal clearance between hole and shaft that ensures friction with layer of lubricant.
2. To *fixed joints*: to ensure alignment of parts and transmitting a torque or axial force with guaranteed interference.
3. General requirement: to ensure maximal *life time*.

1.2.5 Tolerance – general idea

In order to understand better the essence of tolerance the following example is considered. In manufacturing process a machine-tool is set (adjusted) for dimension to be performed. Quantity of parts in a batch to be produced with this dimension is equal to n.

The parts cannot be equal by actual dimensions, because many factors will cause random errors of manufacture. These factors are: non-absolute rigidity of system “machine-fixture-workpiece-tool” (MFWT), thermal stresses in the MFWT system, wear of a cutting tool, inequality of allowance along workpiece surface and others.

Actual (measured) dimensions of these parts are (x_1, x_2,…, x_i,…, x_n). The extreme values x_{min} and x_{max} are selected among x_n values. The average (mean) value x_m, considered as the most probable value (true value), is calculated

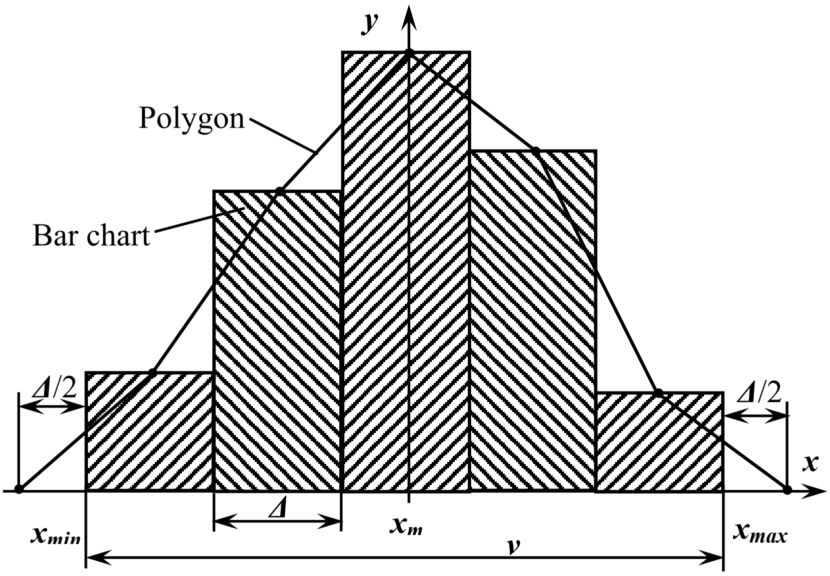


Figure 1.7 – Bar graph and polygon of scatter of actual dimensions

$$x_m = \frac{1}{n}\sum_{i=1}^{n} x_i ,$$

where n is a quantity of produced parts (quantity of measurements).

The scatter (dispersion) band of actual dimensions $v = (x_{max} - x_{min})$ is divided into 5, 7, or 9 (or another odd number) equal intervals Δ. The quantities of the measured dimensions that strike into each interval are calculated. The quantities (or relative quantities – probability density P_r) along axis y is plotted in a form of a histogram (bar graph). The polygon of scatter of actual dimensions (Fig. 1.7) is obtained by connecting the interval middles with a broken line.

The broken line (polygon) can be smoothed and approximated by continuous curve. This curve is called Gauss error curve; it describes normal law (Gauss law) of dimensions dispersion

$$y = \frac{1}{\sigma\sqrt{2\pi}} e^{-\frac{\Delta x^2}{2\sigma^2}} ,$$

where Δx – error (deviation) – difference between actual value of parameter x_i and average value x_m (assumed as a true value),

$$\Delta x = x_i - x_m;$$

σ – standard deviation that characterises influence of random errors,

$$\sigma = \sqrt{\frac{1}{n-1}\sum_{i=1}^{n} \Delta x^2} ;$$

e – base of natural logarithm; y – density of probability distribution.

The error values of considered dimension obeys the Gauss law and the scatter band of actual dimensions can be spread from average (true) value to left and right sides with the known value of 3σ, that is, (-3σ) and $(+3\sigma)$.

For satisfactory large quantity of measurements n and intervals Δ the experimental curve will coincide with the theoretical one. This range (confidence interval) from (-3σ) to $(+3\sigma)$ includes approximately 100 % (99.73 %) produced parts (Fig. 1.8). This interval ω is conventionally written down into standard and thereby the error is legalised and named as tolerance $IT = 6\sigma$. Symbol IT is considered as an international designation of tolerance for manufacture.

Each type of machining methods applied in industry (turning, milling, grinding, etc.) has its own attainable $\boldsymbol{\sigma}$ value and, hence, $\boldsymbol{IT}$ value.

So, the tolerance is a legalised error of manufacture and it is numerically specified in a drawing and equal to a difference between maximum limit and minimum limit of dimension:

for hole $\boldsymbol{T_D = D_{max} - D_{min}}$;

for shaft $\boldsymbol{T_d = d_{max} - d_{min}}$,

or is numerically equal to an algebraic difference between upper and lower deviations:

for hole $\boldsymbol{T_D = ES - EI}$;

for shaft $\boldsymbol{T_d = es - ei}$.

In these formulas $\boldsymbol{D_{max}}$, $\boldsymbol{ES}$, $\boldsymbol{d_{max}}$, $\boldsymbol{es}$ correspond to right boundary of confidential interval ($+3\boldsymbol{\sigma}$), and values $\boldsymbol{D_{min}}$, $\boldsymbol{EI}$, $\boldsymbol{d_{min}}$, $\boldsymbol{ei}$ correspond to left boundary of confidential interval ($-3\boldsymbol{\sigma}$).

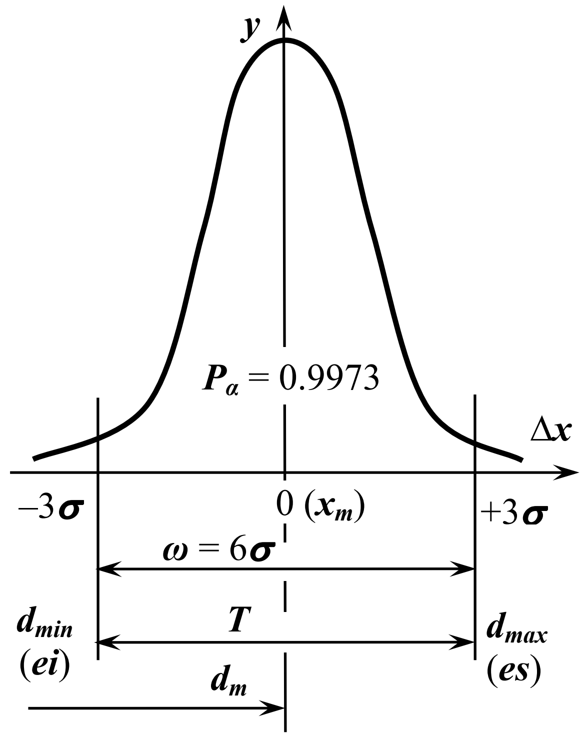


Figure 1.8 – Theoretical distribution density of actual dimensions in confidence interval and tolerance

Tolerance is always positive. In standards the tolerance values are given in micrometers and specified in drawings in millimetres.

1.2.6 Technological aspect

Initial data for planning of manufacturing process is the drawing of a part, output of parts, as well as, available basic equipment, tooling, methods for production of blanks and production periods.

This means that, when making analysis of the part drawing, first of all, one pays attention to accuracy of basic surfaces themselves and to accuracy of their relative location that determines necessary methods of their machining, datums (locating elements) and diagrams for mounting of a workpiece in operations, sequence of operations along manufacturing process. The output determines type of production, and, hence, degree of mechanisation and automation of manufacturing process including application of advanced methods for inspection of a part accuracy and other aspects. And so it is obvious that the manufacturing processes for production of 5, or 500, or 5000 parts will be significantly different.

The *task of technology* is *unconditional (obligatory)* performance of the drawing requirements, and doing that with the *highest efficiency* – with highest productivity and at least costs. This is a distribution of works (responsibilities) between designer and technologist during creation of products.

When planning a manufacturing process it is assumed in calculations that confidence interval for actual dimensions $\omega = 6\sigma = T_d$, that is, the part is considered to be the quality one, if its actual dimension does not go beyond limits of tolerance band.

Comments for providing the geometric accuracy are depicted in Fig. 1.9. A designer specifies parameters of accuracy: maximum d_{max} and minimum d_{min} limit dimensions, mean diameter d_m, tolerance T_d. In a manufacturing process setting-up (adjustment) of machine-tool is performed regard to the diameter d_m. Scatter v of actual dimensions is determined by maximum x_{max} and minimum x_{min} actual dimensions. Mean value x_m is typically displaced at setting-up error value μ. In order to obtain quality parts without reject the scatter v must not exceed tolerance T_d, the value x_{max} must not exceed specified value d_{max}, and x_{min} must not be less than d_{min}.

In industrial practice the quality control of a large batch of N parts is also performed by measurements of batch sampling of n parts and processing of measurements results with the *statistical methods*. In this case, standard deviation σ, confidence interval ω, maximum x_{Nmax} and minimum x_{Nmin} values are calculated for comparison with the specified tolerance T_d, maximum d_{max} and minimum d_{min} limit dimensions (ref. Fig. 1.9).

When realizing a manufacturing process, a technologist strives to perform two conditions:

1) $x_m \rightarrow d_m$ – to ensure coincidence of middle of scatter band x_m with mean diameter d_m (with centre of tolerance band em) by reduction of setting-up error μ;

2) Confidence interval ω is ensured of smaller than tolerance T_d value: $\omega = k \cdot T_d = k \cdot 6\sigma$, where coefficient $k < 1$.

Applied equipment and machining methods should guarantee performance of these conditions. This conditions increase production costs, but exclude spoilage (rejects).

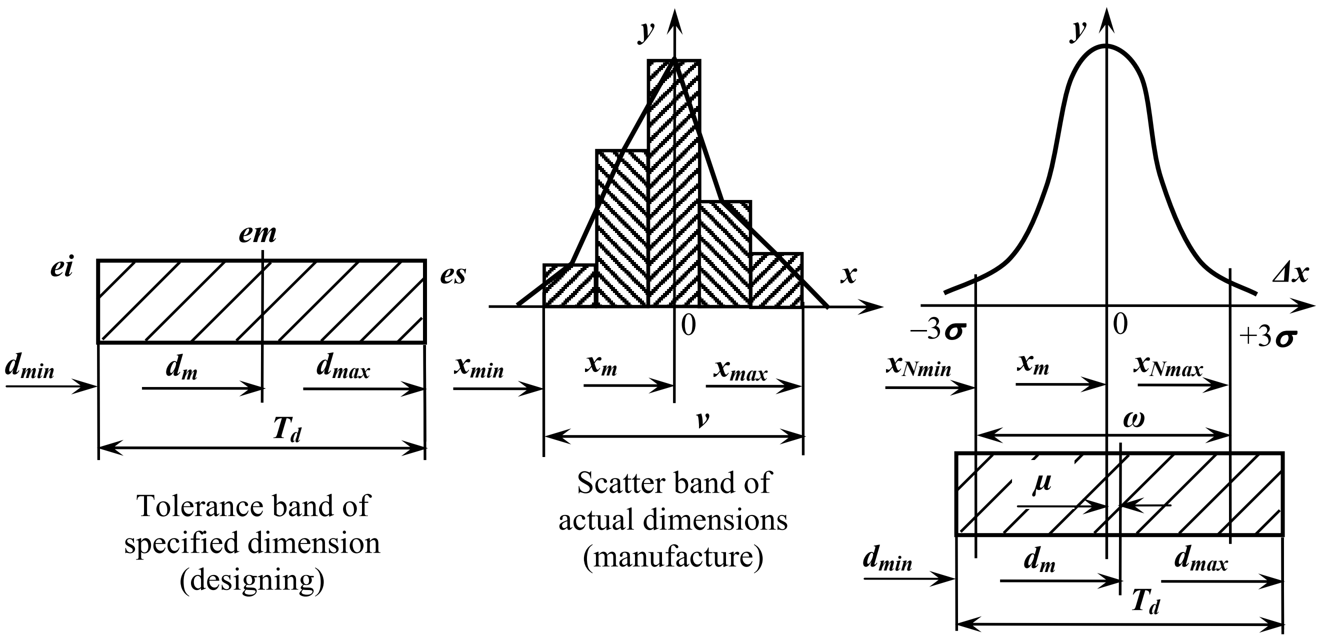


Figure 1.9 – Diagram for guaranteed providing the geometric accuracy

1.2.7 Preferred numbers of linear series

In order to reduce variety of dimensions, specified after calculations, a nominal dimension, as a rule, is rounded off with increase for shafts and with decrease for holes to the values of *preferred numbers*.

Preferred numbers and their series are the basis for selection of numerical values of parameters. Their purpose is to agree items, semi-products, materials, manufacturing equipment, inspection and other equipment.

In standards the numbers are rounded off. The standard contains 4 basic series of numbers R5, R10, R20, R40 located in geometric progression. Series from R5 to R40 contain correspondingly 5, 10, 20, and 40 numbers in each decimal interval, and numbers in these intervals are obtained by multiplication or division of numbers by 10, 100, 1000, that is, series of preferred numbers are unlimited (allow unlimited development of dimensions values) in both directions of decreasing and increasing of numbers (Table 1.1).

Table 1.1 – Basic series of preferred numbers for linear dimensions

R5	**R10**	**R20**	**R40**
Series index $\boldsymbol{\varphi} = \sqrt[n]{10}$			
$\sqrt[5]{10} \approx 1.6$	$\sqrt[10]{10} \approx 1.25$	$\sqrt[20]{10} \approx 1.12$	$\sqrt[40]{10} \approx 1.06$
Preferred numbers			
1; 1.6; 2.5; 4; 6.3; 10; 16; 25; 40; 63; 100 …	1; 1.25; 1.6; 2; 2.5; 3.15; 4; 5; 6.3; 8; 10; 12.5; 16 …	1; 1.12; 1.25; 1.4; 1.6; 1.8; 2; 2.2; 2.5; 2.8; 3.15; 3.6; 4 …	1; 1.06; 1.12; 1.15; 1.25; 1.3; 1.4; 1.5; 1.6 …

The standard does not cover *manufacturing operation dimensions* and the dimensions connected with other adopted dimensions by design relationships.

Preferred numbers series are widely applied in modern engineering including standards for tolerances and fits.

1.2.8 Specifying the dimensions

Dimensions are divided into *mating* and *non-mating* ones.

Mating dimensions are those, by which parts contact each other creating movable and fixed joints. Mating dimensions are important to ensure interchangeability; they should satisfy high requirements for accuracy (as a rule, accuracy grades 5 to 11).

Non-mating dimensions determine those surfaces, along which a part does not contact with other parts in item (as a rule, accuracy grades 12 to 18).

As a rule, tolerance bands or limit deviations of non-mating dimensions are not specified, and described by a general writing in engineering requirements of a drawing. Numerical values of unspecified limit deviations of linear dimensions may be set on the basis of either accuracy grades or special accuracy classes (Table 1.2).

Examples of specifying the accuracy of non-mating dimensions in engineering requirements of drawings are: 1) + $\boldsymbol{t_2}$, – $\boldsymbol{t_2}$, ± $\boldsymbol{t_2}$/2; 2) ± $\boldsymbol{t_2}$/2; 3) H14, h14, ± $\boldsymbol{t_2}$/2.

More often in aircraft branch the typical writing is applied in engineering requirements of drawings: "Unspecified limit deviations of dimensions are: H12 for holes, h12 for shafts, ± IT12/2 for others".

Table 1.2 – Special accuracy classes $\boldsymbol{t_i}$ for non-mating dimensions

Precise	Middle	Rough	Very rough
$\boldsymbol{t_1}$	$\boldsymbol{t_2}$	$\boldsymbol{t_3}$	$\boldsymbol{t_4}$
Respective accuracy grades			
(11), 12	13,14	15,16	17, (18)

Note. The accuracy grades, which are not recommended for application, are given in parentheses.

Taking into account the terms of dimension, tolerance, deviation, satisfactory information about specified design dimension may include (Fig. 1.10):

1) Two limit dimensions, or;
2) Nominal dimension, basic deviation, tolerance, or;
3) Nominal dimension, upper and lower deviations.

In *technological* documentation *executive dimension* is specified by limit value of dimension $\boldsymbol{D_{min}}$ or $\boldsymbol{d_{max}}$ and tolerance $\boldsymbol{T_D}$ or $\boldsymbol{T_d}$, which are directed "into metal":

$$\text{for hole } \boldsymbol{D_{min}}^{+T_D} \quad \text{and} \quad \text{for shaft } \boldsymbol{d}_{max_{-T_d}},$$

that means: for hole – nominal dimension is equal to the minimum dimension, upper deviation equals tolerance with plus, lower deviation equals zero; for shaft – nominal dimension is equal to the maximum dimension, upper deviation equals zero, lower deviation equals tolerance with minus.

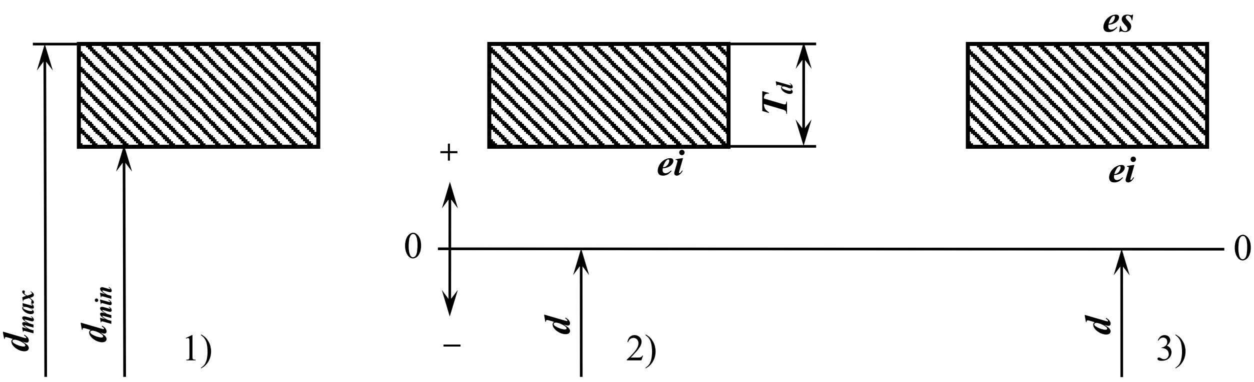


Figure 1.10 – Methods for specifying a design dimension

2 BASIC STANDARDS OF INTERCHAGEABILITY

Unified system of tolerances and fits (USTF) is a family of limits and fits series issued in the form of standards. Systems for typical joints have been developed: slick, conical, threaded, key, spline, gear and other joints. Each system is determined by a number of basic features.

Principles of system of tolerances and fits for slick joints are the following:

1. Intervals of nominal dimensions.
2. Unit of tolerance.
3. Grades of accuracy.
4. Basic part.
5. Positions of tolerance bands (basic deviation).
6. System of hole and system of shaft.
7. Temperature conditions.

2.1 Intervals of nominal dimensions

The ***first principles*** states that nominal dimensions are divided into ranges and intervals to simplify the tables of tolerances and fits.

Ranges are:

- Less than 1 mm – the small range;
- From 1 to 500 mm – the middle – is in the widest practice;
- More than 500 to 3150 mm – the large;
- More than 3150 to 10000 mm – the very large.

In their turn, the ranges are divided into ***intervals*** (*basic* and *intermediate*). For all dimensions, which are in the interval, the same tolerance is specified. The purpose of such an approach is to exclude the inconvenience of tables (for instance, 500 values with step of 1 mm for middle range) and excessive fractions of tolerance values of neighbouring dimensions included in this interval.

Here the following *rule* should be observed: dimensions are spread along intervals in such a manner that the tolerances, calculated from extreme values of each interval, differ not more than (5–8) % from the tolerances, calculated from average value of dimension in the same interval.

Calculations of tolerances and limit deviations for each interval are performed from geometric average value $\boldsymbol{D_i}$ and its limit values $\boldsymbol{D_{i\,min}}$ and $\boldsymbol{D_{i\,max}}$:

$$\boldsymbol{D_i} = \sqrt{\boldsymbol{D_{i\,min}} \cdot \boldsymbol{D_{i\,max}}}\ .$$

For example, according to standard the nominal dimension of 90 mm falls into interval (80–120) mm. The geometric average value of interval equals to $\boldsymbol{D_i} = \sqrt{80 \cdot 120} \approx$ ≈ 98 mm.

If difference of (5–8) % for tolerances, calculated from average value of dimension, is unacceptable, then basic intervals are additionally divided into intermediate intervals in respective places of the standard (Table 2.1).

Table 2.1 – Example of basic and intermediate intervals of dimensions

Basic interval	Intermediate intervals
120-180	120-140
	140-160
	160-180

If dimension value is equal to the extreme value of interval, deviations should be taken from that interval of the standard, in which this dimension appears first.

2.2 Tolerance unit

The ***second principle*** of USTF is a *tolerance unit* accepted in accordance with the rule of scale coefficients. The necessity of this term is explained with application of the following examples (Table 2.2).

Table 2.2 – Relative accuracy of dimensions

Case	Nominal dimension, mm	Tolerance, mm	Accuracy of dimension, mm
1	40	0.3	more accurate
	40	0.6	less accurate
2	120	0.3	more accurate*
	40	0.3	less accurate
3	120	0.35	?
	40	0.25	?

* With dimension increase it is more difficult to ensure the same tolerance, because accuracy of enlarged dimension increases too.

If nominal values of dimensions are equal, the more accurate will be that dimension, which has the smaller tolerance (see Table 2.2, case 1). If tolerances are equal, the more accurate will be that dimension, which has the larger nominal value (see Table 2.2, case 2). Accuracy comparison of dimensions with different nominal values and different tolerances is a certain problem (see Table 2.2, case 3). Therefore, it is reasonable to introduce term of "*tolerance unit*".

For determination of tolerance value it is necessary to separate the tolerance portions depending on nominal dimension and accuracy:

$$\boldsymbol{IT} = \boldsymbol{i} \cdot \boldsymbol{a},$$

where ***i*** – tolerance unit (function of dimension); ***a*** – number of tolerance units (function of accuracy).

In order to specify tolerances it is necessary to determine the relationship of their change depending on value of nominal dimension. Therefore, for construction of tolerances system the *tolerance unit* ***i*** is stated; ***i*** is a multiplier factor in tolerance formula, being the function of nominal dimension and serving for determination of numerical value of tolerance.

For middle dimension range (1–500) mm the tolerance unit is calculated from empirical relationship

$$\boldsymbol{i} = 0.45 \cdot \sqrt[3]{\boldsymbol{D_i}} + 0.001\boldsymbol{D_i}, \text{ micrometer } (\mu\text{m}),$$

where $\boldsymbol{D_i}$ – geometric average value of interval specified in millimetres (mm).

For example, for dimension of 90 mm (interval (80–120) mm) the geometric average value $\boldsymbol{D_i} \approx 98$ mm, and tolerance unit is

$$\boldsymbol{i} = 0.45 \cdot \sqrt[3]{98} + 0.001 \cdot 98 = 0.45 \cdot 4.61 + 0.098 \approx 2.17 \ \mu\text{m}.$$

Tolerance unit values have been calculated, rounded off and adopted in the form of standard.

2.3 Gradation of accuracy

The ***third principle*** of USTF system is *gradation of accuracy* (accuracy series). In each item the parts of different purpose are produced with different accuracy depending on specified requirements. In order to normalise specified accuracy the *accuracy grades* are introduced.

There are naturally constructed series of tolerances for each accuracy grade. In one series the different dimensions have the same relative accuracy determined by coefficient $\boldsymbol{a}$. Tolerances of dimensions of the same accuracy contains equal *number* $\boldsymbol{a}$ of tolerance units.

20 accuracy grades are established. They are designated by numbers increasing with tolerance value increase: 01; 0; 1; 2; 3… 17; 18.

Tolerance value is specified by letters IT (ISO Tolerance) and number of accuracy grade, for example: IT6, IT12.

Tolerance values for the 5–18 accuracy grades are determined by relationship $\boldsymbol{IT} = \boldsymbol{i} \cdot \boldsymbol{a}$ (Table 2.3). For other grades the tolerance values are determined from other relationships.

Table 2.3 – Number of tolerance units in accuracy grades 5 to 18 for dimensions of (1–500) mm

Grades	5	6	7	8	9	10	11	12	13	14	15	16	17	18
Number of tolerance units $\boldsymbol{a}$	7	10	16	25	40	64	100	160	250	400	640	1000	1600	2500

Starting from the 6th grade the number of tolerance units $\boldsymbol{a}$ changes according to geometric progression with the ratio $\boldsymbol{\varphi}$ = 1.6 (basic series R5 of preferred numbers). And so, neighbouring tolerances differ in 60 %, and when making steps in each 5 grades the tolerance value 10 times increases.

An accuracy grade determines tolerance value for machining, and, hence, determines methods and equipment for production of machine parts.

Knowing values of both multipliers in the formula for $\boldsymbol{IT}$, one can determine tolerance value for a dimension (Table 2.4). For example, for dimension of 90 mm in the 7th

grade the tolerance unit number $\boldsymbol{a} = 16$ (see Table 2.4 for accuracy grade 7), tolerance unit $\boldsymbol{i} = 2.17$ µm (see the previous example)

$$\boldsymbol{IT}_{\begin{pmatrix}90-\text{dimension}\\7-\text{grade}\end{pmatrix}} = \boldsymbol{i}_{\begin{pmatrix}80-120\\\text{interval}\end{pmatrix}} \cdot \boldsymbol{a}_{(7-\text{grade})} \approx 2.17 \cdot 16 \approx 35 \text{ µm}.$$

In order to check the tolerance value for dimension of 90 mm of the accuracy grade 7 see the Table 2.4.

Table 2.4 – Tolerance values for dimensions from 1 to 500 mm

No.	Basic intervals of dimensions, mm	Tolerance unit, $\boldsymbol{i}$, µm	Accuracy grades							
			5	6	7	8	9	10	11	12
			Tolerance $\boldsymbol{IT}$, µm							
1	From 1 to 3 including	0,55	4	6	10	14	25	40	60	100
2	Above 3 up to 6 incl.	0,73	5	8	12	18	30	48	75	120
3	Above 6 up to 10 incl.	0,90	6	9	15	22	36	58	90	150
4	-"- 10 up to 18 -"-	1,08	8	11	18	27	43	70	110	180
5	-"- 18 up to 30 -"-	1,31	9	13	21	33	52	84	130	210
6	-"- 30 up to 50 -"-	1,56	11	16	25	39	62	100	160	250
7	-"- 50 up to 80 -"-	1,86	13	19	30	46	74	120	190	300
8	-"- 80 up to 120 -"-	2,17	15	22	35	54	87	140	220	350
9	-"- 120 up to 180 -"-	2,52	18	25	40	63	100	160	250	400
10	-"- 180 up to 250 -"-	2,89	20	29	46	72	115	185	290	460
11	-"- 250 up to 315 -"-	3,22	23	32	52	81	130	210	320	520
12	-"- 315 up to 400 -"-	3,54	25	36	57	89	140	230	360	570
13	-"- 400 up to 500 -"-	2,52	27	40	63	97	155	250	400	630
	Number of tolerance units	$\boldsymbol{a}$	7	10	16	25	40	64	100	160
	Tolerance	$\boldsymbol{IT} = \boldsymbol{i} \cdot \boldsymbol{a}$	$7\boldsymbol{i}$	$10\boldsymbol{i}$	$16\boldsymbol{i}$	$25\boldsymbol{i}$	$40\boldsymbol{i}$	$64\boldsymbol{i}$	$100\boldsymbol{i}$	$160\boldsymbol{i}$

Note. Calculated values have been rounded off.

2.4 Positions of tolerance bands of basic parts

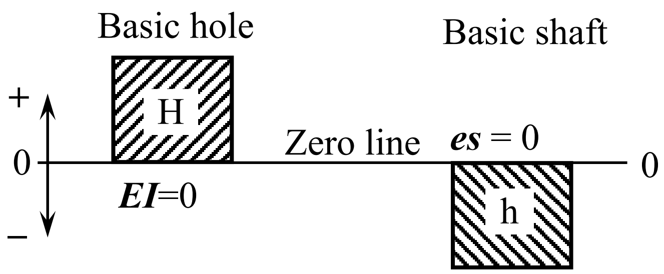


Figure 2.1 – Diagram for positions of tolerance bands of basic parts

The ***forth principle*** states limit one-side positions of tolerance bands of *basic parts* (Fig. 2.1).

Basic hole (H) is a hole, lower deviation of which is equal to zero ($\boldsymbol{EI} = 0$).

Basic shaft (h) is a shaft, upper deviation of which is equal to zero ($\boldsymbol{es} = 0$).

2.5 Basic deviations

The ***fifth principle*** establishes positions of basic deviations (tolerance bands).

Positions of tolerance bands are determined by standardised values of 28 basic de-

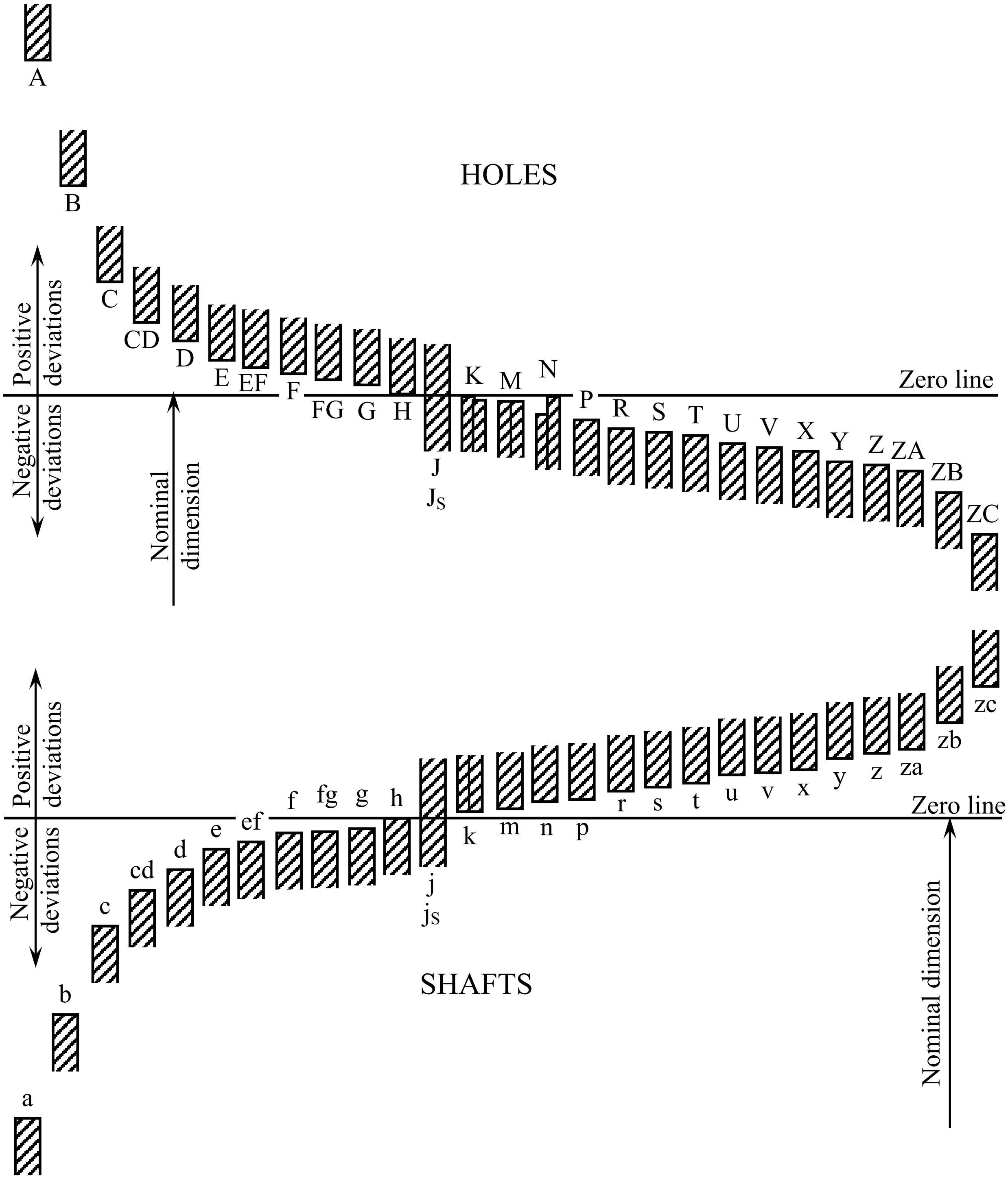


Figure 2.2 – Diagram for positions of basic deviations for holes and shafts

viations for shafts and 28 basic deviations for holes, which are designated by Roman letters (Fig. 2.2):

- Small letters from a to zc for shafts;
- Capital letters from A to ZC for holes.

Basic deviation is one of two limit deviations (the upper or the lower), which determines position of tolerance band relative to zero line. *Basic deviation* is a deviation

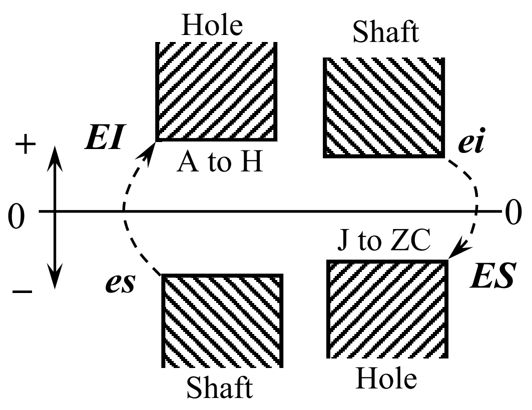


Figure 2.3 – Diagram of symmetric positions of basic deviations of shaft and hole

nearest to zero line.

Basic deviations are standardised and their numerical values nave been determined from the empirical formulas ***for shafts*** depending on intervals of nominal dimensions, for example:

- Basic deviations "e" and "f" (upper deviations):

$$es_e = -11 \cdot D_i^{0.41};\ es_f = -5.5 \cdot D_i^{0.34};$$

- Basic deviations "v" and "x" (lower deviations):

$$ei_v = IT7 + 1.25 \cdot D_i;\quad ei_x = IT7 + 1.6 \cdot D_i,$$

where D_i – geometric average value of interval of nominal dimensions.

Basic deviations for holes are mirror-like images of basic deviations for shafts (Fig. 2.3, see also Fig. 2.2 for comparison).

General rule: *basic deviation for hole should be symmetric to basic deviation for shaft of the same letter designation relative to zero line*.

Basic deviations for holes from A to H are calculated from the formula

$$EI = -es.$$

Basic deviations for holes from J to ZC are calculated from the formula

$$ES = -ei.$$

There is an exception from the general rule.

Special rule: *two fits in the system of hole and system of shaft, in which holes of certain accuracy grade are joined with shafts of higher accuracy grade, should have the same clearances and interferences* (Fig. 2.4).

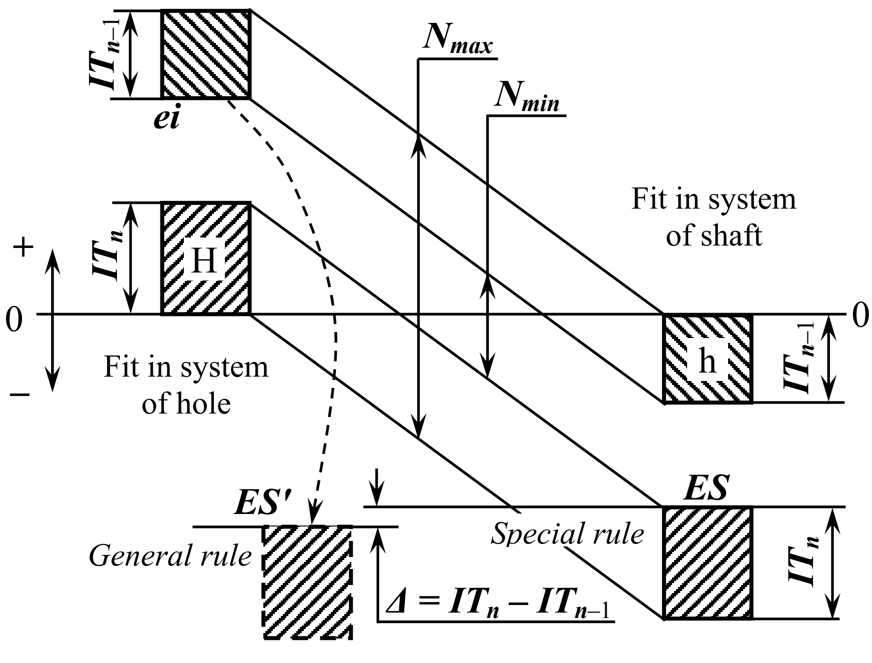


Figure 2.4 – Analytical diagram for calculations of basic deviation of hole according to special rule

Special rule is actual for holes with basic deviations J, K, M, N from the 5th to the 8th accuracy grades (transition fits) and for holes with basic deviations from P to Z from the 5th to the 7th accuracy grades (interference fits).

According to the special rule basic deviation for hole is calculated from the formula

$$ES = -\,ei + \Delta,$$

where $\Delta = IT_n - IT_{n-1}$ is a difference between tolerance of the considered accuracy grade and tolerance of the nearest more accurate grade.

Special rule ensures obtaining the same limit interferences of the same-name fits in system of hole and system of shaft at the condition that the hole tolerance is one accuracy grade lower than the shaft tolerance.

Basic deviations for all dimensions included into a certain interval of nominal dimensions are the same (equal) and do not depend on accuracy grade, because basic deviation value is a function of only dimension (Fig. 2.5).

Second (non-basic) deviation of any tolerance band is determined from values of basic deviation and tolerance:

- For hole: $ES = EI + T_D$; $EI = ES - T_D$;
- For shaft: $es = ei + T_d$; $ei = es - T_d$.

Tolerance band is formed by combination of basic deviation (letters) and number of accuracy grade (*digits*), for example, H7, s6.

Example of specifying the dimension:

Ø160s6,

where Ø – diameter; 160 – nominal value of dimension; s – basic deviation; 6 – accuracy grade; s6 – tolerance band.

Preferred tolerance bands. Tolerance bands are divided into 2 series: *basic* (recommended) and *additional* (of restricted application).

The narrower series of *preferred tolerance bands* is extracted from basic series for the first-priority application. They provide (90–95) % fits of general purposes and are 3–5 times cheaper in production.

Preferred tolerance bands are highlighted with square blocks or with bold type print in the standards tables.

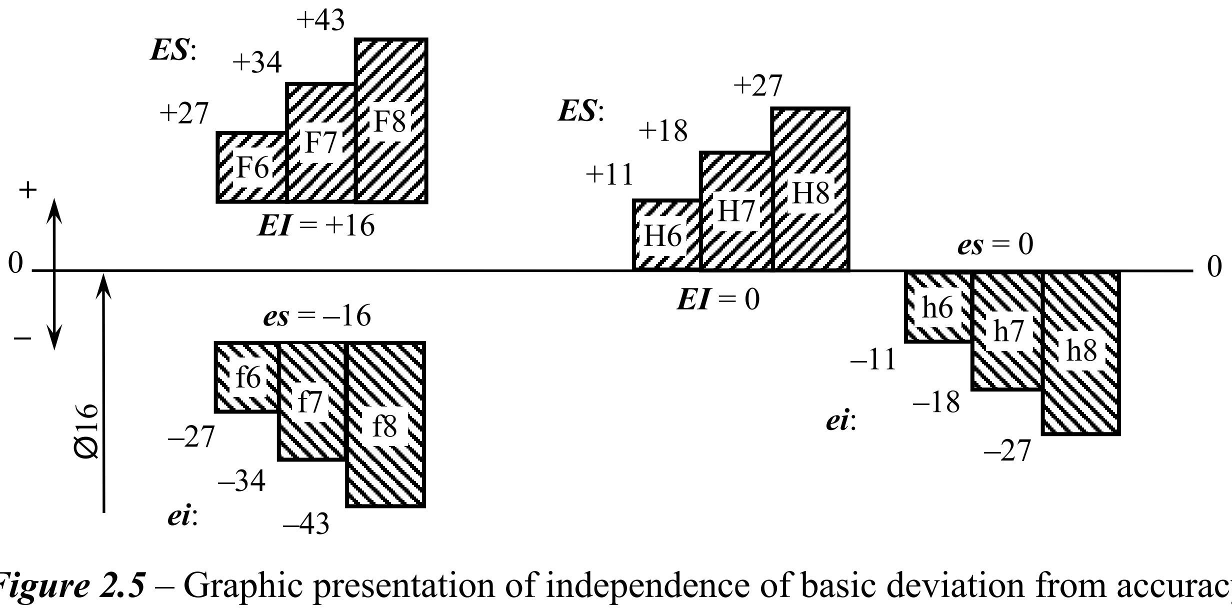


Figure 2.5 – Graphic presentation of independence of basic deviation from accuracy

2.6 System of hole and system of shaft

The ***sixth principle*** establishes fits in the *system of hole* and in the *system of shaft* (Fig. 2.6).

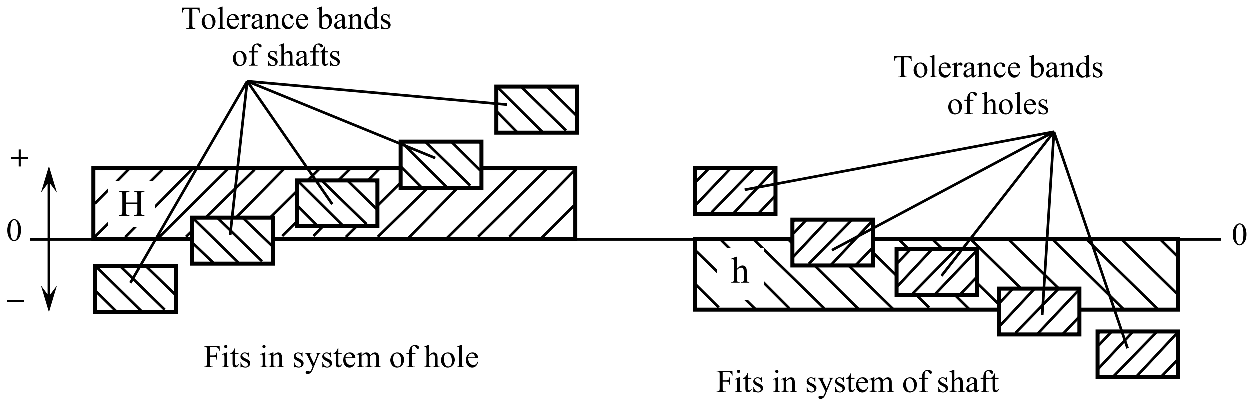


Figure 2.6 – Diagram of positions of tolerance bands in system of hole and system of shaft

In the ***system of hole*** the required clearances and interferences of fits are formed by combination of various tolerance bands of shafts with tolerance band of *basic hole* (H).

In the ***system of shaft*** the required clearances and interferences of fits are formed by combination of various tolerance bands of holes with tolerance band of *basic shaft* (h).

In a fit typically a hole has lower accuracy (larger grade number $\boldsymbol{IT_n}$) and a shaft has higher accuracy (smaller grade number $\boldsymbol{IT_{n-1}}$).

Systems of hole and shaft are formally equal. But *system of hole* is *preferable* in the most of cases. It is more economically sound, because shafts are easier and cheaper for production as compared with holes. *For machining of shafts* mainly versatile cutting tools are applied, access to work surfaces (outside open surfaces) is convenient. *For machining of holes* more complicated and precision cutting tools (drills, core drills, reamers, broaches, cutting tools fixed on boring bars) are applied, access to work surfaces (internal semi-closed surfaces) is restricted. These precision cutting tools are produced in a great amount for machining of H holes (basic holes).

But in some cases the system of shaft is more preferable – cheaper and technically more reasonable:

- When single-diameter smooth shaft should be inserted into several holes with different types of fits (Fig. 2.7);
- When applying the standardised units and parts produced under certain systems, for example, outside diameter of frictionless bearings is produced under the system of shaft, because it is unknown with what fit it will be mounted in a mechanism;
- When it is not reasonable to make multi-diameter (stepped) shaft to avoid stress concentration at the places of fillets.

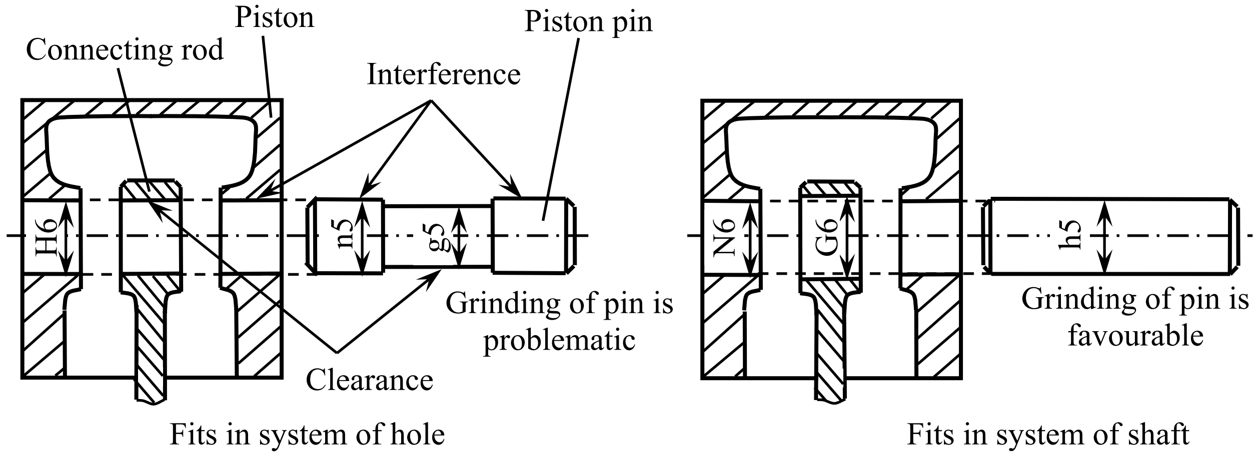


Figure 2.7 – "Piston-pin-connecting rod" assembly in system of hole and system of shaft

Specifying the dimensions. For example, the dimension of fit Ø90H7/g6 can be specified several variants (Fig. 2.8). In drawing a designer, as a rule, specifies dimensions in the form convenient for him: hole Ø90H7, shaft Ø90g6, fit Ø90H7/g6. It is permissible to specify dimensions with numerical values of upper and lower deviations: hole $Ø90^{+0.035}$, shaft $Ø90^{-0.012}_{-0.034}$, fit $Ø90\frac{+0.035}{\substack{-0.012\\-0.034}}$. The last form is also convenient for a technologist, because it gives numerical values of deviations. It is also permissible to a designer to specify dimensions in a combined form: hole $Ø90H7(^{+0.035})$, shaft $Ø90g6\left(^{-0.012}_{-0.034}\right)$, fit $Ø90\frac{H7}{g6}\left(\frac{+0.035}{\substack{-0.012\\-0.034}}\right)$.

Executive dimensions are dimensions, which are specified in operation sketches for machining the parts (ref. Fig. 2.8). For holes executive dimension is a minimum limit dimension with positive upper deviation (equal to the tolerance T_D); for shaft it is a maximum limit dimension with negative lower deviation (equal to the tolerance T_d). That is, in technological sketches deviations are specified "into body" of workpiece (into metal) that ensures *maximum material condition* for manufacture and higher probability of quality items.

Setting-up dimensions are dimensions, by which setting-up (adjustment) of machine-tools is performed in batch production, as well as of machines with computer numerical control (CNC). Numerically setting-up dimension is equal to the mean value of maximum and minimum limit dimensions. Graphically it is shown in the middle of tolerance band of shaft (ref. Fig. 2.8). Tolerance for a setting-up dimension approximately equals (20–30) % tolerance of executive dimension.

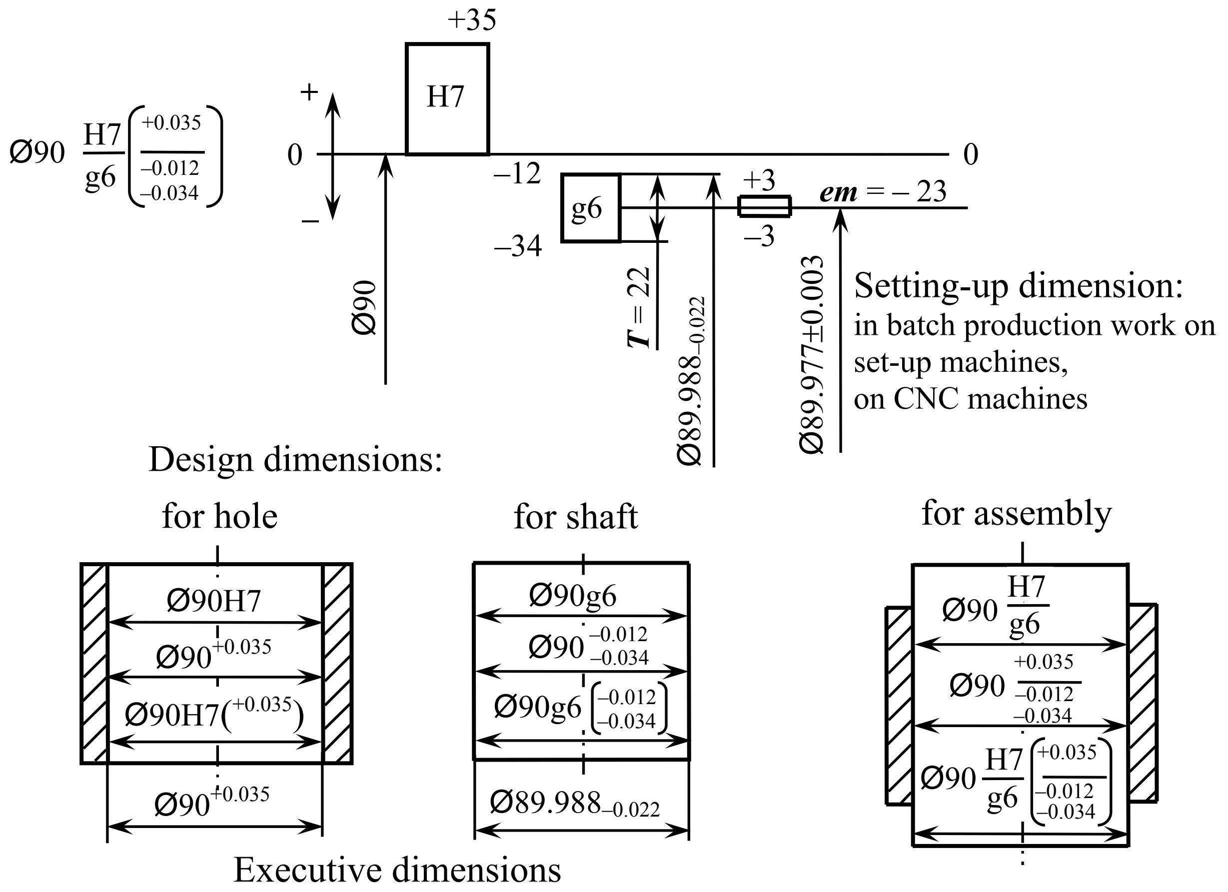


Figure 2.8 – Specifying dimensions of holes, shafts, assembly units in drawings and in technological documentation

2.7 Temperature conditions

Standardised tolerances and fits in the USTF system are calculated from the condition of parts inspection at normal temperature of +20ºC.

If temperatures of item t_1° and measuring tool t_2° do not coincide and differ by more than 2ºC from +20ºC, then the correction is applied:

$$\Delta l \approx l\,(\alpha_1 \Delta t_1^\circ - \alpha_2 \Delta t_2^\circ),$$

where l – measured dimension, mm; α_1 and α_2 – linear expansion coefficients of materials of part and measuring tool, respectively; Δt_1° and Δt_2° – difference between temperature of part t_1°, temperature of measuring tool t_2° and normal temperature 20ºC, respectively: $\Delta t_1^\circ = t_1^\circ - 20$; $\Delta t_2^\circ = t_2^\circ - 20$.

3 MEASUREMENTS AND INSPECTION

3.1 Basic definitions

Measurement is a determination of value of physical parameter with aid of special measuring means, for example, rule, vernier calliper, micrometer, that is, measuring means with a graduated scale.

Measurements are performed in pilot or small-batch productions, at repair and experimental works, and in batch and mass production – at setting-up of machine-tools or statistical methods of inspection (samplings).

Actual dimension is a dimension determined by *measurement* with permissible *error*.

Measurement is a determination of actual value of physical parameter by experimental way with aid of special measuring means.

Measuring error (inaccuracy) is a deviation of measurement result from true value of measured parameter. It depends on accuracy of measuring equipment. Accuracy of readings from scale and rounding-off, as a rule, is equal to half-division of scale (0.5 div.).

Error value is a difference between actual (real) value and specified value of parameter. The following ***types of errors*** are distinguished.

Systematic errors are called those errors, which are constant by absolute value and sign or changing by a certain law depending on character of non-random factors. For example, if there is an error in calibration of scale of measuring device, then this error is included into all readings. So as the error magnitude and sign are known, these errors are eliminated with corrections.

Correction is a value of parameter similar to the measured one, added to the value obtained at measurement with aim to exclude systematic error.

Random errors are called those errors, which are non-constant by absolute value and sign, which appear at manufacture and measurement and depend on random factors (uneven allowance removed from workpiece, inaccuracy of workpiece mounting, etc.).

Blunders or ***rough mistakes*** are called those errors, which significantly exceed expected systematic or random errors under the specified conditions of measurements.

Direct measurement is a measurement, at which desired value of parameter is obtained directly (by reading of measuring instrument), for example, measurement of length with a vernier calliper or micrometer.

Indirect measurement is a measurement, at which desired value of parameter is calculated from the known relationship between this parameter and parameters obtained from experimental data. For example, determination of area of a rectangle is performed by multiplication of measured values of length and width.

Relative measurement is a measurement, at which deviation of dimension from the selected measure or reference part, conventionally assumed as a unit, is measured. For example, measurement with a dial indicator set-up to dimension with gauge-block assembly is used.

Measurement methods can be of contact or non-contact type.

Contact measurement method is a method, at which sensitive element of instrument comes in contact with a subject being measured, for example, temperature measurement with a thermometer contacting with a part.

Non-contact measurement method is a method, at which sensitive element of instrument does not come in contact with a subject being measured, for example, temperature measurement with a pyrometer not contacting with a part.

Inspection is an ascertainment of the fact that parameter being checked complies with the specified accuracy *without determination of its actual value* (whether the actual value falls into specified tolerance band or does not), for example, inspection of parts with fixed gauges.

3.2 Material limits

Maximum limit condition is a limit size, to which the largest quantity (volume) of a part material corresponds, that is, maximum limit dimension of shaft or minimum limit dimension of hole.

Minimum limit condition is a limit size, to which the smallest quantity (volume) of a part material corresponds, that is, minimum limit dimension of shaft or maximum limit dimension of hole.

In batch production the limit dimensions of shaft and hole are checked with *gauges*. Maximum limit dimension of shaft and minimum limit dimension of hole are checked with *GO-gauges*. These limits are called *maximum material limits*. Minimum limit dimension of shaft and maximum limit dimension of hole are checked with *NOT-GO gauges* and respective limits are called *minimum material limits*.

3.3 Check of parts with gauges

The ***similarity principle*** is a basis for *check* of parts with *fixed gauges*.

Similarity principle: fixed gauge should imitate a segment of that item, with which a checked part will be in contact. Hole is checked by gauge-shaft (plug gauge), and shaft is checked by gauge-hole (ring gauge and snap gauge).

Perfect gauges should be combined, that is, should check both dimension and shape. If, for example, plug gauge is a flat part, then only dimension is checked, but not form.

There are *in-process* and *post-process inspection methods*.

When applying in-process inspection methods in batch and mass productions (for example, in-process snap gauge), in essence, performance of manufacturing process is checked. Warning is generated when certain deviation of checked parameter appears

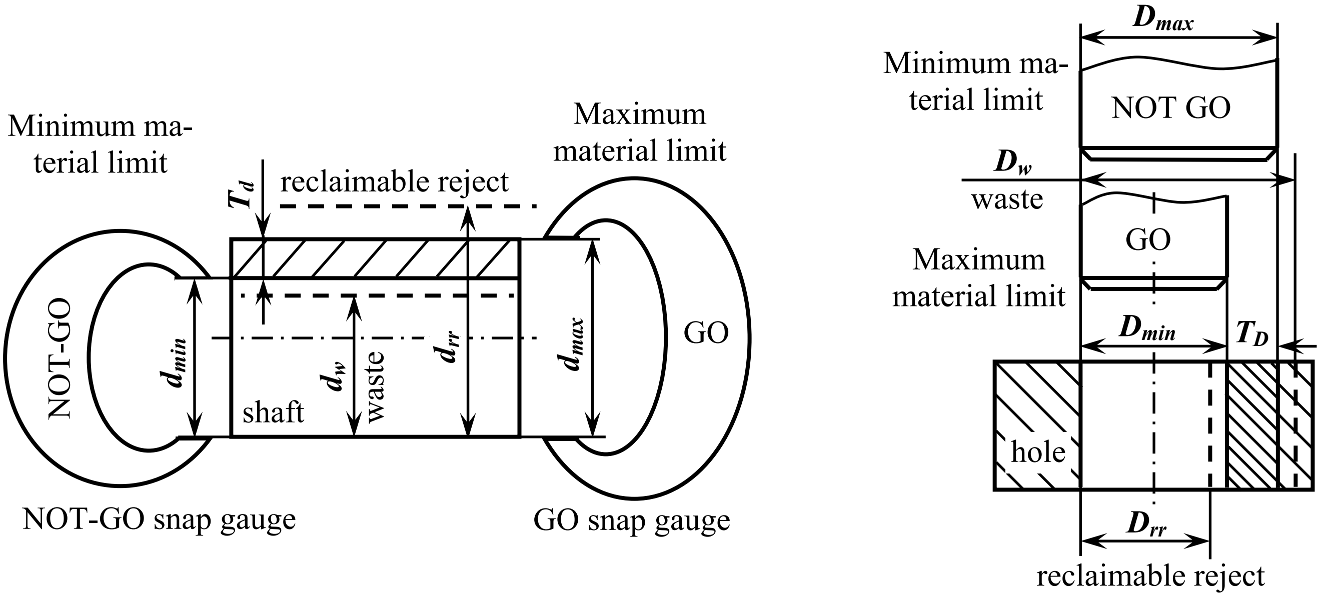


Figure 3.1 – Diagram for check of a shaft with snap gauges and check of a hole with plug gauges

before the moment when spoilage is produced – the tuning of manufacturing process is needed.

Post-process inspection methods (for example, fixed gauges) check conformity of ready-made parts with the specified dimensions in the drawing. Figure 3.1 shows that:

- This is inspection, but not measurement;
- Shaft is checked with a “hole” – snap gauge;
- Hole is checked with a “shaft” – plug gauge.

Snap gauge checks only dimension, in contrast to a complex gauge – ring gauge, which checks both dimension and form.

The part is good by dimension, if go-gauge (GO) *goes* through the part dimension, and not-go gauge (NOT GO) – *does not go*. If the part does not satisfy one of these conditions, it is a faulty part (reject).

Rejects can be reclaimable or waste. Reclaimable rejects are those faulty parts, which have actual dimension of shaft d_{rr} more than maximum limit dimension d_{max} and actual dimension of hole D_{rr} less than minimum limit dimension D_{min}. Wastes are shafts of diameter d_w smaller then minimum limit dimension d_{min} and holes of dimension D_w larger then maximum limit dimension D_{max}.

Variants of gauges are:

1) single-end single-limit;
2) double-end double-limit;
3) single-end double-limit.

Inspection rules:

- A worker checks with a new gauge, inspector – with a partially worn gauge;
- Must not apply large force;

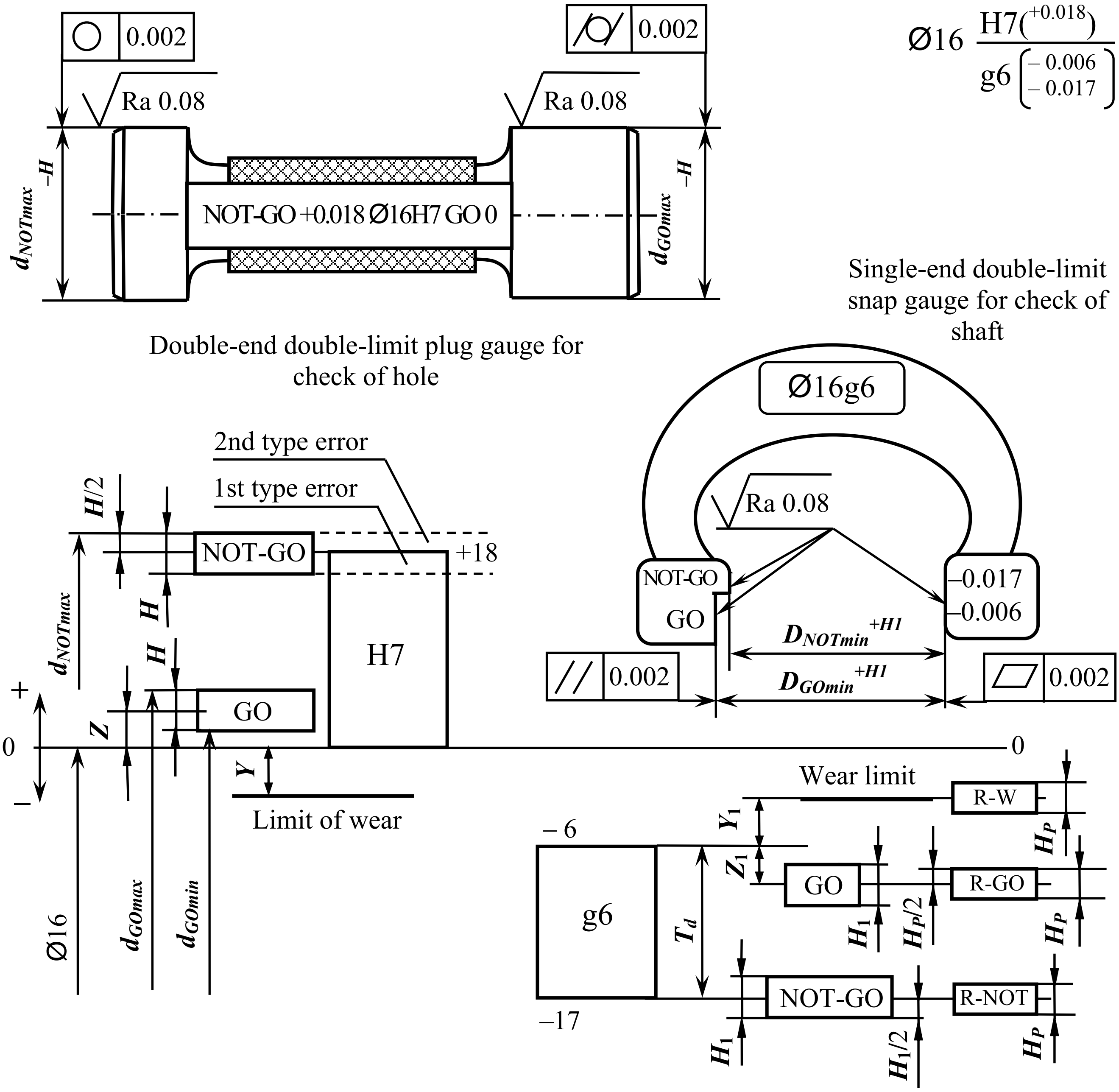


Figure 3.2 – Types of work gauges and diagram for positions of tolerance bands of items, work and reference gauges

- Observe temperature conditions.

Fixed gauges (Fig. 3.2) are divided into:

- Work: GO, NOT-GO;
- Inspection: I-GO, I-NOT-GO;
- Reference: R-GO, R-NOT-GO, R-W (check of Wear) – for inspection of work gauges *for shafts*. There are no reference gauges for inspection of work gauges for a hole.

Fixed gauges are typically applied in batch and mass productions.

Check of hole part is performed with a gauge-shaft; check of shaft part is performed with a gauge-hole, and check of gauge-hole is performed with a countergauge-shaft.

Errors of inspection with gauges. When item dimensions are near to the tolerance limits, there is a probability to make the following mistakes (ref. Fig. 3.2):

- *Errors of the 1st type* – when proper dimension is considered as a reject;
- *Errors of the 2nd type* – when reject is accepted as quality product.

Rules for rounding off the executive dimensions of fixed gauges:

1. Dimensions of work and inspection gauges for items are rounded off to the value divisible by 0.5 μm (0.0005 mm).
2. Dimensions, which end in 0.25 and 0.75 μm, are rounded off to the value divisible by 0.5 μm with decrease of manufacture tolerance (T_D or T_d).
3. In a drawing the deviations of executive dimensions are specified "into body" of gauge (ref. Fig. 3.2).

4 STATISTICAL METHODS FOR PRODUCT QUALITY CONTROL

In production the actual dimensions of parts are varying depend on systematic and random factors (errors). Systematic errors are essentially reduced by corrections at the stage of tuning (adjustment) of a manufacturing process.

Random errors are difficult or impossible to exclude. They should be reduced to the permissible value that does not exceed the value of tolerance for dimension or form. Random errors of measuring instruments and errors of manufacture are typically described by the normal distribution law.

In order to estimate quality of the parts produced under influence of random factors the statistical methods are applied.

4.1 Rules for measurements with errors

1. If *systematic error* is determinative, that is, its value is essentially larger than value of random error, then a measurement can be performed only one time.
2. If *random error* is determinative, the measurement should be performed several times.

Number of measurements should be selected in such a manner that random error of arithmetic mean x_m is less than the systematic error in order to the latter determines total error of measurements results.

For many parts *random errors* are described by *statistical characteristics*.

4.2 Basic statistical parameters and formulas

Basic statistical parameters and formulas are submitted in Table 4.1.

Table 4.1 – Statistical characteristics of random errors for batch of parts

Statistical characteristic	Definition, formula
$\boldsymbol{x_m}$	Empirical centre of grouping or arithmetic mean of considered parameter values, $x_m = \frac{1}{n}\sum_{1}^{n} x_i$, assumed as a true value of measured parameter
$\boldsymbol{M(x)}$	Statistical expectation – theoretically constant parameter, characterises centre of scatter of random errors, $M(x) \approx x_m$
$\boldsymbol{\Delta x_i}$	Absolute error, $\Delta x_i = x_i - x_m$
$\boldsymbol{v}$	Scatter band, $v = x_{max} - x_{min}$, where x_{max} and x_{min} – maximum and minimum actual (measured) dimensions, respectively
$\boldsymbol{\sigma}$	Standard deviation of random values of parameter from grouping centre, dimension parameter. It is a convenient characteristic for estimation of random errors values
$\boldsymbol{S_n}$	Mean-square error for n measurements, $S_n = \sqrt{\frac{1}{n-1}\sum_{1}^{n}\left(x_i - x_m\right)^2}$. If number of observations (measurements) is very large, then $S_n \to \sigma$, that is, $\sigma = \lim\limits_{n\to\infty} S_n$ (statistical limit). In reality one always calculates not σ, but its approximate value S_n , which is nearer to σ with n increase
$\boldsymbol{D}$	Dispersion, $D = \sigma^2$. It characterises degree of scatter of random errors around scatter centre. Sample variance $S_n^2 = \frac{1}{n-1}\sum_{1}^{n}\left(x_i - x_m\right)^2$. General dispersion $\sigma^2 = \lim\limits_{n\to\infty} S_n^2$.
$\boldsymbol{y}$	Probability density, $y = \frac{1}{\sigma\sqrt{2\pi}} e^{-\frac{(\Delta x)^2}{2\sigma^2}}$ or $\acute{o} = \frac{t}{\sigma}$, dimension function ([1/mm], [1/μm], etc.)
$\boldsymbol{P_\alpha}$	Probability, integral of the y function, $P_\alpha = P(-\Delta x \le \Delta x \le +\Delta x)$ for confidence interval from ($-\Delta x$) to ($+\Delta x$). It characterises level of reliability of obtained result. It is an area under the distribution curve bounded by abscissa axis and limits of interval
$\boldsymbol{P_\beta}$	Risk, sum of two quantiles, $P_\beta = (1 - P_\alpha)\cdot 100\%$, $P_\beta = 0.27$ % at $P_\alpha = 0.9973$

Table 4.1, continued

Statistical characteristic	Definition, formula
ω	Confidence interval, $\omega = 2z\sigma$, between $(-\Delta x = -z\sigma)$ and $(+\Delta x = +z\sigma)$
z	Argument of Laplace function $z_i = \frac{\Delta x_i}{\sigma} = \frac{x_i - x_m}{\sigma}$, dimensionless parameter
t	Probability density or probability distribution of random value $t = \frac{1}{\sqrt{2\pi}} e^{-\frac{z^2}{2}}$, dimensionless function
$\Phi_0(Z)$	Integral probability or Laplace normalised function, $\Phi_0(Z) = \frac{1}{\sqrt{2\pi}} \int_0^z e^{-\frac{z^2}{2}} dz$, dimensionless function

Distribution laws for random errors state relationships between values of random errors and probability of their appearance.

The mostly applied law in practice is *normal law of random errors distribution – Gaussian distribution law* (Fig. 4.1)

$$y = \frac{1}{\sigma\sqrt{2\pi}} e^{-\frac{(x-x_m)^2}{2\sigma^2}}.$$

With aid of Gauss curve it is possible to determine how often an error of some value will appear.

Standard deviation σ serves as an estimation of convergence of processing results, that is, degree of their concentration relative to the arithmetic mean value x_m. Dimensions dispersion increases with the σ increase. This means that large errors Δx (deviations from the x_m true dimension) will appear more often with the standard deviation σ increase.

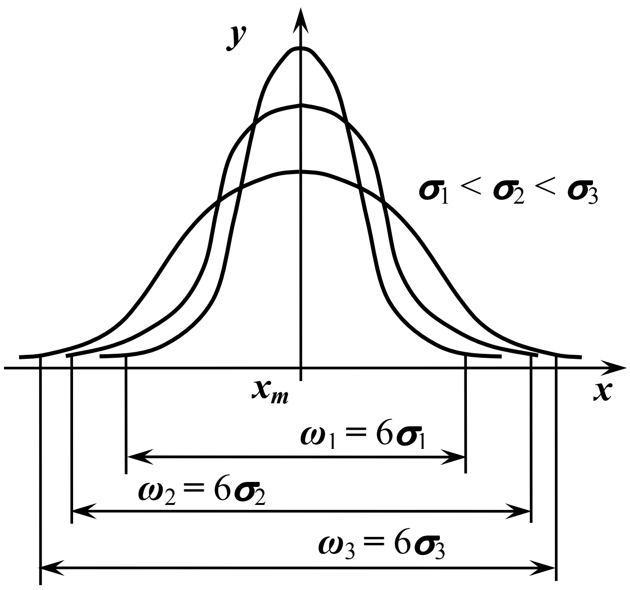


Figure 4.1 – Influence of standard deviation σ on the form of normal distribution curve

4.3 Properties and peculiarities of Gauss curve

1. The curve has two points of inflexion (convexity, concavity) at two values $\Delta x = \pm \boldsymbol{\sigma}$ (Fig. 4.2).
2. Curve is symmetric relative to the y_{max} value.
3. Two branches of curve tend to zero: at $\Delta x \to \pm \infty$, $y \to 0$.
4. Probability of sure event equals one (area under the curve)

$$P = \frac{1}{\sigma\sqrt{2\pi}} \int_{-\infty}^{+\infty} e^{-\frac{\Delta x^2}{2\sigma^2}} dx = 1.$$

5. The integral submitted in the paragraph 4 was transformed to obtain dimensionless argument of probability function: $z = \Delta x / \sigma$, $\Delta x = z\sigma$, and $dx = \sigma dz$. *Integral probability function* (*Laplace normalized function*)

$$\Phi_0(z) = \frac{1}{\sqrt{2\pi}} \int_0^z e^{-\frac{z^2}{2}} dz.$$

This function has the following properties: $\Phi_0(0) = 0$, $\Phi_0(-z) = \Phi_0(+z)$, $\Phi_0(-\infty) = \Phi_0(+\infty) = 0.5$, $2\Phi_0 = P = 1$ (Fig. 4.3).

Function $\Phi_0(z)$ is more convenient form of Gauss function, because it is dimensionless function and its values depends on only the values of dimensionless z number. It is applicable for any process and measurements results obeying Gauss distribution low. Therefore the function values are calculated and given in reference books in the form of table. Using these values and above formulas (ref. Table 4.1) it is easy to calculate values of function y and probability P.

Application of the σ standard deviation is convenient due to this parameter has respective certain value of confidence probability (ref.

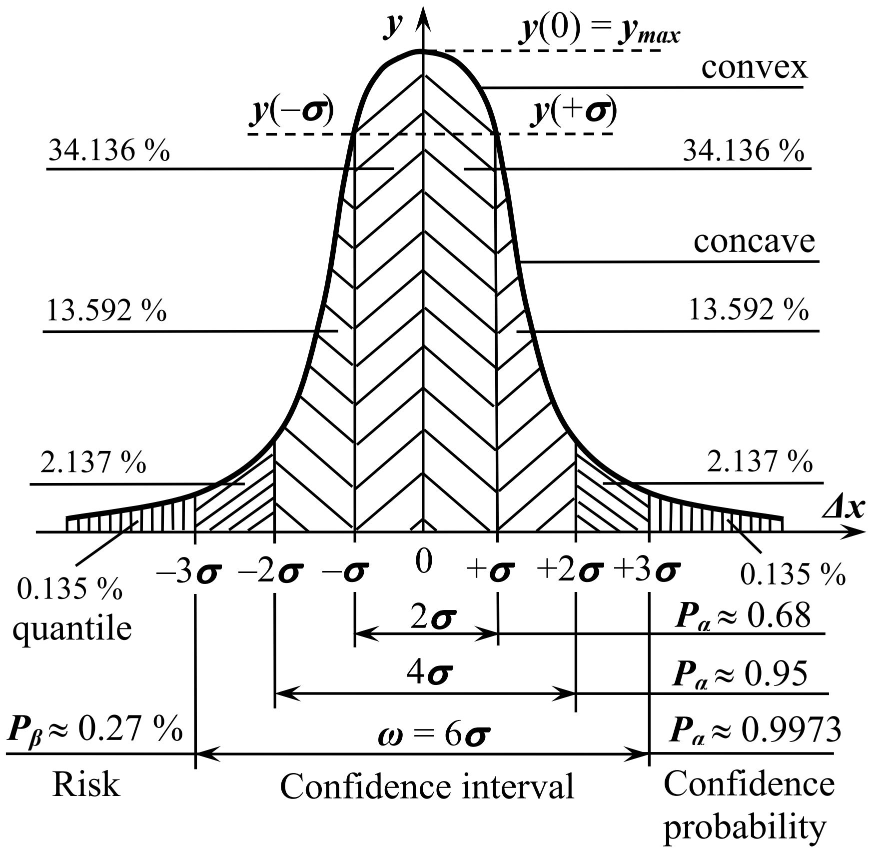


Figure 4.2 – Properties of the Gauss normal distribution curve ***y*** and areas ***P*** under the curve

Fig. 4.2).

When random parameters are distributed according to the Gaussian law, dispersion band $\boldsymbol{\omega} = 6\boldsymbol{\sigma}$ (in the interval from $(-\boldsymbol{\Delta x}$ to $+\boldsymbol{\Delta x})$ is assumed as practical limit confidence interval of these parameters, that is, with probability very near to one (0.9973). Here the probability of coming a random parameter out of the limits of $(\pm 3\boldsymbol{\sigma})$ is equal to $\boldsymbol{P_\beta} = 0.0027$ (0.27 %), that is, only 3 parts will be reject of batch of 1000 parts. So it is possible to assume that the random parameter will not come out the limits of $(\pm 3\boldsymbol{\sigma})$.

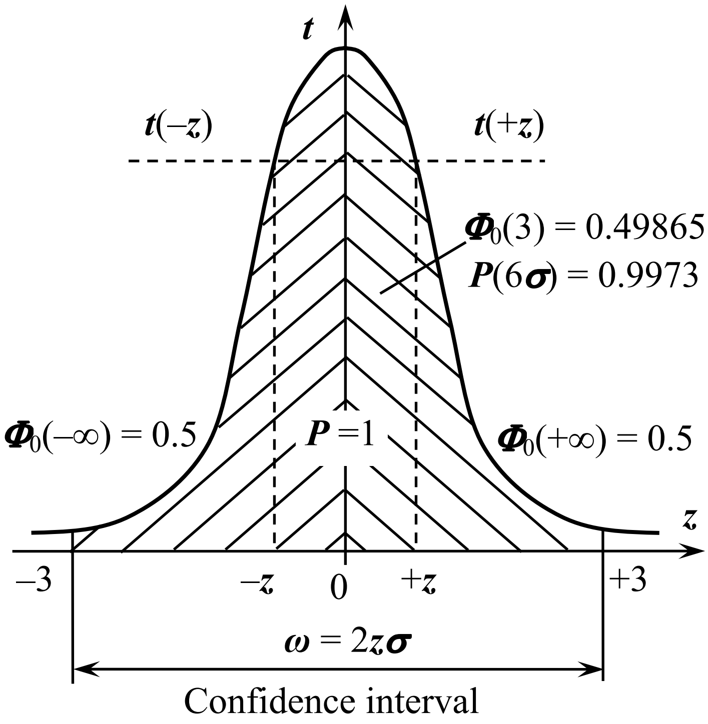


Figure 4.3 – Properties of dimensionless distribution function $\boldsymbol{t}$ (analog of the $\boldsymbol{y}$ function)

Normal distribution law is applied only then, when total error appears as a result of several random factors and each of them brings small portion into total error. That is, the main condition is absence of factors with dominant error.

4.4 Summation law for random parameters

Theory of probability proves that *mean-square error (standard deviation) of sum* or difference of two or several parameters, their measurement results being independent, equals the square root of *sum of dispersions of separate addendums* and for determination of total error it is necessary to add not errors themselves, but their squared values

$$\boldsymbol{S}_\Sigma = \sqrt{\sum_1^p \boldsymbol{S}_j^2}\ ; \quad \boldsymbol{\sigma}_\Sigma = \sqrt{\sum_1^p \boldsymbol{\sigma}_j^2}\ .$$

These follow from the law:

1) Significance (role) of separate errors very quickly decreases with their decrease. For example, for fit of hole and shaft the formula is $\boldsymbol{S}_\Sigma = \sqrt{\boldsymbol{S}_D^2 + \boldsymbol{S}_d^2}$.

Let us assume that $\boldsymbol{S}_d = ½\boldsymbol{S}_D$, then

$$\boldsymbol{S}_\Sigma = \sqrt{\boldsymbol{S}_D^2 + \left(\frac{1}{2}\boldsymbol{S}_D\right)^2} = \sqrt{\frac{5}{4}\boldsymbol{S}_D^2} \approx 1.1\boldsymbol{S}_D\ .$$

Total error only 10 % increases at the expense of the second error (equals 50 % the first error).

Conclusion: In order to improve a measurement accuracy of summarized parameter it is necessary, first of all, to decrease the largest error.

2) Arithmetic mean value x_m of series of measurements has smaller error, than the result of each separate measurement. This is formulated by the following law.

4.5 Law of accuracy improvements with increase of number of observations

Mean-square error S_{xm} of the arithmetic mean x_m is equal to mean-square error of each separate result divided by square root of observations number

$$S_{xm} = \frac{S_n}{\sqrt{n}},$$

where S_n – mean-square error of a single result (one batch of parts).

For example, in order to improve 2 times accuracy of the arithmetic mean x_m it is necessary to carry out 4 measurements instead of one, 3 times – 9 measurements instead of one, etc.

5 TOLERANCES AND FITS FOR FRICTIONLESS BEARINGS

Frictionless bearing is a complicated assembly unit. Typically it consists of outer and inner rings (including basic attachment surfaces), rolling elements and cage. Rolling elements are balls, rolls or needles in needle roller bearings.

Frictionless bearings are standardised items (Fig. 5.1). They have complete interchangeability by attachment surfaces of rings and restricted (incomplete) interchangeability for rings races and rolling elements. Therefore selective assembling is used for bearings (group interchangeability).

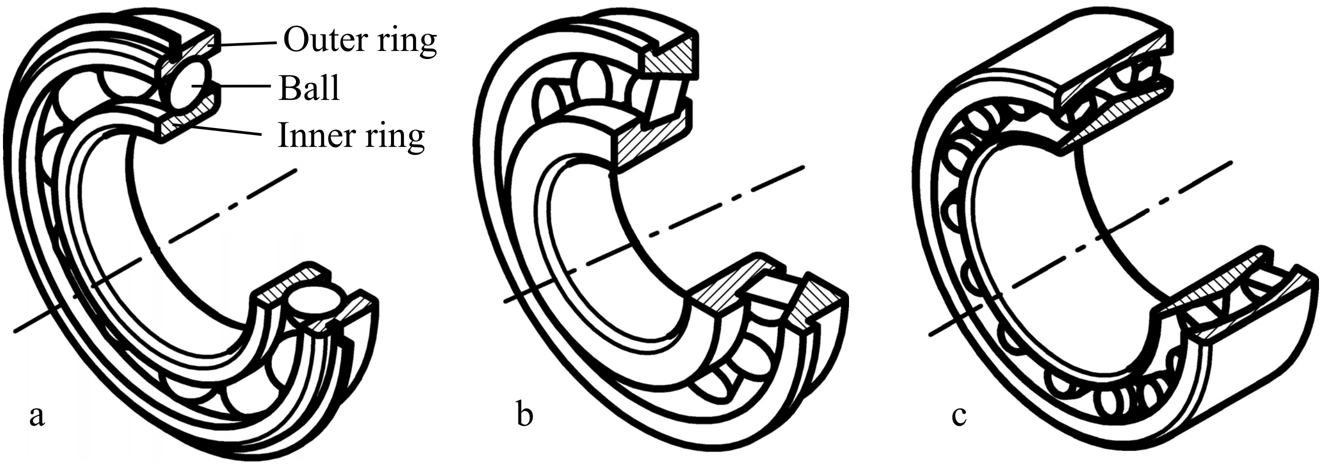


Figure 5.1 – Designs of frictionless bearings: a – radial ball single-row with a lock ring along outer ring (a design peculiarity); b – angular-contact single-row tapered roller with step along outer ring; c – radial roll double-row spherical

5.1 Distinctions of bearing fits from USTF fits

1. Hole – attachment internal surface of inner ring – is designated by small letter ***d***, and attachment outside surface of outer ring – shaft – by capital letter ***D***.

2. Quality of a bearing is described by *5 accuracy classes*: 0 – normal precision; 6 – extended; 5 – high; 4 – very high; 2 – extra-high. *Accuracy classes do not correlate with accuracy grades.*

3. Determination of dimensions conformity is performed by mean diameters:

- For inner ring $d_m = \frac{d_{max} + d_{min}}{2}$;
- For outer ring $D_m = \frac{D_{max} + D_{min}}{2}$.

Rings of many bearings are easily deformed because of their small thickness and so before assembling they can have ovality and dimensions exceeding the permissible ones. But after assembling they take proper shape and dimensions corresponding to permissible limits.

Thus, quality bearings are those, which have d_m and D_m located in the limits of tolerance.

4. Position of tolerance band of inner-ring hole diameter L differs from basic deviation H by location not upwards, but downwards relative to zero line (Fig. 5.2). Basic deviation of outer-ring diameter (shaft) is designated not by letter “h”, but “*l*” (L, *l* – lager (German) – bearing). This engineering solution is accepted because it is unknown what fits will be used for mounting a bearing in some mechanism.

Standardisation of bearing fits involves statement of limit deviations (accuracy classes) of bearing rings attachment surfaces, series of tolerance bands (accuracy grades) for shafts and holes in housings contacted with a bearing (Fig. 5.3).

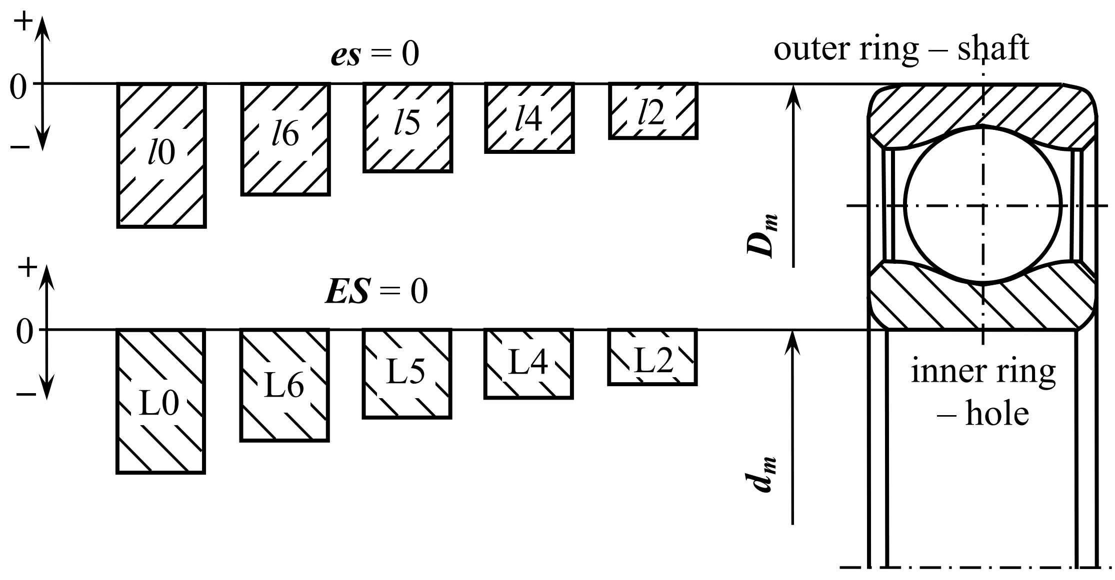
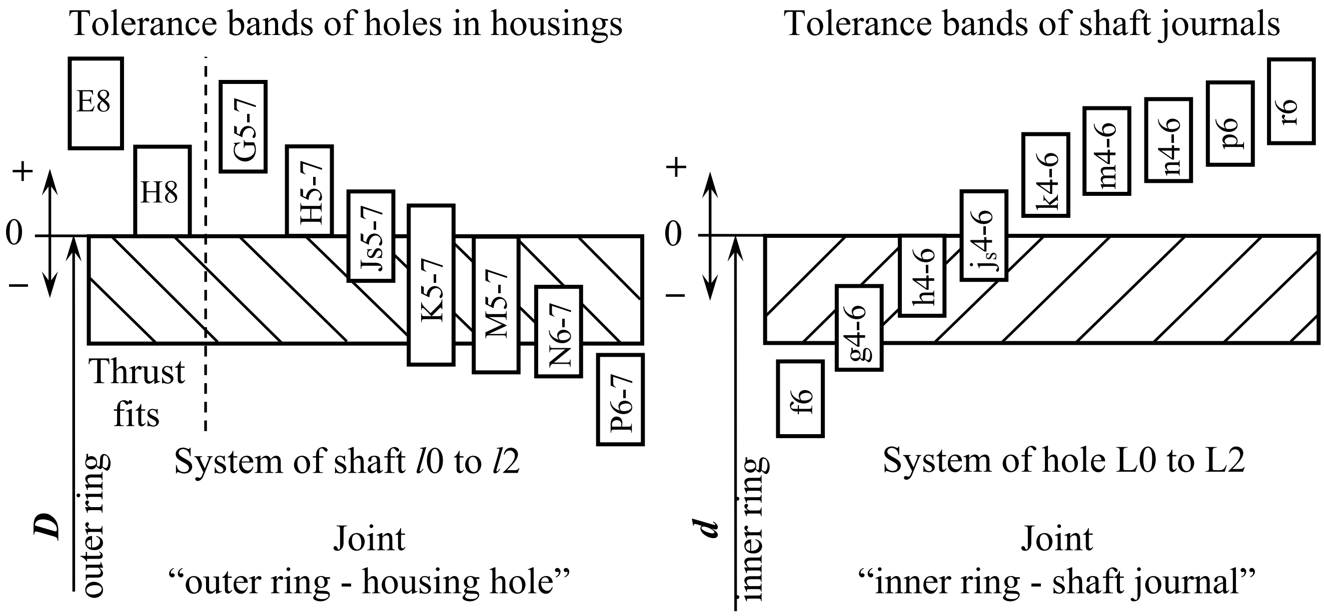


Figure 5.2 – Diagram for positions of tolerance bands for outer diameter and inner diameter of frictionless bearings



Accuracy class	Accuracy grade
0, 6	7
5, 4	6
2	5

Accuracy class	Accuracy grade
0, 6	6
5, 4	5
2	4

Figure 5.3 – Diagrams for positions of tolerance bands for fits on outer diameter and for fits on hole diameter of a frictionless bearing (radial and angular-contact types)

Applications of bearings accuracy classes:

0; 6 – requirements for precision of rotation are not specified;

5; 4 – at large rotational speed and more severe requirements for precision of rotation (for example, spindles of precision machine-tools);

2 – for special purposes (high-precision instruments, etc.).

5.2 Designation of bearings

Zeros located on the left hand from significant digit are not shown, for example, not 5–0001318, but 5–1318.

Let us consider designation of bearing 5–2431318.

Bearing designation is analysed *starting from the right end*:

- *Two first digits* – bearing inner diameter multiplied by 5. Exceptions are: 00 – bearing inner diameter equals 10 mm; 01 – 12 mm; 03 – 15 mm; the next and further designations are according to the rule: 04 × 5 = 20 mm, etc. (in the example: 18 × 5 = 90 mm);

- *The third digit* together *with the seventh* – bearing series by diameters: 0 – super light; 1 – extra light; 2 – light; 3 – medium; 4 – heavy. By width the bearings are divided into series: extra narrow, narrow, normal, wide and extra wide. In the example the bearing is of medium series by diameter and normal by width.

- *The forth digit* – bearing type: 0 – radial ball single-row; 1 – radial ball double-row spherical, etc. In the example the bearing type is radial ball double-row spherical.

- *The fifth and sixth digits* – designate design peculiarities.

Accuracy class (6, 5, 4, 2) is shown via dash. Precision class “0” is not specified. In this example the bearing is of 5th accuracy class.

5.3 Selection of fits for frictionless bearings

Fits are selected for each ring depending on rotation or fixation of a ring, type of loading, operation conditions, bearing type.

Determinative factor is *type of loading*: local, circulating, oscillating.

At *local loading* a *non-rotating ring* bears radial load with restricted segment of race and transmit it to restricted segment of mating surface of shaft or housing (Fig. 5.4 and 5.5). At local loading a fit is specified with small mean-probability clearance (ref. Fig. 5.3). During operation ring will periodically slide and turn under the action of shocks and vibrations, wear of race will be more uniform and so service life of ring will increase significantly.

At *circulating loading* a *rotating ring* bears a load, applied to mating surface of shaft or housing, sequentially along all length of race (ref. Fig. 5.4 and 5.5).

At *oscillating loading* a ring bears a radial load with restricted segment of race, but load changes its direction during one revolution in the limits of some angle, as well as its magnitude (Fig. 5.6).

At *circulating* and *oscillating loading* a fixed joint of ring with shaft is necessary. Fit of rotating ring under circulating load should ensure guaranteed interference (ref. Fig. 5.3) to exclude displacement or sliding of these rings relative to the parts.

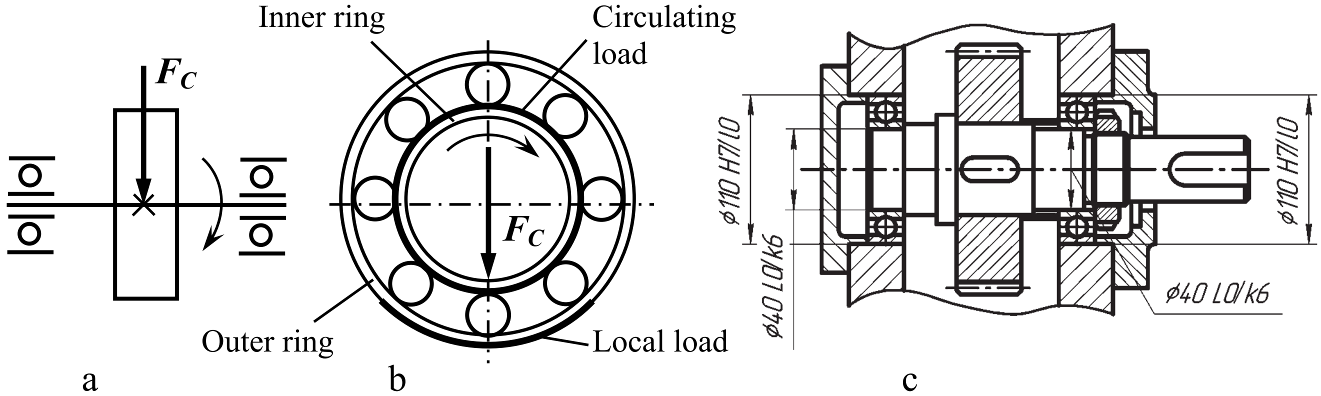


Figure 5.4 – Kinematic diagram with rotating inner ring of bearing (a) under constant force $\boldsymbol{F_C}$, types of loads for bearing rings (b) and typical fits for outer and inner rings (c) for the diagram

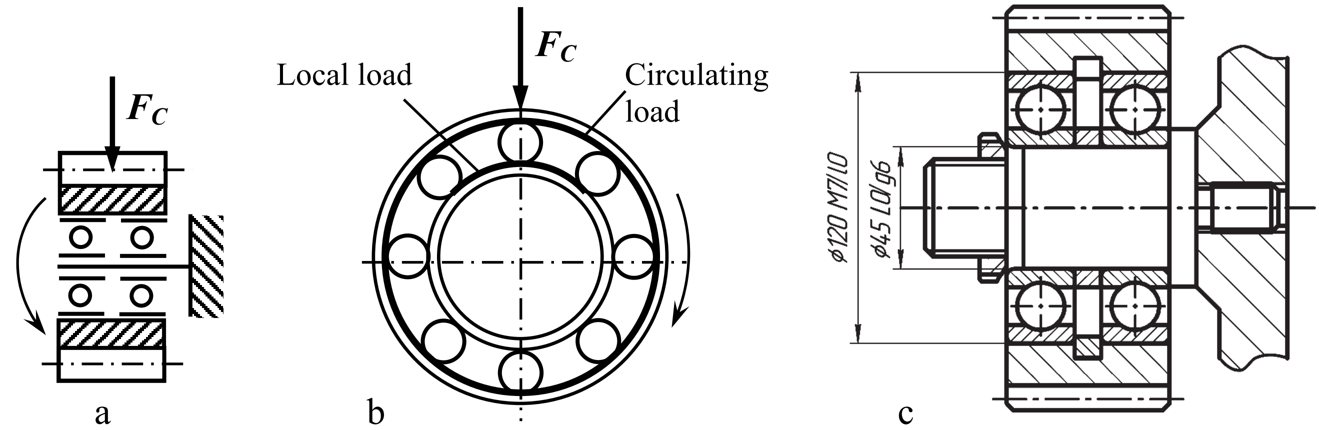


Figure 5.5 – Kinematic diagram with rotating outer ring of bearing (a) under constant force $\boldsymbol{F_C}$, types of loads for bearing rings (b) and typical fits for outer and inner rings (c) for the diagram

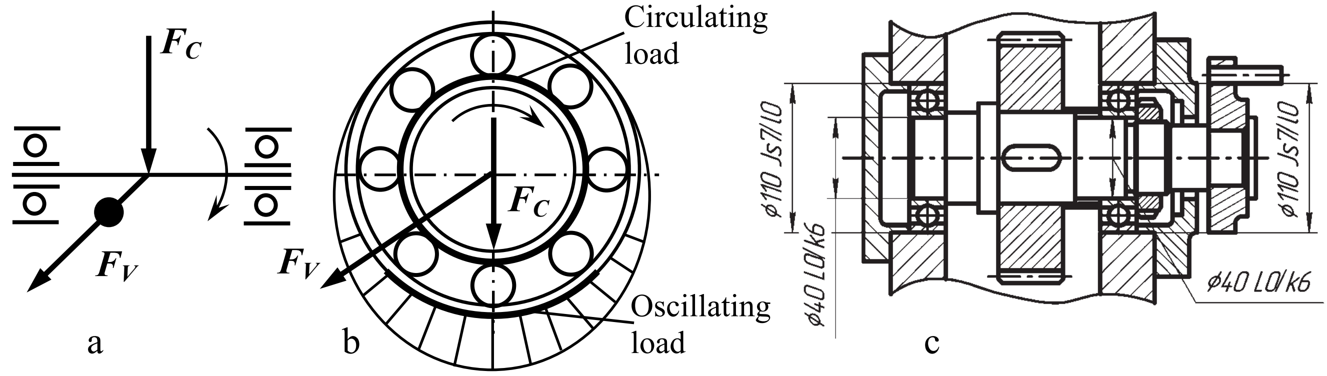


Figure 5.6 – Kinematic diagram with rotating inner ring of bearing (a) under constant $\boldsymbol{F_C}$ and varying $\boldsymbol{F_V}$ forces, types of loads for bearing rings (b) and typical fits for outer and inner rings (c) for the diagram

At *oscillating loading* if one of loads ($\boldsymbol{F_C}$ or $\boldsymbol{F_V}$) significantly exceeds the other, then the smaller load can be neglected and loading can be considered as local or circulating.

Main types of loads for frictionless bearings rings are submitted in the Table 5.1.

Operation conditions of a bearing and its service life are determined by ratio of load and dynamic load-carrying capacity (Table 5.2).

Requirements for accuracy of attachment surfaces of mating parts become more severe with increase of bearing accuracy class. Attachment dimensions of parts for bearings is restricted by permissible *form deviation* for cylindricity and limit deviations for *face runout* of shoulders of shafts and housing holes (Fig. 5.7). The parameter *skewness angle* between axes of inner and outer rings is established, which takes into account total errors of axes alignment resulted from all types of errors. Also parameter of *roughness* R_a is specified for surfaces of shafts and holes in housings.

Table 5.1 – Main types of loads for frictionless bearings rings depending on operation conditions

Operation conditions		Type of load of	
Radial load	Rotating ring	inner ring	outer ring
Constant by direction	Inner	Circulating	Local
	Outer	Local	Circulating
Constant by direction and rotating (of smaller value)	Inner	Circulating	Oscillating
	Outer	Oscillating	Circulating
Constant by direction and rotating (of bigger value)	Inner	Local	Circulating
	Outer	Circulating	Local
Constant by direction	Inner and outer rings in the same or opposite directions	Circulating	Circulating
Rotating with inner ring		Local	Circulating
Rotating with outer ring		Circulating	Local

Table 5.2 – Bearing operation conditions

Light	Normal	Heavy	Extra heavy*
Ratio of load ***F***, N, to dynamic load-carrying capacity ***C***, N			
$\frac{\boldsymbol{F}}{\boldsymbol{C}} < 0.07$	$0.07 \le \frac{\boldsymbol{F}}{\boldsymbol{C}} \le 0.15$	$0.15 < \frac{\boldsymbol{F}}{\boldsymbol{C}}$	
Service life, hours			
More than 10 thousands	5 – 10 thousands	2.5 – 5 thousands	–

*Operation at shock and vibration loads. Fits are the same as for heavy conditions.

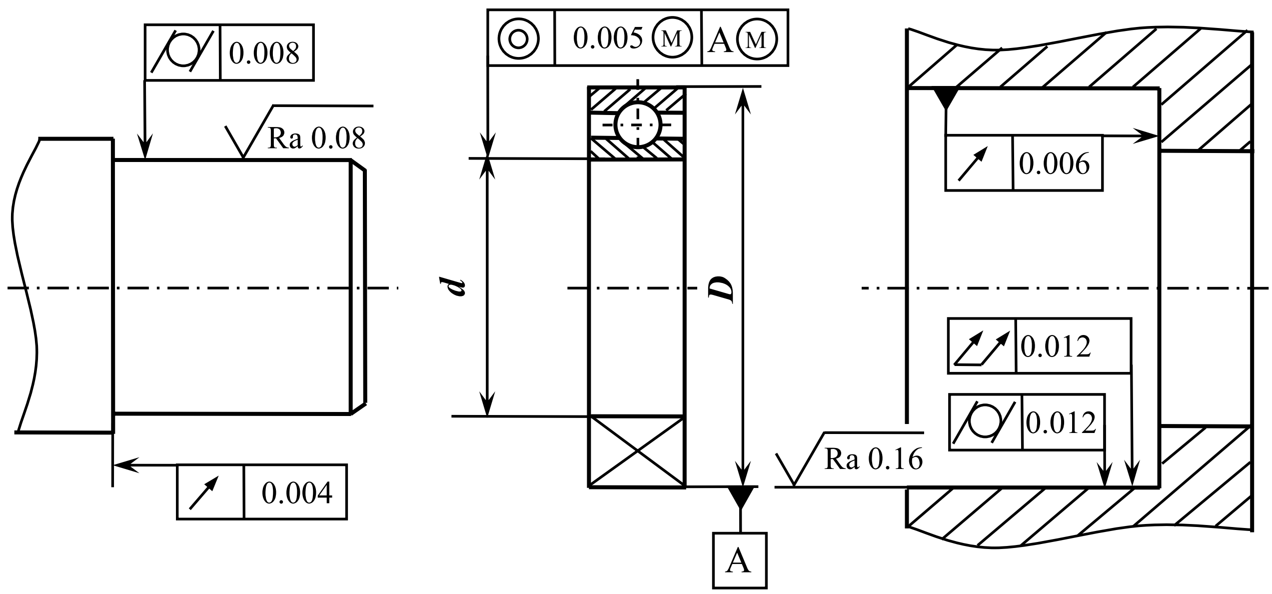


Figure 5.7 – Requirements for accuracy and surface finish of attachment surfaces of mating parts for a bearing mounting

6 DIMENSION CHAINS

For proper operation of machines or mechanisms it is necessary that their constituent parts and surfaces of these parts occupy definite positions relative each other that correspond to design purpose.

6.1 Fundamentals of theory of dimension chains

Dimension chain is a system of interconnected dimensions, which determine relative positions of surfaces of one or several parts and create close contour.

Dimensions included into dimension chain are called *links*, which are designated in diagrams with capital letters with indexes, for example, $\boldsymbol{A_1}$, $\boldsymbol{A_2}$, $\boldsymbol{A_3}$, $\boldsymbol{A_\Sigma}$ (Fig. 6.1).

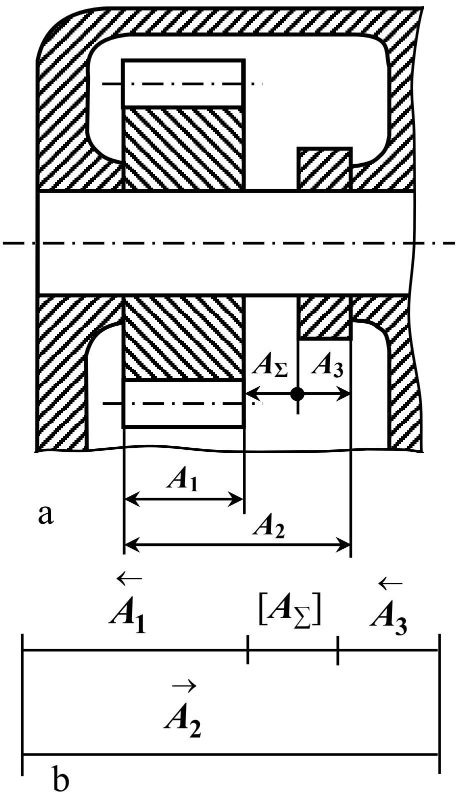


Figure 6.1 – Fragment of drawing (a) and diagram of dimension chain (b)

Dimension chains are divided to:

- Related to a part;
- Related to an assembly (ref. Fig. 6.1);
- Linear;
- Angular;
- Design;
- Manufacturing;
- Measuring (metrological).

Linear dimension chains are divided to:

- Spatial;
- Flat: with parallel links (collinear) (ref. Fig. 6.1) and with non-parallel links.

Links of dimension chain are divided to:

- Initial, $[\boldsymbol{A_\Sigma}]$;
- Closing, $\boldsymbol{A_\Sigma}$;
- Constituent links: increasing, $\overrightarrow{\boldsymbol{A_2}}$; decreasing, $\overleftarrow{\boldsymbol{A_1}}$, $\overleftarrow{\boldsymbol{A_3}}$ (ref. Fig. 6.1).

Constituent (increasing and decreasing) are called those links of dimension chain, change of which results in change of closing link $\boldsymbol{A_\Sigma}$, but can not and should not change initial link $[\boldsymbol{A_\Sigma}]$. Enlargement of increasing link enlarges value of closing link, and enlargement of decreasing link reduces value of closing link.

Initial $[\boldsymbol{A_\Sigma}]$ is called that link, to which basic requirement for accuracy is submitted and which determines quality of an item. *It is specified by designer nominal dimension* with tolerance, limit deviations, which should be ensured as a result of solution of dimension chain. This term (initial link) is used in *design calculations*.

In manufacturing process of machining or assembling of an item the initial link is obtained last and thus it closes the dimension chain. Therefore in technological calculations it is called *closing link* $\boldsymbol{A_\Sigma}$. This term (closing link) is used in *check calculations*.

Calculations and analysis of dimension chains is an obligatory stage of designing and manufacture of machines, which enhances providing of interchangeability, reduction of labour expenditures, quality improvements.

Essence of dimension chains calculations is in determination of nominal dimensions, tolerances and limit deviations for all links under the requirements of design or technology.

Completeness (closure) of dimension chain is a necessary condition for starting the calculations

$$\boldsymbol{A_\Sigma} = \sum_{1}^{m} \overrightarrow{\boldsymbol{A}}_j - \sum_{1}^{n} \overleftarrow{\boldsymbol{A}}_q ,$$

where $\overrightarrow{\boldsymbol{A}}_j$ – increasing links; $\overleftarrow{\boldsymbol{A}}_q$ – decreasing links; $\boldsymbol{m}$ – number of increasing links; $\boldsymbol{n}$ – number of decreasing links.

For example, for the diagram depicted in Fig. 6.1 the equation of dimension chain is

$$\boldsymbol{A_\Sigma} = \overrightarrow{\boldsymbol{A}}_2 - \left(\overleftarrow{\boldsymbol{A}}_1 + \overleftarrow{\boldsymbol{A}}_3 \right).$$

When calculating dimension chains 2 types of tasks are solved:

1) Determination of nominal dimension, tolerance and limit deviations for closing link $\boldsymbol{A_\Sigma}$ from the given nominal dimensions and limit deviations of constituent links – *check calculations*;

2) Determination of nominal dimension, tolerance and limit deviations for desired constituent link from other known dimensions of chain and given initial dimension [$\boldsymbol{A_\Sigma}$] – *design calculations.*

6.2 Procedure for design calculations of dimension chains

1. Plot the diagram of dimension chain from the drawing.
2. Find the initial link [$\boldsymbol{A_\Sigma}$]. Determine its nominal dimension, limit deviations, tolerance [$\boldsymbol{T_\Sigma}$], middle coordinate of tolerance band (mean deviation).
3. Find constituent links (increasing and decreasing).
4. Compile main equation of dimension chain (closeness equation), calculate nominal dimensions of all links $\boldsymbol{A_i}$ including the dependent link $\boldsymbol{A_X}$. When selecting a dependent link the condition of minimal production costs is taken into account.
5. Select method and technique to ensure the given accuracy of closing link $\boldsymbol{A_\Sigma}$.

Methods for ensuring a specified accuracy of closing link:

1) *Maximum-minimum method* – for complete interchangeability;

2) *Probabilistic method* – for incomplete interchangeability (for example, risk $P_\beta = 0.27$ % provides low production costs);

3) Other methods: fitting, adjustment, etc.

Techniques for ensuring a specified accuracy of closing link: a) *of equal tolerances*; b) *of equal accuracy grades*.

6. Calculate tolerance for each constituent link A_i:

a) *Technique of equal tolerances*: Calculate the mean value of tolerance T_{mi} for each constituent link:

1) Maximum-minimum method: $T_{mi} = \dfrac{[T_\Sigma]}{m+n}$;

2) Probabilistic method: $T_{mi} = \dfrac{[T_\Sigma]}{\sqrt{m+n}}$,

where $[T_\Sigma]$ – tolerance of initial link (specified in a drawing), m – number of increasing links; n – number of decreasing links.

Taking into account the calculated T_{mi} values assign the nearest standard tolerance values T_i for each constituent link A_i (except A_X) with determination of accuracy grades;

b) *Technique of equal accuracy grades*: Select values of tolerance units i_i for each constituent link A_i from the standard by their nominal dimensions. Determine mean level of accuracy for all constituent links with mean number $a_{m\,calc}$ of tolerance units:

1) Maximum-minimum method: $a_{m\,calc} = \dfrac{[T_\Sigma]}{\sum i_i}$;

2) Probabilistic method: $a_{m\,calc} = \dfrac{[T_\Sigma]}{\sqrt{\sum i_i^2}}$.

According to the calculations result the nearest standardised value a_m and respective accuracy grade are accepted (ref. Table 2.2). Tolerance T_i for each link A_i (including A_X) is selected from the standard by their nominal dimensions and accuracy grade.

7. Check validity of determined tolerances and perform corrections, if necessary. Tolerance of closing link T_Σ should be less or equal to the tolerance of initial link

$$T_\Sigma \le [T_\Sigma].$$

Tolerance of closing link is determined from the formulas:

- $T_\Sigma = \sum_{1}^{m+n} T_i$, when calculating with maximum-minimum method;

- $T_\Sigma = \sqrt{\sum_{1}^{m+n} T_i^2}$, when calculating with probabilistic method.

Corrections are made by assigning higher accuracy (smaller number of accuracy grade) and, hence, smaller tolerance value to one or more constituent links.

8. Assign tolerance bands for each constituent link $\boldsymbol{A_i}$ (except $\boldsymbol{A_X}$) according to the accuracy grade and type of dimension (for holes – H basic deviation, for shafts – h, for others – js).

9. From the standard for each constituent link $\boldsymbol{A_i}$ (except $\boldsymbol{A_X}$) determine limit deviations and calculate mean deviations

$$\Delta m_i = (\Delta S_i + \Delta I_i)/2,$$

where ΔS_i – upper deviation of A_i nominal dimension; ΔI_i – lower deviation.

10. Determine middle coordinate of tolerance band Δm_X of dependent link A_X from the formula

$$[\Delta m_\Sigma] = \sum_1^m \overrightarrow{\Delta m}_j - \sum_1^n \overleftarrow{\Delta m}_q ,$$

where $[\Delta m_\Sigma]$ – mean deviation of $[A_\Sigma]$ initial link.

Mean deviation Δm_X can be in the group of increasing links or in the group of deceasing links.

11. According to the nominal dimension A_X, accepted accuracy grade and tolerance T_X select in the standard the tolerance band with mean deviation $\Delta m_{ST} = (\Delta S_{ST} + \Delta I_{ST})/2$ nearest to the calculated deviation Δm_X.

12. In order to check correctness of the selection calculate mean deviation Δm_Σ of closing link with the Δm_{ST} selected value. Mean deviation Δm_Σ should be as near to the mean deviation $[\Delta m_\Sigma]$ of initial link as possible

$$\Delta m_\Sigma = \sum_1^m \overrightarrow{\Delta m}_j - \sum_1^n \overleftarrow{\Delta m}_q \rightarrow [\Delta m_\Sigma].$$

13. In order to check correctness of constituent links parameters calculate the upper ΔS_Σ and lower ΔI_Σ deviations of closing link and limit dimensions $A_{\Sigma max}$ and $A_{\Sigma min}$ of closing link:

$$\Delta S_\Sigma = \Delta m_\Sigma + 0.5T_\Sigma; \quad \Delta I_\Sigma = \Delta m_\Sigma - 0.5T_\Sigma;$$

$$A_{\Sigma max} = A_\Sigma + \Delta S_\Sigma; \quad A_{\Sigma min} = A_\Sigma + \Delta I_\Sigma,$$

where A_Σ – nominal dimension of closing link equals nominal dimension of initial link $[A_\Sigma]$.

If the calculated values satisfy conditions $A_{\Sigma max} \le [A_{\Sigma max}]$ and $A_{\Sigma min} \ge [A_{\Sigma min}]$, the calculation procedure is completed with selected parameters for all constituent links. If even one of these conditions is not observed, it is necessary to perform respective corrections in parameters of constituent links and, first of all, in parameters of the dependent link following with check calculations again.

When calculating dimension chains with *maximum-minimum method* only limit deviations of constituent links are taken into account. This method ensures complete interchangeability of parts and units. The method is economically reasonable for manufacture of items under small-batch and pilot production conditions, for the parts of low accuracy or for the chains including small number of links. In other cases the tolerances can be too severe and difficult for manufacture.

Probabilistic method is applied for manufacture of items under large-scale and mass production conditions. So as in the tuned and steady production process the random factors are dominant, it is assumed that dispersion of actual dimensions obeys to the normal distribution law (Gauss law). This means that the parts with mean dimensions come to the assembling in the quantity much larger than the quantity of parts with dimensions near to both limits. Oscillations of closing link dimension will be less as compared with maximum-minimum method. Probabilistic method allows greatly increase of tolerances of constituent links at the constant tolerance of closing link. For example, when calculating the mean tolerance $\boldsymbol{T_{mi}}$ with technique of equal tolerances (ref. Paragraph 6.2, Point 6) the probabilistic method allows several times increasing the tolerances for constituent links:

1) Maximum-minimum method: $\boldsymbol{T_{m\,i}} = \dfrac{[\boldsymbol{T_\Sigma}]}{\boldsymbol{m}+\boldsymbol{n}} = \dfrac{140}{3} \approx 46.7$ µm;

2) Probabilistic method: $\boldsymbol{T_{m\,i}} = \dfrac{[\boldsymbol{T_\Sigma}]}{\sqrt{\boldsymbol{m}+\boldsymbol{n}}} = \dfrac{140}{\sqrt{3}} \approx 80.8$ µm.

The effect becomes stronger with increase of number of constituent links.

7 INTERCHANGEABILITY OF THREADED JOINTS

Threaded joints are widely applied in machine-building industry and in aeronautics brunch (Fig. 7.1).

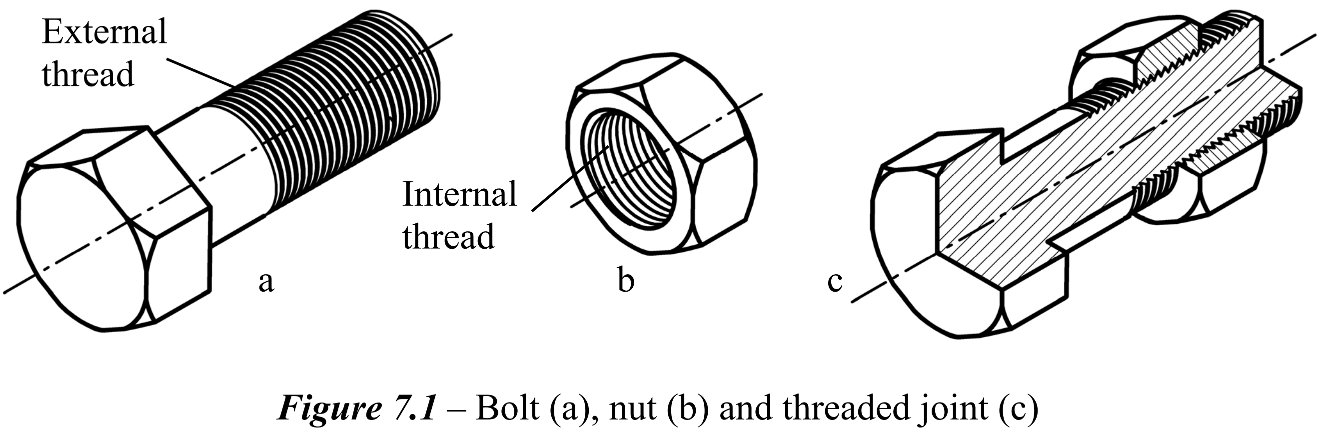


Figure 7.1 – Bolt (a), nut (b) and threaded joint (c)

Threads are divided into the following types by their application purpose:

- *Fastener threads* – metric and Whitworth threads – for detachable joints of parts;
- *Kinematic threads* – trapezoidal and square threads – ensure precision motions of mechanism parts. They are applied, for example, for feed screws of turning machines;
- *Buttress threads* – transform rotary motion into linear motion. They are applied in presses and screw-jacks. They provide smoothness of motion at high load-carrying capacity;
- *Pipe threads* (*fitting threads*) – provide pressure-tightness of joints.

7.1 Main parameters of threads

Types of threads by profile and their symbols are: M – metric, cylindrical; M_K – metric, taper (conical); Tr – trapezoidal; S – buttress; S×45° – reinforced; Rd – round; G – pipe.

Metric cylindrical threads are mostly widely spread in industry, and so parameters of metric threads and accuracy of the joints are considered in this chapter (Fig. 7.2).

Initial parameters of threads (Table 7.1) are:

- ***d*** – nominal diameter, equals major diameter of thread;
- ***P*** – pitch of thread.

Various pitches (one coarse and several fine pitches) are standardised for one thread diameter. Selection of a thread pitch is performed from design conditions.

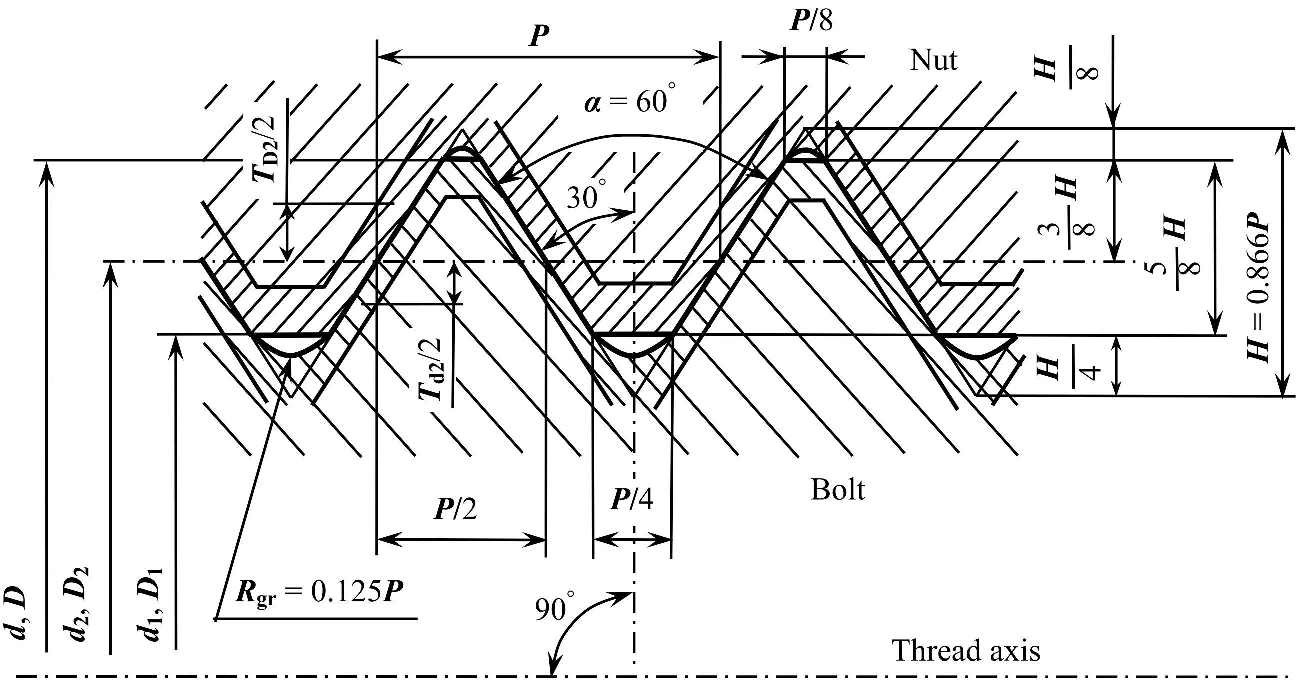


Figure 7.2 – Nominal profile of metric thread and tolerance bands of threads in internal thread (nut) and external thread (bolt) for H/h fit

Table 7.1 – Main parameters of threads

d, D	d_2, D_2	d_1, D_1	α	P
Major diameter, mm	Pitch diameter, mm	Minor diameter, mm	Angle of thread, degree	Pitch, mm

Examples of the threads designations are:

- M24: M – metric type of thread; 24 – major diameter. If pitch value is not specified, then the pitch is coarse (only one value for thread diameter);
- M24×1.5: symbols are the same, except 1.5 – pitch of fine-pitch thread (1.5 mm);
- M24×3(P1)LH: symbols are the same, except 3 – three-start thread; (P1) – fine pitch of 1 mm; LH – left-hand thread (right-hand thread is not shown in designation).

7.2 Threaded fits with clearance

Basic line for limit deviations of thread is its nominal profile. Basic deviation that determines position of tolerance band relative to nominal contour (that is, the nearest to it) is ***es*** for external thread (bolt) and ***EI*** for internal thread (nut).

Deviation is measured from nominal profile in the direction perpendicular to the thread axis.

Critical for a thread joint is a character of contact along sides of a thread, that is, along pitch diameter. Therefore, fits along major diameter d and minor diameter d_1 can differ from fits along pitch diameter d_2 (Table 7.2).

Table 7.2 – Thread standardised degrees of accuracy and basic deviations

Thread type	Thread diameter	Accuracy degree	Basic deviations
External thread (bolt)	d – major	4, 6, 8	d, e, f, $\boxed{\text{g}}$, h
	d_2 – pitch	3, 4, 5, 6, 7, 8, 9, 10*	
Internal thread (nut)	D_2 – pitch	4, 5, 6, 7, 8, 9*	(E), (F), G, $\boxed{\text{H}}$
	D_1 – minor	4, 5, 6, 7, 8	
Notes. 1. Accuracy of minor diameter of bolt thread d_1 and major diameter of nut thread D are not standardised. Basic deviations for them are selected equal to the basic deviations for pitch diameters of bolt and nut respectively. 2. Basic deviations and accuracy degrees marked: with frame are preferred; with round brackets are of special purpose; with asterisk are only for threads in plastic parts.			

Thread tolerance band is determined by combination of tolerance bands of diameters d, d_1, d_2 (Fig. 7.3).

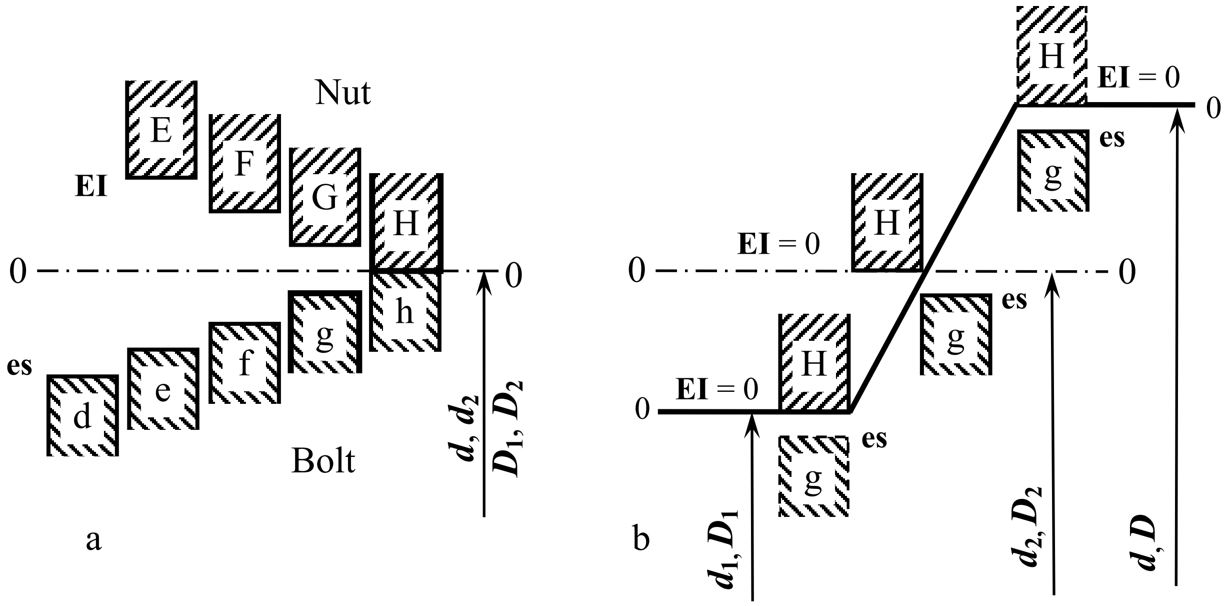


Figure 7.3 – Basic deviations for fits of bolt and nut with clearance (a) and preferred basic deviations along thread profile (b)

Tolerance band of certain thread diameter is designated by combination of digit (number of accuracy degree) and letter (basic deviation). In contrast to slick joints designation of thread accuracy starts from number of accuracy degree, and letter is put after the number, for example:

- 7G: 7 – accuracy degree of nut, G – basic deviation, tolerance bands 7G are equal for pitch $\boldsymbol{D_2}$ and minor $\boldsymbol{D_1}$ diameters;
- 6h7h – accuracy of bolt along pitch diameter $\boldsymbol{d_2}$ (6h) and major diameter $\boldsymbol{d}$ (7h);
- 5H6H – accuracy of nut along pitch diameter $\boldsymbol{D_2}$ (5H) and minor diameter $\boldsymbol{D_1}$ (6H).

Values of basic deviations are calculated from the standardised formulas, for example:

- For bolts with basic deviation “g”

$$\boldsymbol{es}_{\mathbf{g}} = -\,(15 + 11\cdot\boldsymbol{P}),\ \mu\text{m};$$

- For nuts with basic deviation “G”

$$\boldsymbol{EI}_{\mathbf{G}} = +\,(15 + 11\cdot\boldsymbol{P}),\ \mu\text{m},$$

where $\boldsymbol{P}$ – thread pitch specified in mm.

Series of the 6th accuracy degree for bolts is adopted as *basic series of tolerances* for all diameters. The tolerance value is determined for the bolt pitch diameter $\boldsymbol{d_2}$ from the formula

$$\boldsymbol{T}_{\mathbf{d2}}(6) = 90\,\boldsymbol{P}^{0.4}\boldsymbol{d}^{0.1},\ \mu\text{m},$$

where $\boldsymbol{d} = \sqrt{\boldsymbol{d}_{\boldsymbol{min}}\cdot\boldsymbol{d}_{\boldsymbol{max}}}$ – geometric average value of bolt major diameter;

$\boldsymbol{d_{min}}$ and $\boldsymbol{d_{max}}$ – minimum and maximum values of basic interval of thread diameters; $\boldsymbol{P}$ and $\boldsymbol{d}$ are specified in mm.

Tolerances for other accuracy degrees are determined by multiplication of the 6th degree tolerances by the coefficients obtained from the series of preferred numbers R10 = $\sqrt[10]{10} \approx 1.25$ (Table 7.3).

Table 7.3 – Accuracy classes, coefficients and degrees of accuracy for threads

Accuracy class	Coefficient from series of preferred numbers R10	Accuracy degree
Precise	0.5	3
	0.63	4
	0.8	5
Middle	1.0	6
Rough	1.25	7
	1.6	8
	2.0	9
	2.5	10

Tolerances for all *threaded holes* of all accuracy degrees are calculated from the formula

$$\boldsymbol{T_{D2}} = 1.32 \cdot \boldsymbol{T_{d2}},$$

when nut and bolt are of the same accuracy degree.

Limit deviations and tolerances of threads, generally, are related to screwing length. Threads are divided by *screwing length*:

- L – long – more than $6.7 \cdot \boldsymbol{Pd}^{0.2}$;
- N – normal – not less than $2.24 \cdot \boldsymbol{Pd}^{0.2}$ and not more than $6.7 \cdot \boldsymbol{Pd}^{0.2}$;
- S – short – less than $2.24 \cdot \boldsymbol{Pd}^{0.2}$.

Screwing length, to which the thread tolerance is related to, should be specified in millimetres in a thread designation in the following cases: 1) if it is related to the group L; 2) if it is related to the group S, when screwing length is less than whole length of thread. For example, designation of threaded joint with non-standardized screwing length

M12×1.5–7H/7g6g–24,

where M – metric thread; 12 – nominal (major) thread diameter, mm; 1.5 – pitch of fine thread, mm; 7H – tolerance bands of nut thread along pitch diameter $\boldsymbol{D_2}$ and minor diameter $\boldsymbol{D_1}$ (the same 7H); 7g6g – tolerance bands of bolt thread along pitch diameter $\boldsymbol{d_2}$ (7g) and major diameter $\boldsymbol{d}$ (6g); 24 – non-standardised screwing length, mm.

7.3 Threaded fits with interference

Fits with interference are applied in fixed threaded joints (stud in a housing hole). Immobility and strength of a joint is ensured by interference fits at the expense of interference along pitch diameter (Table 7.4, Fig. 7.4).

Table 7.4 – Basic deviations and accuracy degrees of thread diameters for fits with interference

Thread diameter	Basic deviation at pitch P equal to		Accuracy degree
	Up to 1.25 mm	Above 1.25 mm	
d	e	c	6
d_2	n, p, r		2; 3
D	H		–
D_2			2
D_1	D	C	4; 5

The standard states interference fits only in the *system of hole* (H) having large production advantages in comparison with the system of shaft. Tolerance band for minor diameter d_1 of outside thread (stud, bolt) are not stated by the standard. It states only upper deviation es_1 equal the upper deviation es_2 of tolerance band for pitch diameter d_2. Also the upper deviation ES of major diameter D of internal thread (nut) is not stated.

Interference fits are typically performed with group assembling to provide the guaranteed interference N_2 along pitch diameter d_2 (ref. Fig. 7.4). Quantity of selection groups is equal to 2 for the 3p tolerance band and 3 for the 3n tolerance band. Example of designation of the fixed threaded joint is

$$\mathrm{M16} - \frac{2\mathrm{H}\,4\mathrm{C}(3)}{3\mathrm{n}(3)},$$

where M – metric thread; 16 – nomi-

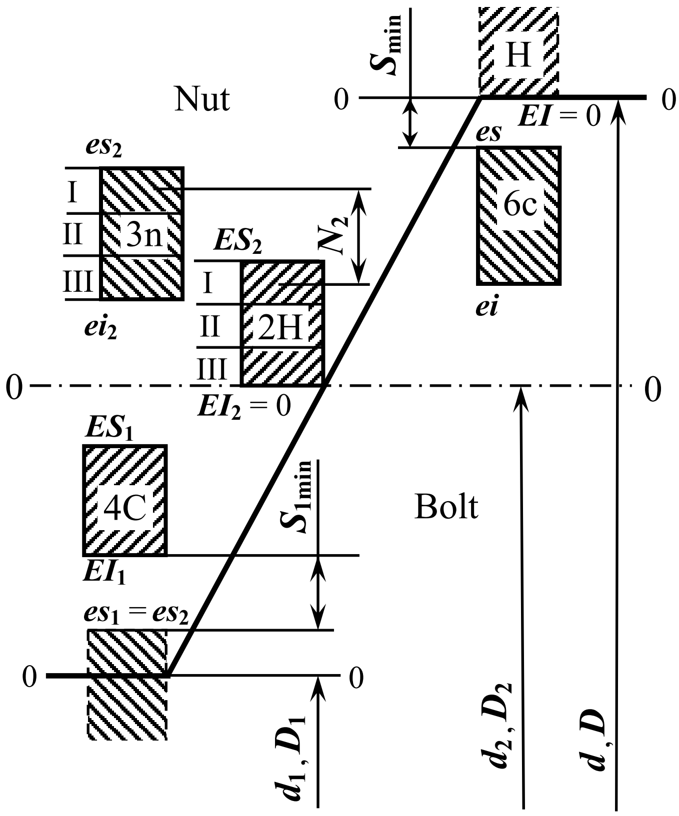


Figure 7.4 – Diagram of tolerance bands for threaded transition fit $\frac{2\mathrm{H}\,4\mathrm{C}(3)}{3\mathrm{n}(3)}$ with interference and group assembling

nal (major) thread diameter, mm; pitch of coarse thread (2 mm) is not shown; 2H – tolerance band of pitch diameter $\boldsymbol{D_2}$ in housing (hole); 4C – tolerance band of minor diameter $\boldsymbol{D_1}$ in housing; 3n – tolerance band of stud pitch diameter $\boldsymbol{d_2}$; tolerance band (6c for pitch more than 1.25 mm) of stud major diameter $\boldsymbol{d}$ is not shown; (3) – number of selection groups (group assembling). Data for nut (housing hole) is given in numerator, for bolt (stud) – in denominator.

When making assembling with a selective method, the bolts from selection group I go in fits with the nuts from respective group I, bolts of the II group are coupled with nuts of the II group, respectively bolts of the III group – the III-group nuts (ref. Fig. 7.4). In the example considered above the interference is guaranteed along pitch diameter at typical transition fits (when both interference and clearance are possible) due to application of selective method.

7.4 Errors of thread machining

Errors of thread machining are determined by deviations of its geometric parameters: $\boldsymbol{d}$ ($\boldsymbol{D}$), $\boldsymbol{d_2}$($\boldsymbol{D_2}$), $\boldsymbol{d_1}$($\boldsymbol{D_1}$), $\boldsymbol{P}$, $\boldsymbol{\alpha}/2$, as well as thread groove radii of bolt and nut and form of side surfaces of thread.

7.4.1 Errors of pitch

Deviation of thread pitch $\boldsymbol{\Delta P}_{\mathrm{n}}$ consists of progressing pitch errors $\boldsymbol{\Delta P}$ proportional to number $\boldsymbol{n}$ of threads along screwing length (Fig. 7.5). The errors appear because of kinematic error of a machine (pitch inaccuracy of a machine's feed screw). From geometry analysis the diametrical compensation of pitch error is

$$f_P = ctg\frac{\alpha}{2}\cdot \Delta P_n, \ \mu\text{m},$$

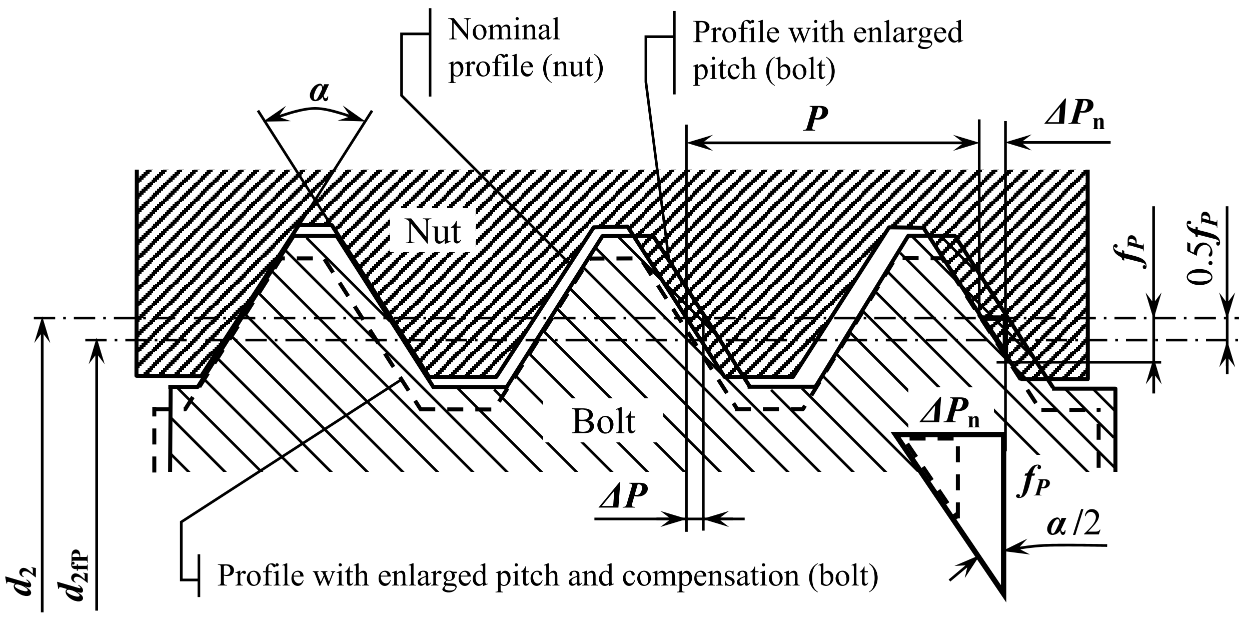


Figure 7.5 – Thread pitch error $\boldsymbol{\Delta P}_{\mathrm{n}}$ and its diametrical compensation $\boldsymbol{f_P}$

where ΔP_n is specified in micrometers.

At $\alpha = 60^\circ$ for metric thread the compensation is

$$f_P = 1.732 \cdot \Delta P_n.$$

7.4.2 Errors of half-angle of profile

Deviations of half-angle of thread profile $\Delta\alpha/2$ can be a result of skewness of thread profile relative to the axis of a part or profile deviations (Fig. 7.6).

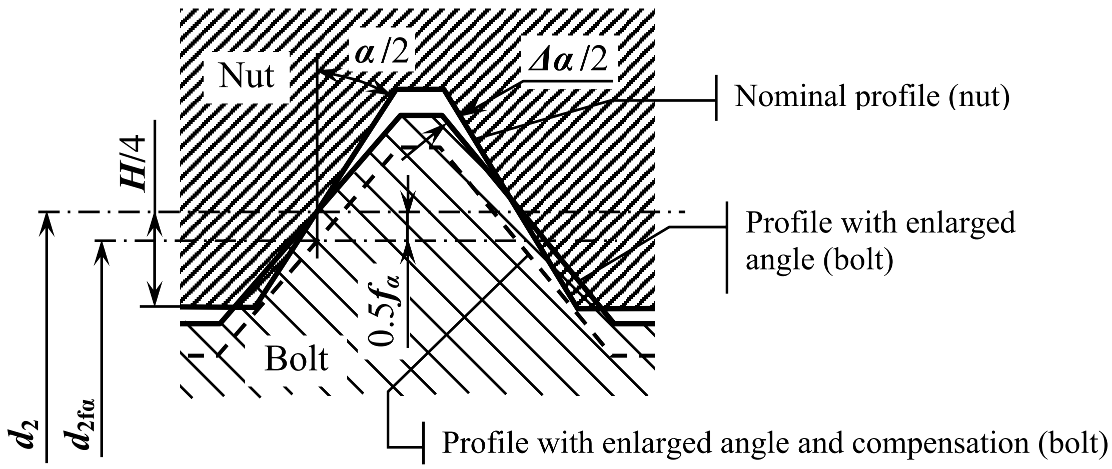


Figure 7.6 – Thread profile half-angle $\Delta\alpha/2$ error and its diametrical compensation f_α

From geometry analysis the diametrical compensation for half-angle error of metric thread is

$$f_\alpha = 0.36\, P \frac{\Delta\alpha}{2}, \ \mu\text{m},$$

where angle $\Delta\alpha/2$ – average angle error of thread sides determined as a half-sum of absolute error values on left and right sides of thread profile, specified in minutes; P – pitch specified in millimetres.

7.4.3 Conditions for quality and screwing

Errors of pitch diameter Δd_2 (ΔD_2) itself at machining of a thread typically correspond to the tolerances of 8th and 9th accuracy grades of slick joints.

In a result the total diametrical influence of three types of errors must not exceed limits of the standard tolerance for pitch diameter:

- For bolt $T_{d2} = \Delta d_2 + f_P + f_\alpha$;
- For nut $T_{D2} = \Delta D_2 + f_P + f_\alpha$.

The *reduced* pitch diameters for external and internal threads (Fig. 7.7) are calculated from the formulas:

- For bolt $d_{2R} = d_{2A} + f_P + f_\alpha$;
- For nut $D_{2R} = D_{2A} - (f_P + f_\alpha)$,

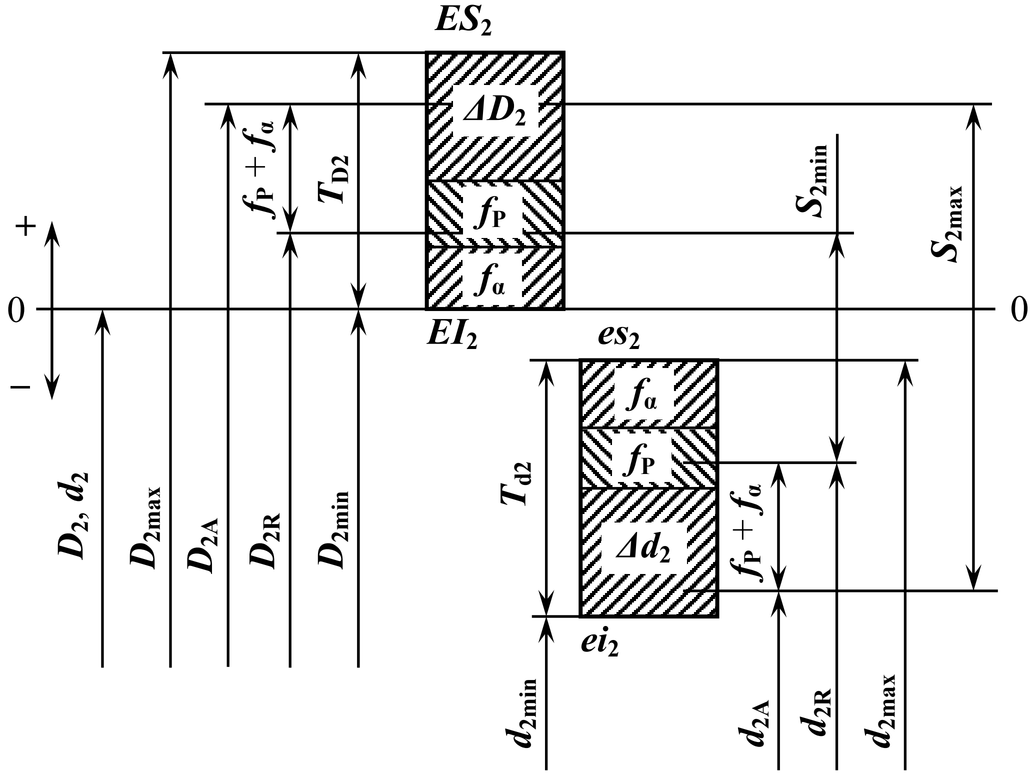


Figure 7.7 – Illustration to the conditions of screwing of bolt and nut

where $\boldsymbol{d_{2A}}$ – actual (measured) pitch diameter of bolt; $\boldsymbol{D_{2A}}$ – actual (measured) pitch diameter of nut.

Using these parameters the *conditions for quality of threads* can be formulated as (ref. Fig. 7.7):

- For bolt: $\boldsymbol{d_{2A} \geq d_{2min}}$ and $\boldsymbol{d_{2R} \leq d_{2max}}$;

- For nut: $\boldsymbol{D_{2A} \leq D_{2max}}$ and $\boldsymbol{D_{2R} \geq D_{2min}}$,

where $\boldsymbol{d_{2min} = d_2 + ei_2}$ – minimum pitch diameter of bolt; $\boldsymbol{d_{2max} = d_2 + es_2}$ – maximum pitch diameter of bolt; $\boldsymbol{D_{2min} = D_2 + EI_2}$ – minimum pitch diameter of nut; $\boldsymbol{D_{2max} = D_2 + ES_2}$ – maximum pitch diameter of nut.

The *condition for screwing of bolt and nut* is

$$\boldsymbol{D_{2R} \geq d_{2R}}.$$

In threaded joint the minimum clearance $\boldsymbol{S_{2min}}$ will be at the presence of pitch and half-angle errors:

$$\boldsymbol{S_{2min} = D_{2R} - d_{2R}},$$

and the maximum clearance $\boldsymbol{S_{2max}}$ will be when the pitch and half-angle errors are absent (equal zero):

$$\boldsymbol{S_{2max} = D_{2A} - d_{2A}}.$$

For more careful determination of pitch diameter compensations it is necessary to consider thread errors listed at the beginning of Paragraph 7.4.

7.5 Check of metric threads with gauges

Check of tread quality may be performed by elements or in complex. If errors $\boldsymbol{\Delta P_n}$ and $\boldsymbol{\Delta\alpha}/2$ are limited by tolerance of pitch diameter, then reduced pitch diameter is calculated and conclusion on quality is made. This method is not rational and under production conditions a thread is checked in complex – by thread gauges.

Major diameter $\boldsymbol{d}$ of bolts and minor diameter $\boldsymbol{D}$ of nuts are checked with ordinary *plain gauges*.

Quality control of the rest elements is performed by GO and NOT-GO thread gauges. Term “go” means that a controlled quality part is screwed easily with a GO gauge, ant term “not-go” means that a controlled quality part is not screwed with a NOT-GO gauge.

According to the Taylor’s principle *thread go gauges are replicas* of mating parts and have full profile and normal screwing length.

Go-gauges check assembling ability of parts, and so they should limit the dimension of reduced pitch diameter ($\boldsymbol{d_{2R}}$, $\boldsymbol{D_{2R}}$) and actual major ($\boldsymbol{d_A}$, $\boldsymbol{D_A}$) and minor ($\boldsymbol{d_{1A}}$, $\boldsymbol{D_{1A}}$) diameters. In order to check three diameters and thread half-angle ($\boldsymbol{\alpha}/2$), a go gauge should have thread of full profile (Fig. 7.8, positions 1 and 2), and to check all pitch $\boldsymbol{P}$ errors – work thread length of not less than 0.8 screwing length. In essence, go-gauges should be made according to nominal thread profile and be by shape the largest quality nut and the smallest quality bolt.

Not-go gauges check only quality of actual pitch diameter ($\boldsymbol{d_{2A}}$, $\boldsymbol{D_{2A}}$). In order to diminish influence of pitch errors, not-go gauges have a shortened length (2.5–3 turns of thread), and to diminish influence of errors of thread half-angle they have a shortened tread depth and widened groove. Due to this feature the side surfaces of gauge thread contact with thread turns of a part only along narrow band near to pitch diameter (ref. Fig. 7.8, position 3). Rules for gauges applications allow screwing of a not-go gauge with a quality checked part up to 2 turns.

Work gauges for check of bolts (ref. Fig. 7.9,a) have the similar geometry configurations (profiles) and control surfaces (mirror-like reflected contours) like work gauges for nuts (ref. Fig. 7.8,a). Thread ring gauges and adjustable snaps (in the shape of chasers or rollers) are used for check of a bolt thread.

Tolerance bands for thread gauges (ref. Figures 7.8,b and 7.9,b) are constructed like for plain gauges, but separately for each of three diameters. There are several tolerance bands, which are designed not for check of parts dimensions, but only for production of gauges (symbolised with words “No check”), because these surfaces of gauges are not in contact with parts surfaces to be checked.

As distinct from threaded parts, tolerances for thread gauges are stated separately for each of five thread parameters (ref. Table 7.1).

System of gauges includes *work* plain and thread go (GO) and not-go (NOT-GO) gauges (ref. Figures 7.8 and 7.9) and *counter-gauges* (RGO-GO, RNOT-GO, RNOT-NOT, RW-NOT, S-NOT, S-GO) for check (R) and setting (S) of work thread snaps and rings.

Setting gauges S-NOT and S-GO are applied only in those cases, when work thread snaps and rings are of adjustable type.

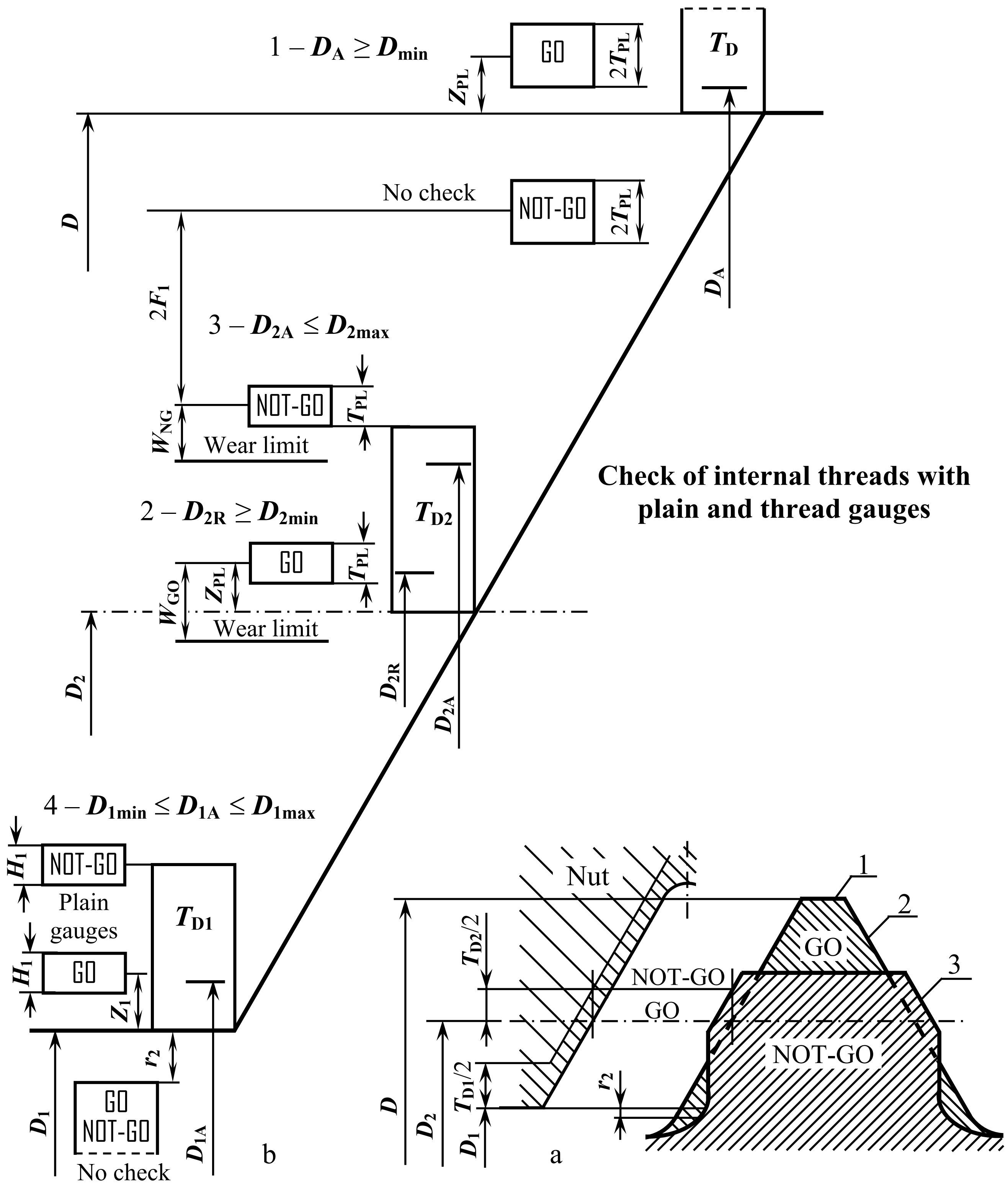


Figure 7.8 – Profiles of GO and NOT-GO work thread plug gauges for a nut (a), locations of tolerance bands for work gauges (b) and quality conditions checked by control surfaces of gauges: 1 – $D_A \geq D_{min}$ (GO gauge); 2 – $D_{2R} \geq D_{2min}$ (GO gauge); 3 – $D_{2A} \leq D_{2max}$ (NOT-GO gauge); 4 – $D_{1min} \leq D_{1A} \leq D_{1max}$ (plain GO and NOT-GO gauges)

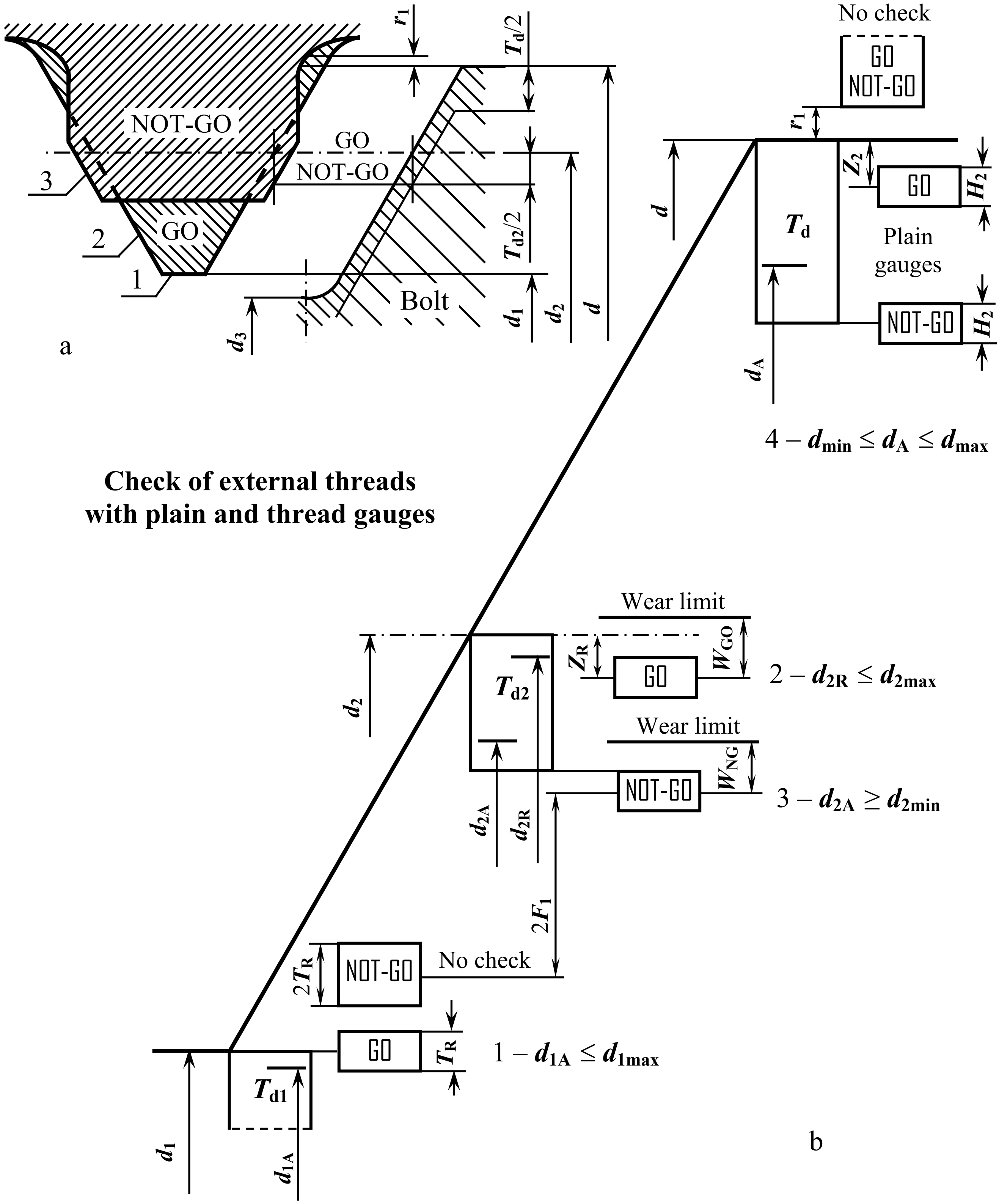


Figure 7.9 – Profiles of GO and NOT-GO work thread snap gauges for a bolt (a), locations of tolerance bands for work gauges (b) and quality conditions checked by control surfaces of gauges: 1 – $\boldsymbol{d_{1A}} \le \boldsymbol{d_{1max}}$ (GO gauge); 2 – $\boldsymbol{d_{2R}} \le \boldsymbol{d_{2max}}$ (GO gauge); 3 – $\boldsymbol{d_{2A}} \ge \boldsymbol{d_{2min}}$ (NOT-GO gauge); 4 – $\boldsymbol{d_{min}} \le \boldsymbol{d_A} \le \boldsymbol{d_{max}}$ (plain GO and NOT-GO gauges)

8 INTERCHANGEABILITY OF TOOTHED GEARINGS

8.1 Types and parameters of toothed gears

Toothed gearings are widely applied in modern machines and aeronautical equipment.

By purpose toothed gearings are divided into:

- *Measurement gearings* – provide high kinematic precision, that is, coordination of rotation;
- *High-speed gearings* – operate smoothly and noiselessly at high rotation speeds;
- *Power gearings* – transmit large forces, large teeth contact patch should be ensured;
- *General-purpose gearings* – gearings without high requirements for accuracy.

By position and shape of teeth gearings are divided into cylindrical (spur), helical, herringbone, and with curvilinear teeth. *Involute cylindrical gearings* with basic-rack profile angle $\boldsymbol{\alpha} = 20^{\circ}$ (Fig. 8.1) are mostly spread in industry.

The basic geometric parameters of spur gear are:

- Base circle diameter d_b;
- Pitch circle diameter $d = m \cdot z = P \cdot z / \pi$;
- Pitch along pitch circle $P = \pi d/z$;
- Teeth number (quantity) z;
- Module $m = d / z = P/\pi$;
- Tooth thickness $s = P/2$;
- Space width $e = P/2$;
- Tooth addendum height $h_a = m$;
- Tooth dedendum height $h_f = 1.25m$.

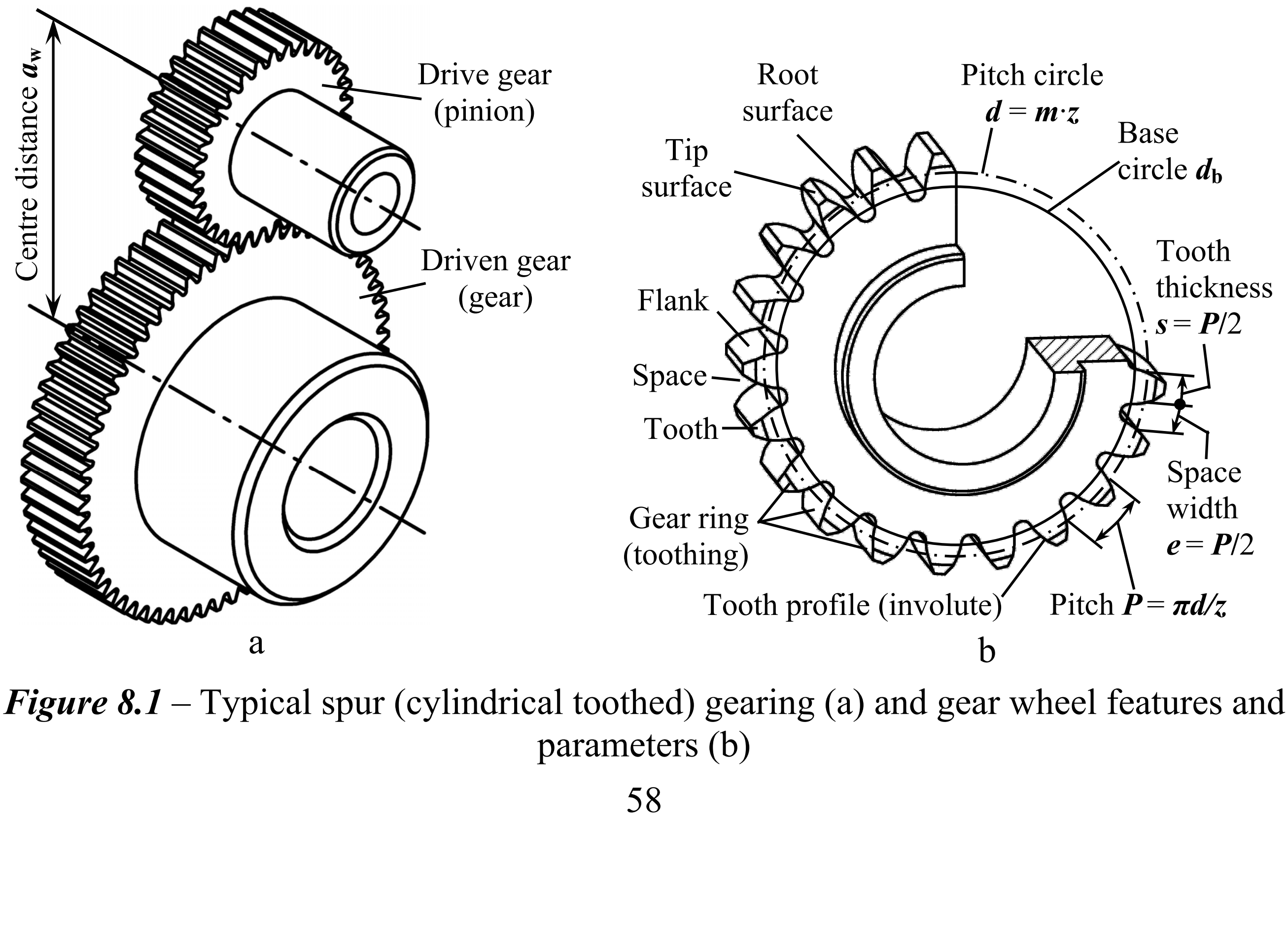


Figure 8.1 – Typical spur (cylindrical toothed) gearing (a) and gear wheel features and parameters (b)

8.2 Accuracy standardisation of toothed gears

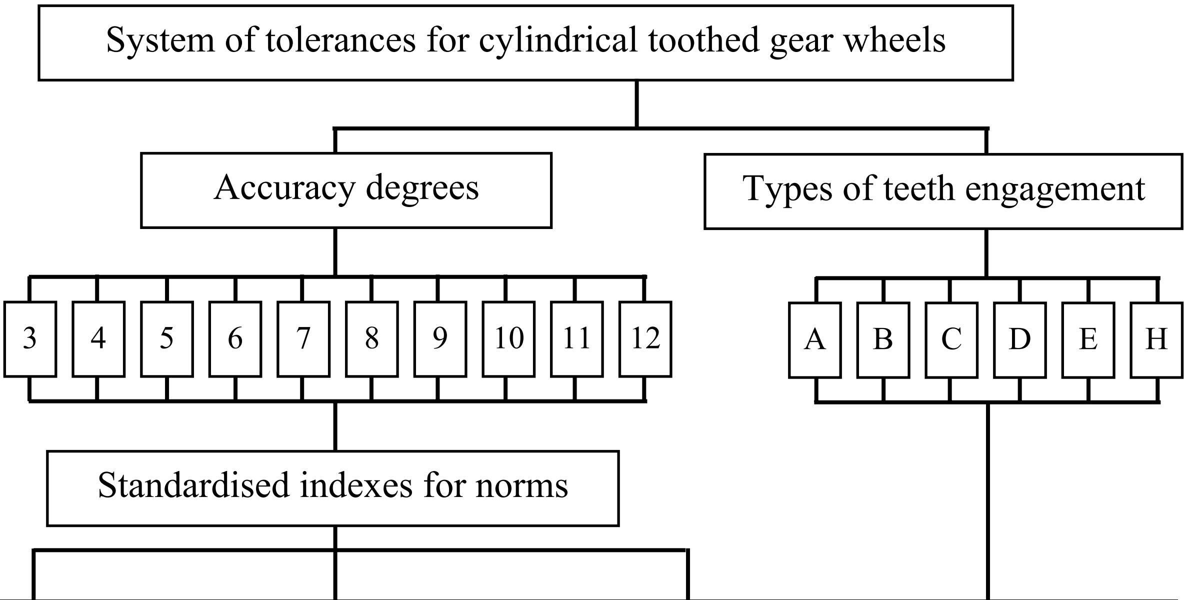
By accuracy of manufacture toothed gears are divided into *12 accuracy degrees* (Fig. 8.2). For the highest accuracy degrees (1 and 2) tolerances and deviations are not stated, because they are planned for future development.

All tolerances are specified for the 6^{th} accuracy degree. Tolerance values for other degrees are determined by multiplication with converting coefficient.

Combined action of errors caused by inaccuracies of profiles of tooth-cutting tools, their mounting in machines, deviations of dimensions and shape of blank, inaccuracy of their mounting and processing in machines, inaccuracies in kinematic links of machines result in errors of gear wheels. These errors are limited by 3 norms of accuracy (ref. Fig. 8.2):

1) *Kinematic accuracy norm*;
2) *Norm of operation smoothness*;
3) *Norm of contact patch* between teeth in a gearing.

Symbols for normalised deviations (tolerances) for all types of toothed gearings are the following: ***F*** – kinematic accuracy; ***f*** – operation smoothness; o – symbol of toothed



Kinematic accuracy	**Operation smoothness**	**Teeth contact patch**	**Indexes of circum-ferential backlash**
For gearing	For gearing	For gearing	For gearing
F'_{io} (accuracy degrees 3 to 8)	f'_{io}, f_{zko}, f_{zzo} (degrees 3 to 8)	Instant contact patch, Total contact patch, f_x, f_y (degrees 3 to 12)	f_a – adjustable CD $j_{n.min}$ – non-adjust-able CD
For wheel	For wheel	For wheel	For wheel
F'_i, F''_i, F_c, F_r, F_{vW}, F_{Pk}, F_P	f'_i, f''_i, f_{zk}, f_{zz}, f_{Pb}, f_{Pt}, f_f, ,	Instant contact patch, Total contact patch, F_β, F_{Pxn}	E_{Hs}, E_{Wms}, E_{cs}, T_H, T_{Wm}, E_{Ws}, $E_{a''s}$

Figure 8.2 – System of tolerances for cylindrical gearings

gearing; ' – single tooth contact; '' – double tooth contact.

The requirements for circumferential backlash between non-contact surfaces of teeth in assembled gearing, gathered in the *norm of circumferential backlash*, are specified independently to manufacturing accuracy. Six types of engagements (A, B, C, D, E, H) determining the guaranteed circumferential backlashes $j_{n.min}$, 8 types of tolerances for backlash (x, y, z, a, b, c, d, h) and 6 classes for centre distance (CD) deviations (and deviations $\pm f_a$ themselves) are established (Fig. 8.3, Table 8.1).

In a gearing with single tooth contact a circumferential backlash j_n should be between non-work (non-contacting) flanks of teeth (ref. Fig. 8.3)

$$j_n = j_{n.min} + k_j,$$

where k_j – compensation backlash necessary for manufacture of gears and mounting of a gearing. It includes random errors of manufacturing and assembling.

Minimum guaranteed circumferential backlash $j_{n.min}$ is necessary to avoid possible cramping (seizure) at warming-up of a gearing and to ensure lubrication conditions.

The minimum guaranteed circumferential backlash $j_{n.min}$ and CD deviations $\pm f_a$ are selected depending on CD absolute value a_w.

Example of designation of toothed gearing accuracy for combination of parameters

8-6-7-Ca/V-128 DSTU 1643–81,

where 8-6-7 – accuracy degrees for kinematic, operation smoothness, teeth contact norms respectively; C – type of engagement (ref. Table 8.1); a – type of tolerance for circumferential backlash; V – accuracy class of centre distance deviations; 128 – reduced designed circumferential backlash, μm, $j^{red}_{n.min} = j_{n.min} - 0.68\ (|f_a^l| - |f_a|)$; f_a^l – deviation of lower accuracy.

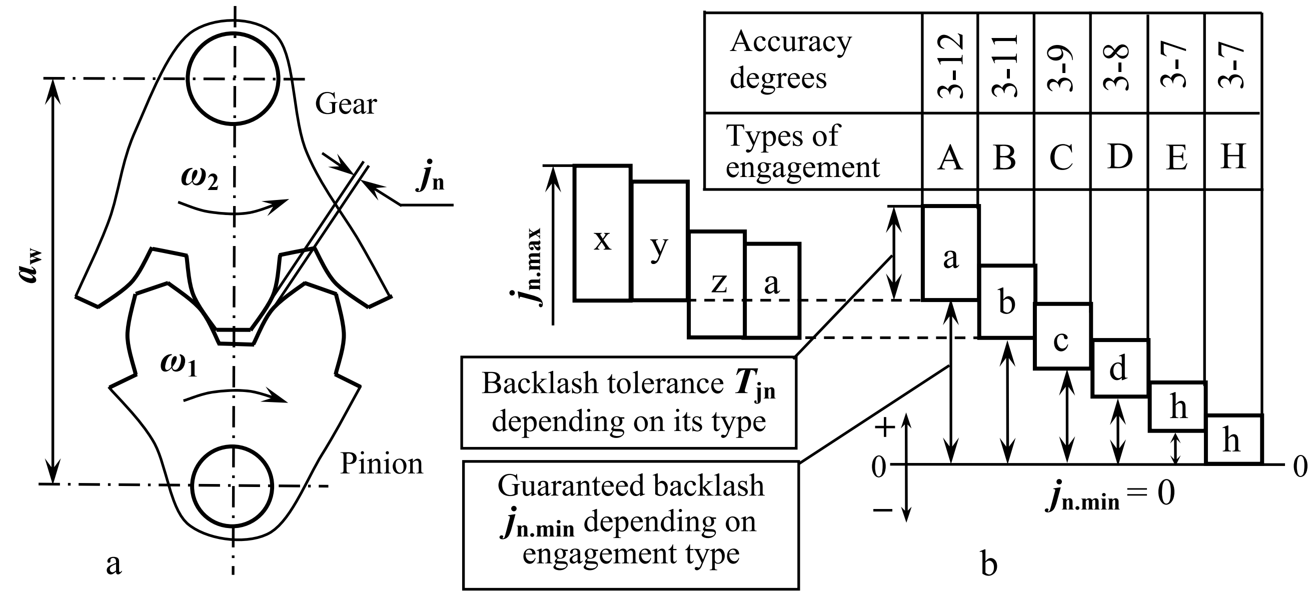


Figure 8.3 – Circumferential backlash j_n (a) and types of engagement and backlash tolerances (b) in a gearing

Table 8.1 – Tolerances for circumferential backlash depending on types of engagement and accuracy of centre distance (CD)

Parameters	Values						
Type of gears engagement	A	B	C	D	E	H	–
Type of tolerance T_{jn} for backlash (x, y, z)	a	b	c	d	h		–
Accuracy class of CD deviations	VI	V	IV	III	II	II	(I)
CD deviations, $f_a = \pm 0.5\, j_{n.min}$	$\pm\frac{IT11}{2}$	$\pm\frac{IT10}{2}$	$\pm\frac{IT9}{2}$	$\pm\frac{IT8}{2}$	$\pm\frac{IT7}{2}$		$\left(\pm\frac{IT6}{2}\right)$
Guaranteed backlash, $j_{n.min}$	IT11	IT10	IT9	IT8	IT7	0	–
Minimum additional modification shift of basic rack, E_{Hs}	–IT11	–IT10	–IT9	–IT8	–IT7	–0.4·IT7	–
Tolerance for additional addendum modification shift of basic rack, T_H	$T_H > F_r$ (tolerance for radial motion of toothing)						

Note. Class (I) is not recommended for application.

If, for example, accuracy degree "7" for all accuracy norms, type of tolerance for circumferential backlash "c" and CD (centre distance) accuracy class "IV" are specified according to type of gears engagement "C" (ref. Table 8.1, column C), the designation will be simple:

7-C DSTU 1643–81.

8.3 Measurements of gearing accuracy parameters

Most of measurements of cylindrical gearing are performed in special equipment for combined control (Fig. 8.4-a). Drive and driven gear wheels are in single tooth contact, that is, contact is on one side (flank) of a tooth, and clearance (backlash) is on another side. High accuracy (reference) rotation is supplied by friction discs, diameters of which are strictly equal to pitch diameters of drive and driven wheels.

At rotation of a drive wheel the friction couple rotates too. Rotation mismatch (error) between driven friction disc and driven gear wheel is recorded by a measuring device. The device is fixed on pitch diameter of driven wheel.

Spindle of driven friction disc reproduces reference rotation and is displaced in such a manner that the errors are measured on pitch diameter. With such a principle the mismatch between actual φ_2 and nominal φ_3 rotation angles of driven wheel are measured. The mismatch is calculated as an arc length of pitch circle. Cycle of measurements of relative position of gear wheels is complete when the first tooth of driven wheel will come in contact again with the first tooth of drive wheel. At further rotation the shapes of curves will be reproduced.

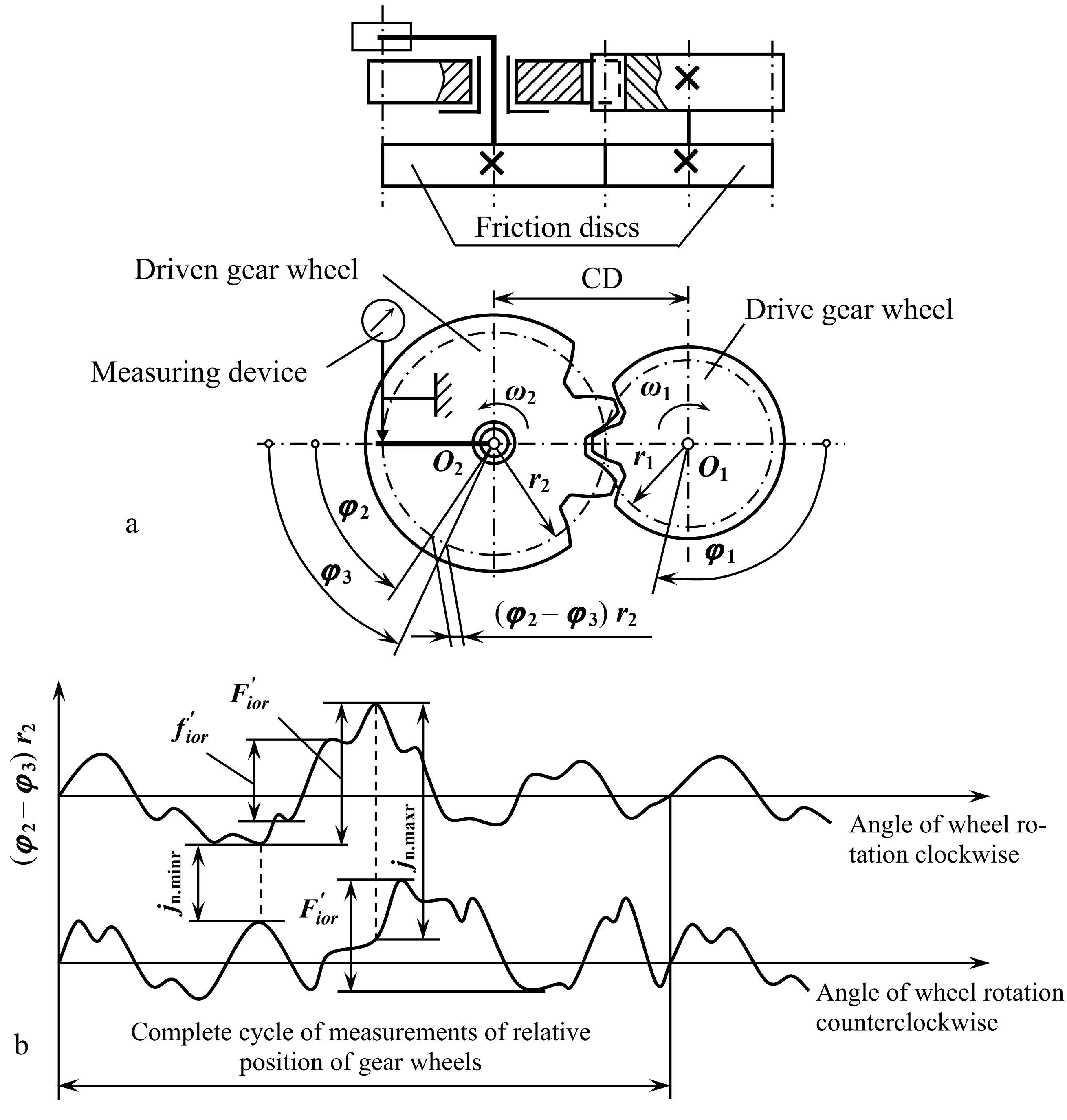


Figure 8.4 – Diagram for combined quality control of toothed gearing (a) and typical curves obtained at direct and opposite rotations (b): r_1 and r_2 – pitch radii of drive and driven wheels respectively; ω_1 and ω_2 – angular rotation speeds; φ_1 – rotation angle of drive wheel; φ_2 – actual rotation angle of driven wheel; φ_3 – nominal rotation angle of driven wheel; CD – centre distance

8.3.1 Measurements of kinematic accuracy parameters

Maximum kinematic error of toothed gearing F'_{ior} is a maximum algebraic difference of values of kinematic error of gearing in duration of *complete cycle of measure-*

ments of relative position of gear wheels (Fig. 8.4,b). It is expressed in linear units as an arc length of pitch circle of a driven wheel. Tolerance for kinematic error of gearing is F'_{io} (ref. Fig. 8.2).

Maximum kinematic error of gear wheel F'_{ir} (Fig. 8.5) is a maximum algebraic difference of values of kinematic error of gear wheel (mounted as a driven wheel) at a complete turn around its work axis rotated by measuring (master) gear wheel (mounted as a drive wheel) at nominal mutual position of these wheels' axes in the range of measured wheel *complete turn* (2π). It is expressed in linear units as an arc length of pitch circle of measured wheel. It is in a relationship with other gear errors:

$$F'_{ir} = F_{Pr} + f_{fr},$$

where F_{Pr} – accumulated error of gear wheel pitch; f_{fr} – tooth profile error.

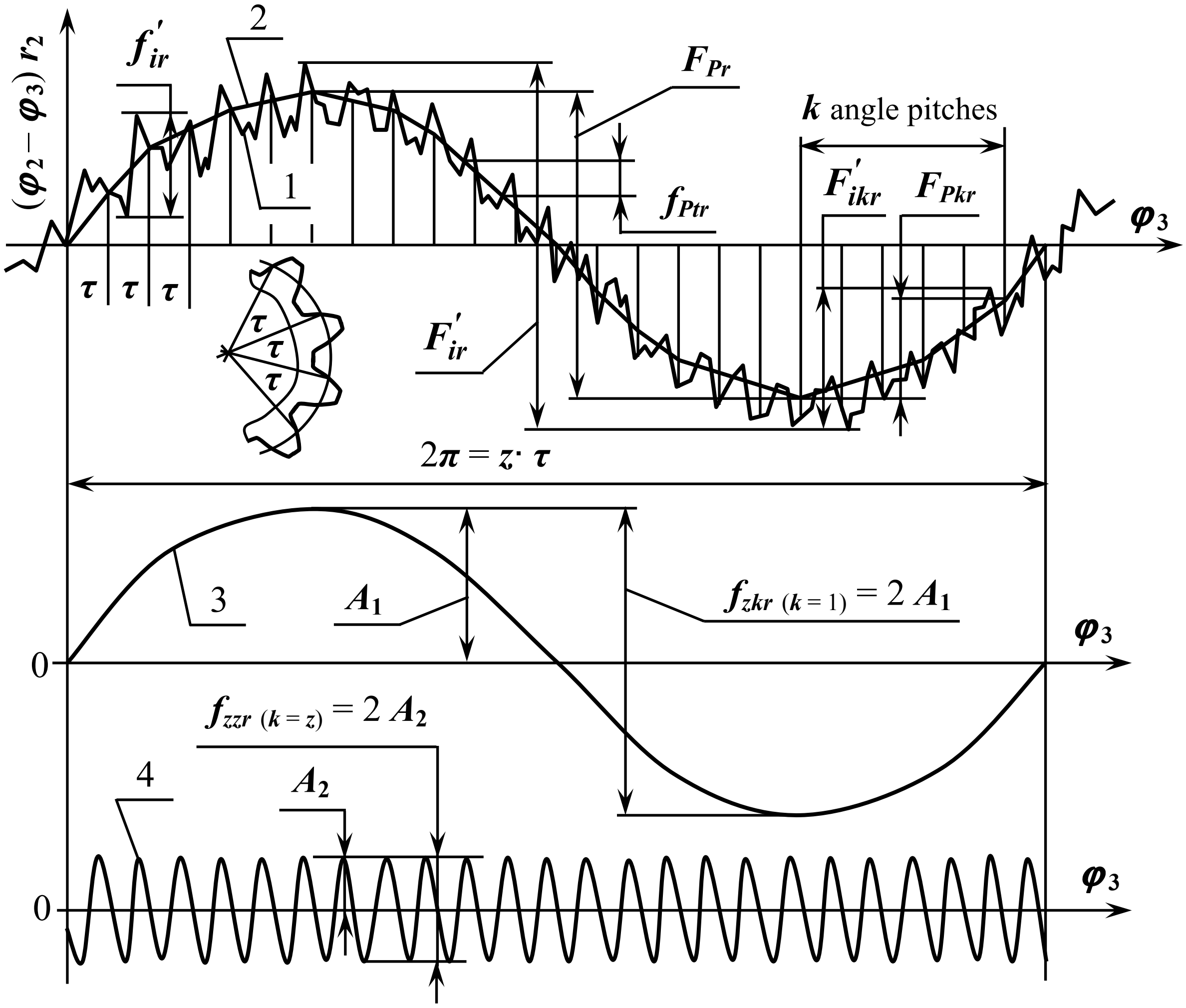


Figure 8.5 – Diagrams for kinematic and operation smoothness errors at single tooth contact of gears: 1 – curve of kinematic error; 2 – curve of accumulated pitch deviations; 3 – the 1st harmonic of curve 1 with amplitude A_1; 4 – harmonic of tooth frequency of curve 1 with amplitude A_2; τ – angle pitch between two like points on neighbour teeth profiles

Tolerance for kinematic error of gear wheel is $\boldsymbol{F_i'}$ (ref. Fig. 8.2).

*Kinematic error of gear wheel at **k** angle pitches* $\boldsymbol{F_{ikr}'}$ (ref. Fig. 8.5) is a difference between actual and nominal angles of gear rotation on its work axis (by considered number of teeth ***k***) driven by measuring gear wheel at nominal position of both wheels axes. It is in a relationship with other gear errors:

$$\boldsymbol{F_{ikr}' = F_{Pkr} + f_{fr}} .$$

Tolerance for kinematic error of gear wheel at ***k*** angle pitches is $\boldsymbol{F_{ik}'}$.

*Accumulated error of **k** angle pitches* $\boldsymbol{F_{Pkr}}$ (ref. Fig. 8.5) is a maximum difference of discrete values of accumulated pitch error of gear wheel at its nominal rotation by ***k*** integer angle pitches ($\boldsymbol{\tau}$).

Tolerance for accumulated error of ***k*** angle pitches is $\boldsymbol{F_{Pk}}$ (ref. Fig. 8.2).

Accumulated error of gear wheel pitch $\boldsymbol{F_{Pr}}$ (ref. Fig. 8.5) is a maximum algebraic difference of accumulated pitch errors around gear wheel.

Tolerance for accumulated error of gear wheel pitch is $\boldsymbol{F_P}$ (ref. Fig. 8.2).

Radial motion of toothing $\boldsymbol{F_{rr}}$ is a difference of actual extreme positions of basic rack along a gear wheel turn from its work axis. The $\boldsymbol{F_{rr}}$ appears because of non-coincidence of work axis of gear wheel with geometric (manufacturing) axes of toothing.

Tolerance for radial motion of toothing of gear wheel is $\boldsymbol{F_r}$ (ref. Fig. 8.2).

8.3.2 Measurements of operation smoothness parameters

Local kinematic error of toothed gearing $\boldsymbol{f_{ior}'}$ is a maximum difference between neighbour extreme values of gearing kinematic error in duration of *complete cycle of measurements* of relative position of gear wheels (ref. Fig. 8.4,b). Tolerance for kinematic error of gearing is $\boldsymbol{f_{io}'}$ (ref. Fig. 8.2).

Local kinematic error of gear wheel $\boldsymbol{f_{ir}'}$ (ref. Fig. 8.5) is a maximum difference between neighbour extreme values of gear wheel kinematic error in the range of its *complete turn* ($2\boldsymbol{\pi}$). It is in a relationship with other gear errors

$$\boldsymbol{f_{ir}' = |f_{Ptr}| + f_{fr}} ,$$

where $\boldsymbol{f_{Ptr}}$ – pitch error of gear wheel; $\boldsymbol{f_{fr}}$ – tooth profile error.

Tolerance for kinematic error of gear wheel is $\boldsymbol{f_i'}$ (ref. Fig. 8.2).

Cyclic error of gear wheel $\boldsymbol{f_{zkr}}$ (at $\boldsymbol{k} = 1$) is equal to double amplitude of harmonic component of kinematic error of gear (ref. Fig. 8.5). Tolerance for cyclic error of gear wheel is $\boldsymbol{f_{zk}}$ (ref. Fig. 8.2).

The first harmonic component ($\boldsymbol{k} = 1$) by its value is practically equal to kinematic error F'_{ir}. The second harmonic component ($k = z$) is a determining factor for operation smoothness that defines a wavy shape of kinematic curve.

Cyclic error of tooth frequency of gear wheel f_{zzr} (at $k = z$) is a cyclic error in engagement with measuring gear (in a gearing) with frequency of repetitions equal to frequency of teeth coming in contact (ref. Fig. 8.5). Tolerance for cyclic tooth-frequency error of gear wheel is f_{zz} (ref. Fig. 8.2).

Pitch error of gear wheel f_{Ptr} (ref. Fig. 8.5) is a discrete value of kinematic error of gear wheel at its rotation in one nominal angle step (τ). Limit deviations for pitch error of gear wheel are: the upper $+f_{Pt}$ and the lower $-f_{Pt}$ (ref. Fig. 8.2).

Base pitch error of gear wheel f_{Pbr} is a difference between actual and nominal base pitches. Limit deviations for base pitch error are: the upper $+f_{Pb}$ and the lower $-f_{Pb}$ (ref. Fig. 8.2).

Base pitch error and pitch error values are in dependency

$$|f_{Pb}| = |f_{Pt}| \cdot \cos \alpha.$$

8.3.3 Measurements of teeth contact patch parameters

If teeth flanks of a drive wheel are covered with paint and both wheels are rotated in a complete turn with slight breaking to provide reliable contact of both wheels teeth, then paint imprints will appear on the teeth flanks of a driven wheel (Fig. 8.6).

Relative dimensions of a contact patch are determined from formulas in percents:

- Along tooth length $\dfrac{a-c}{b}\times 100\%$;

- Along tooth height $\dfrac{h_m}{h_p}\times 100\%$,

where a – patch length; b – tooth width; c – gap in a patch length; h_m – patch height; h_p – height of active tooth flank.

Relative dimensions of a contact patch characterise teeth contact for gearing and wheels.

Instant contact patch is a part of active tooth flank of gear in the assembled gearing, on which paint imprints appear after rotation of the gear *in a complete turn* with slight breaking.

Total contact patch is a part of active tooth flank of gear in the assembled gearing, on which paint imprints appear after rotation of the gear *under the load specified by a designer*.

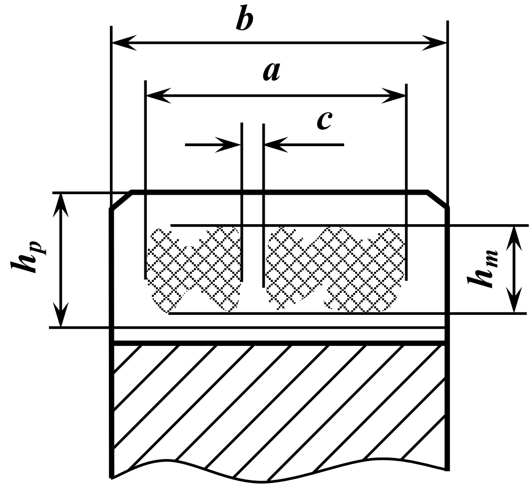


Figure 8.6 – Geometric parameters of contact patch along tooth flank

8.3.4 Measurements of circumferential backlash parameters

Additional addendum modification shift of basic rack $\boldsymbol{E}_{\mathbf{H}}$ from nominal position of basic rack into a body of gear wheel is performed with aim to ensure the minimum guaranteed circumferential backlash $\boldsymbol{j}_{\mathbf{n.min}}$ in a gearing (Fig. 8.7).

The *minimum* additional addendum modification shift equals to:

- For a gear wheel with *external* teeth $-\boldsymbol{E}_{\mathbf{Hs}}$ (upper deviation);
- For a gear wheel with *internal* teeth $+\boldsymbol{E}_{\mathbf{Hi}}$ (lower deviation).

Tolerance for additional addendum modification shift is $\boldsymbol{T}_{\mathbf{H}}$.

According to the Fig. 8.7 the minimum additional addendum modification shift equals to

$$E_{Hs} = -\frac{j_{n.min} + k_j}{4 \sin\alpha}.$$

Minimum (upper) deviation of tooth thickness $\overline{S}_c$ is

$$E_{cs} = E_{Hs} \cdot 2tg\alpha.$$

Tolerance for tooth thickness $\overline{S}_c$ is

$$T_c = T_H \cdot 2tg\alpha.$$

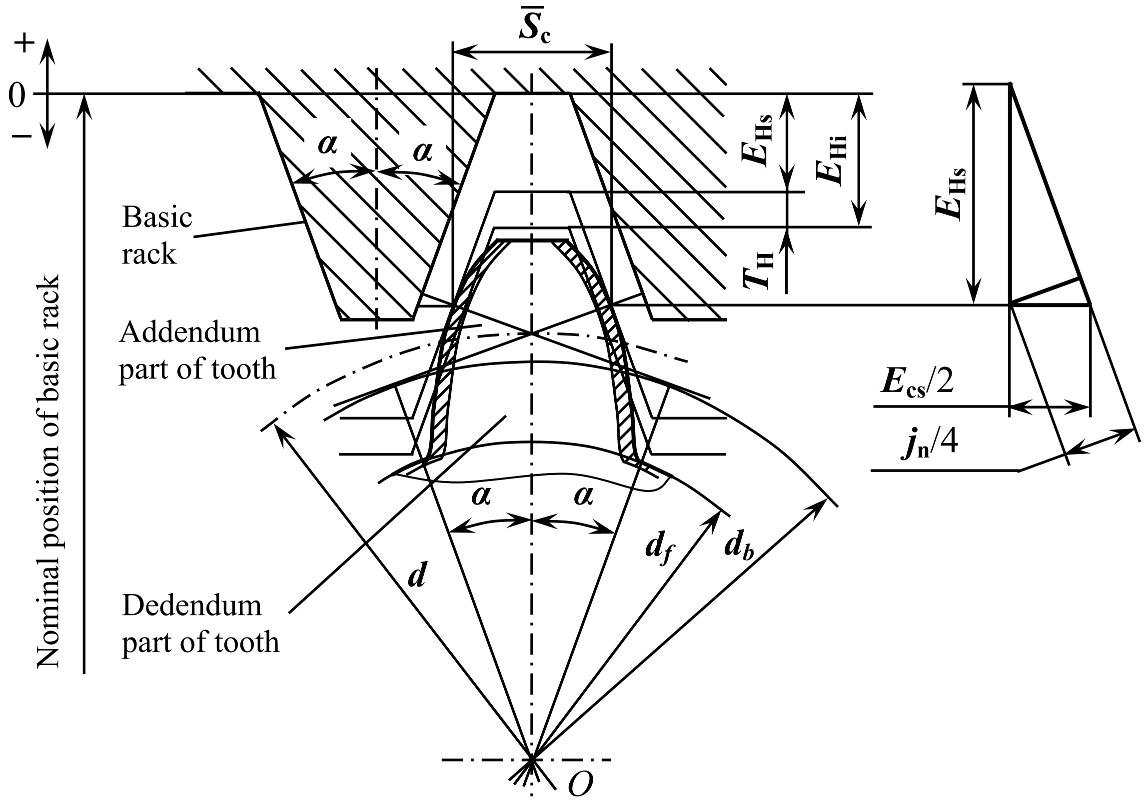


Figure 8.7 – Diagram for additional addendum modification shift from nominal position of basic rack into a body of gear wheel: $\boldsymbol{d}$ – pitch circle; $\boldsymbol{d_f}$ – root circle; $\boldsymbol{d_b}$ – base circle; $\overline{S}_c$ – nominal thickness of tooth along permanent chord

Minimum guaranteed standardised circumferential backlash per one gear wheel without random errors of manufacturing and assembling ($\boldsymbol{k}_{\mathrm{j}} = 0$) equals to

$$\boldsymbol{j}_{\mathbf{n.min}} = \boldsymbol{E}_{\mathbf{Hs}} \cdot 2\boldsymbol{sin\alpha}.$$

Minimum guaranteed standardised circumferential backlash for a gearing equals to

$$\boldsymbol{j}_{\mathbf{n.min}} = (\boldsymbol{E}_{\mathbf{Hs1}} + \boldsymbol{E}_{\mathbf{Hs2}})\cdot 2\boldsymbol{sin\alpha},$$

where $\boldsymbol{E}_{\mathbf{Hs1}}$ – minimum additional addendum modification shift of the 1st gear; $\boldsymbol{E}_{\mathbf{Hs2}}$ – minimum additional addendum modification shift of the 2nd gear.

Maximum circumferential backlash for a gearing (ref. Fig. 8.3) is

$$\boldsymbol{j}_{\mathbf{n.max}} = \boldsymbol{j}_{\mathbf{n.min}} + \boldsymbol{T}_{\mathbf{jn}},$$

$$\boldsymbol{T}_{\mathbf{jn}} = (\boldsymbol{T}_{\mathbf{H1}} + \boldsymbol{T}_{\mathbf{H2}} + 2|\boldsymbol{f}_{a}|)\cdot 2\boldsymbol{sin\alpha},$$

where $\boldsymbol{T}_{\mathbf{H1}}$ – tolerance for additional addendum modification shift of the 1st gear; $\boldsymbol{T}_{\mathbf{H2}}$ – tolerance for additional addendum modification shift of the 2nd gear.

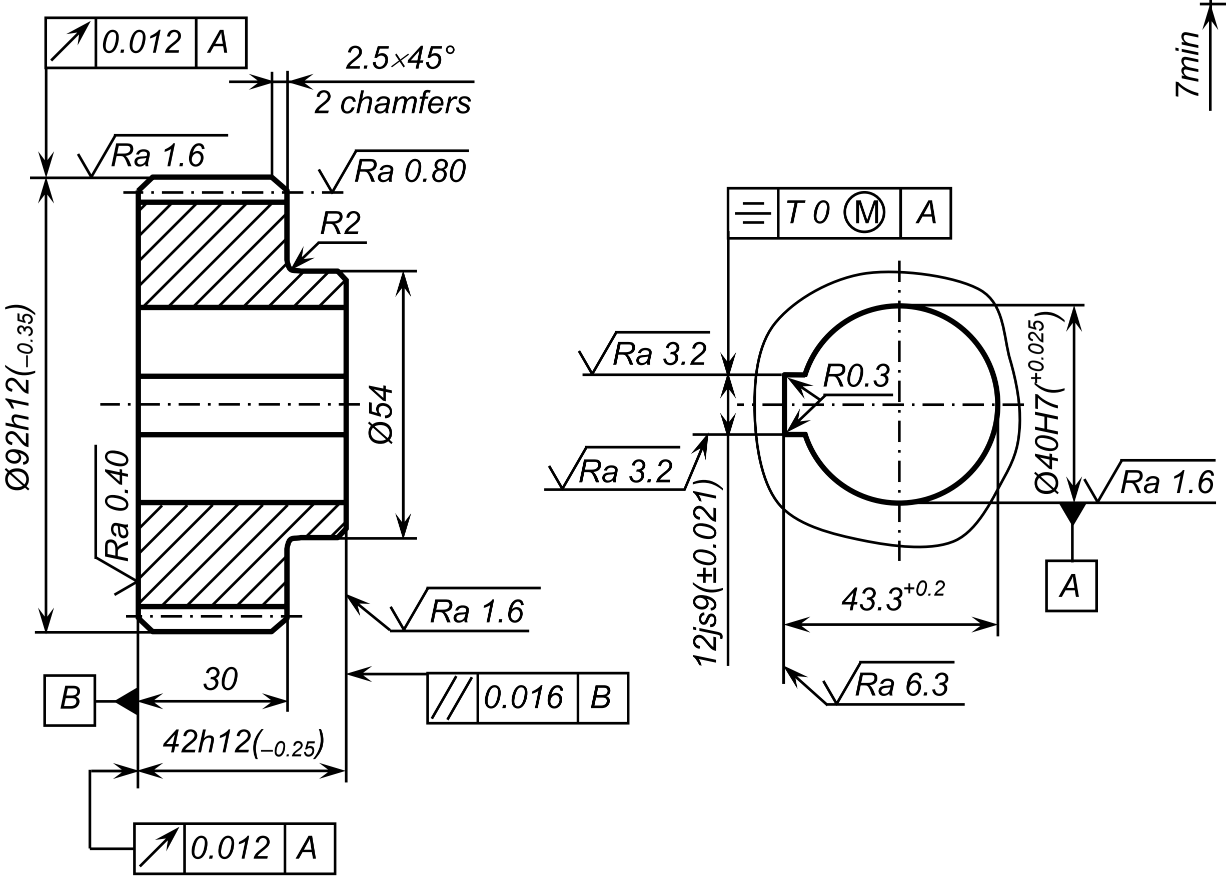
Difference between measured values of $\boldsymbol{j}_{n.maxr}$ and $\boldsymbol{j}_{n.minr}$ (ref. Fig. 8.4,b) is the largest interval of teeth flank clearance in a gearing.

8.4 Specifying accuracy of toothed gears in drawings

Because of peculiarities of gear wheels the requirements for performance of their drawings (Fig. 8.8) are described by special standards. The standards prescribe rules for specifying all elements of *toothing* in *a drawing of gear wheel*. The rest of elements are performed as ordinary items of machine-building in accordance with general requirements of standards of unified system for design documentation (USDD).

Information about *toothing* is partially placed directly on the picture of gear, and partially – in special table (ref. Fig. 8.8) in the upper right-hand corner of a drawing. On the picture of gear, in particular, diameter of tip circle diameter (∅92h12) is shown and, if necessary, tolerance for its radial motion variation, toothing width, permissible motion variation of datum face, dimensions of chamfers and rounding radii for teeth edges, roughness of tooth flanks, etc.

The table with toothing parameters is divided in three segments with solid thick lines. In the first (upper) segment main data is submitted: module, helix angle and its direction, basic rack, addendum modification coefficient, as well as specified accuracy and type of engagement. The second segment is designed for inspection parameters, the third – for reference data.



Ra 12.5 (√)

Module	m	4
Number of teeth	z	21
Basic rack	-	GOST 13755-81
Addendum modification coefficient	x	0
Accuracy degree by GOST 1643-81	-	8-7-6-B
Base tangent length	W	$30.696^{-0.121}_{-0.191}$
Tolerance for radial variation of toothing	F_r	0.05
Tolerance for variation of base tangent length	F_{vW}	0.028
Limit deviations of base pitch	$\pm f_{Pb}$	±0.017
Tolerance for tooth flank	f_f	0.014
Total error of tooth distortion	F_β	0.009
Pitch circle diameter	d	84
Base pitch	P_b	11.808

20 7min 10 35 110

Figure 8.8 – Fragment of drawing for cylindrical toothed gear

9 BASIC NORMS OF INTERCHANGEABILITY FOR KEY AND SPLINE JOINTS

Key and spline joints are designed for creation of detachable joints transmitting torque.

9. 1 Key joints and their accuracy

Key joints (Fig. 9.1) are applied when requirements for alignment accuracy are not severe. Types of keys are: straight, woodruff, taper, tangential.

Dimension of key width $\boldsymbol{b}$ with tolerance band h9 is accepted as a nominal dimension of key joint. Fits of a key with shaft and bushing depends on type of engagement: loose, normal or tight.

Standard designations of key joints for loose engagement (Fig. 9.2) are:

$$\boldsymbol{b}\frac{\mathrm{H9}}{\mathrm{h9}} \text{ for fit of shaft and key and } \boldsymbol{b}\frac{\mathrm{D10}}{\mathrm{h9}} \text{ for fit of bushing and key,}$$

where $\boldsymbol{b}$ – key width, mm; H9 – tolerance band for keyway width in shaft; D10 – tolerance band for keyway width in bushing; h9 – tolerance band for key width.

Standard designations of other key joints (ref. Fig. 9.2) are:

- For normal engagement: $\boldsymbol{b}\frac{\mathrm{N9}}{\mathrm{h9}}$ for fit of shaft and key; $\boldsymbol{b}\frac{\mathrm{Js9}}{\mathrm{h9}}$ for fit of bushing and key;
- For tight engagement: $\boldsymbol{b}\frac{\mathrm{P9}}{\mathrm{h9}}$ for fit of shaft and key; $\boldsymbol{b}\frac{\mathrm{P9}}{\mathrm{h9}}$ for fit of bushing and key.

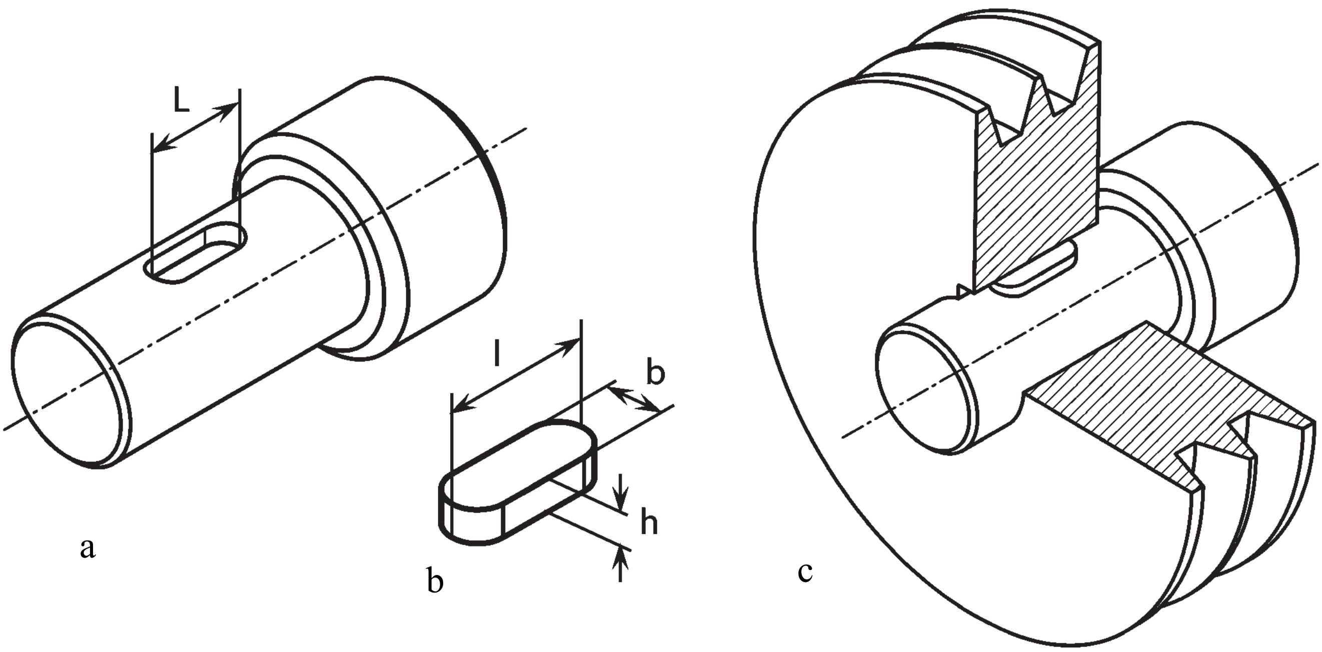


Figure 9.1 – Shaft (a) with keyway, straight key (b) with main dimensions and key joint (c)

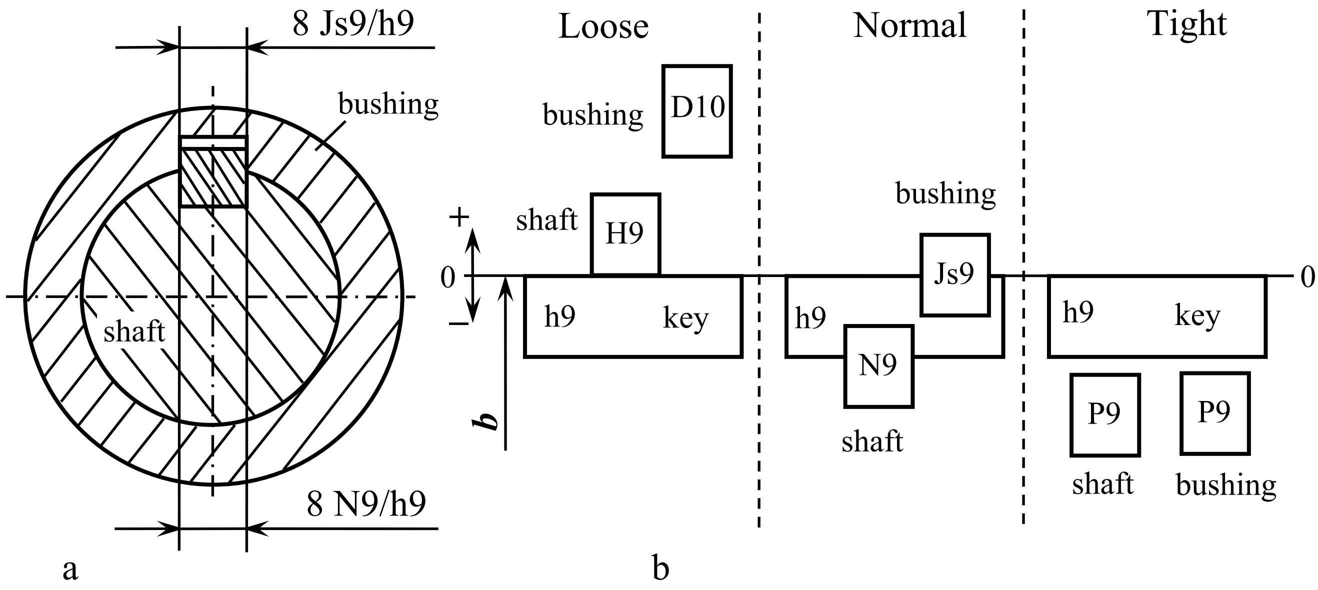


Figure 9.2 – Specifying the fits of key joint in assembly drawing (a) and diagrams for locations of tolerance bands of key-joint elements (b) depending on type of engagement

Tolerance bands of straight key dimensions (ref. Fig. 9.1) are:
- For height ***h***: h11;
- For length ***l***: h14;
- For keyway length ***L***: H15.

Example of designation of straight key with dimensions ***b*** = 16 mm, ***h*** = 10 mm and ***l*** = 80 mm is

Key 16×10×80 GOST 23360-78.

9.2 Spline joints and their accuracy

Spline joints (Fig. 9.3) have significant advantages as compared with key joints by alignment accuracy, transmitting torque, strength.

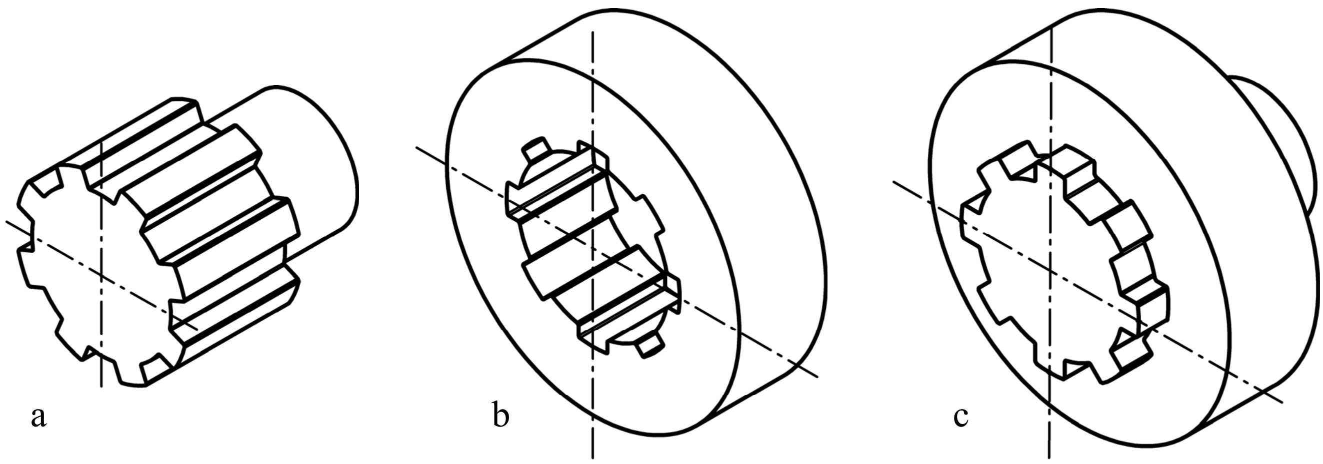


Figure 9.3 – Spline shaft (a), spline bushing (b) and spline joint (c)

By teeth profile splines are divided into straight (ref. Fig. 9.3), involute, triangular, trapezoidal.

9.2.1 Straight spline joints

Methods for alignment of shaft and bushing (Fig. 9.4) both for sliding and fixed joints are:

1) Along minor diameter ***d*** – high alignment accuracy. It is applied for movable joints;

2) Along major diameter ***D*** – for joints transmitting small torque values;

3) Along teeth sides – not high alignment accuracy. It is applied for transmitting large torque values and for transmission reversals.

Example of standard designation of straight-spline sliding joint with alignment along minor diameter ***d*** is

$$\boldsymbol{d} - 8 \times 36 \frac{H7}{f7} \times 40 \frac{H12}{a11} \times 7 \frac{D9}{h9} \text{ GOST 25346-89,}$$

where 8 – number of teeth ***z***; 36 – minor diameter ***d***, mm; $\frac{H7}{f7}$ – clearance fit for alignment along surface ***d***; 40 – major diameter ***D***, mm; $\frac{H12}{a11}$ – clearance fit for loose surface ***D***; 7 – width of tooth ***b***, mm; $\frac{D9}{h9}$ – clearance fit along tooth width ***b***.

The standard states recommended fits according to operation conditions, type of joint (sliding or fixed) and method of alignment.

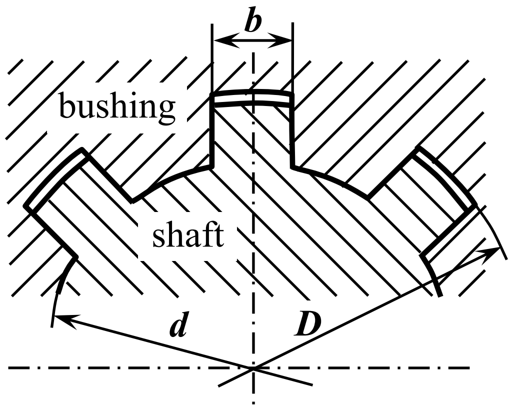


Figure 9.4 – Fragment of straight spline joint with main parameters

9.2.2 Involute spline joints

Involute spline joints surpass straight spline joints by transmitting torque value, accuracy of alignment and teeth direction, cyclic strength and service life. They are relatively easier in manufacture.

Basic characteristics of involute joints (Fig. 9.5) are:

- ***m*** – module;
- ***z*** – number of teeth;
- α = 30º – profile angle of basic rack;
- ***D*** – nominal diameter (major diameter – diameter of bushing root circle);
- ***d*** – reference circle diameter;
- ***s*** – shaft tooth thickness along reference circle (circumferential thickness);
- ***e*** – bushing tooth space width along reference circle (circumferential width).

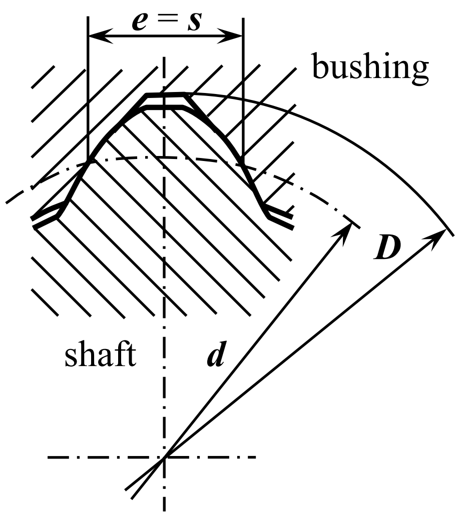


Figure 9.5 – Fragment of involute spline joint with main parameters

Alignment methods for involute spline joints are:
1) Along flanks of teeth and spaces surfaces (***s***, ***e***) – preferable method;
2) Along major diameter ***D*** – ensures maximal alignment accuracy of parts with a shaft;
3) Along minor diameter $\boldsymbol{d_1}$ – permitted, but not recommended.

Tolerances for width of bushing space ***e*** and shaft tooth thickness ***s*** are stated with *accuracy degrees*, but not with accuracy grades:
- For ***e*** – accuracy degrees 7, 9, 11;
- For ***s*** – accuracy degrees 7, 8, 9, 10, 11.

Basic deviation for bushing space ***e*** is only one H, basic deviations for bushing tooth thickness ***s*** are ten: a, c, d, f, g, h, k, n, p, r. Total tolerance ***T*** for tooth space (or for tooth thickness) consists of two portions: $\boldsymbol{T_e}$ (or $\boldsymbol{T_s}$) – tolerance for deviation of space width dimension ***e*** (or tooth thickness ***s***) and tolerance for deviations of form and position of space (tooth) profile elements from theoretical profile (Fig. 9.6). Example of designation of involute spline joint with alignment along teeth sides is

$$50 \times 2 \times \frac{9H}{9g} \text{ GOST 6063-80,}$$

where 50 – nominal diameter of joint ***D***, mm; 2 – module ***m***, mm; 9H – accuracy degree for bushing tooth space width ***e***; 9g – accuracy degree for tooth thickness ***s***.

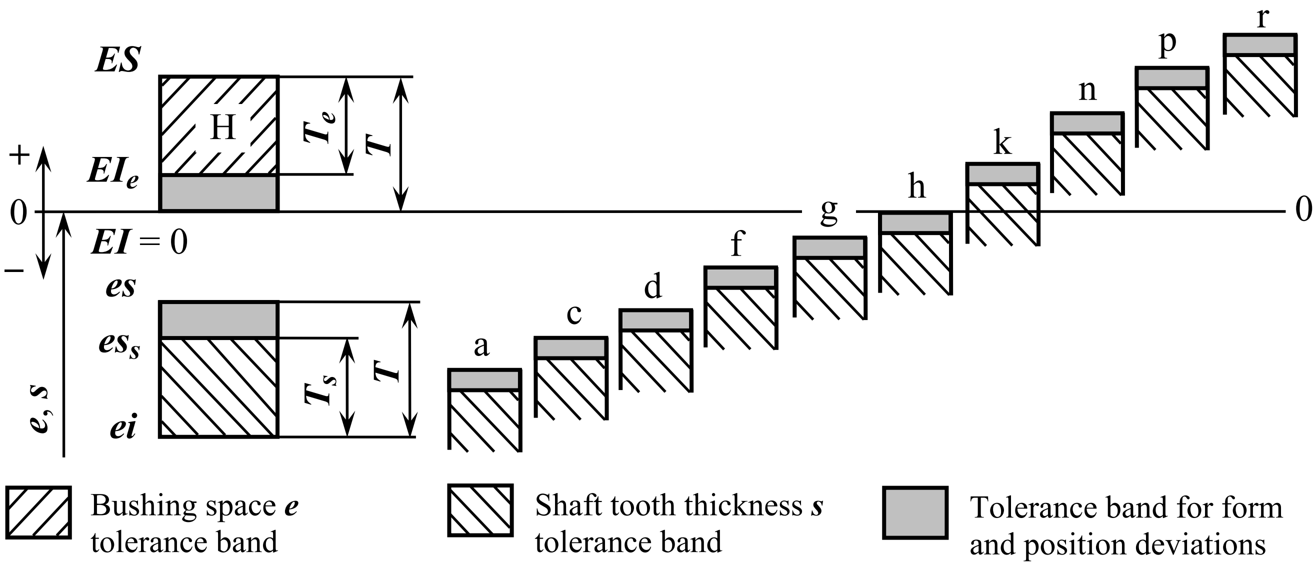


Figure 9.6 – Relative position of bushing space ***e*** and shaft tooth thickness ***s*** tolerance bands and their tolerance bands for form and position in involute spline joint

10 STANDARDISATION OF DEVIATIONS OF FORM AND POSITION. ROUGHNESS AND WAVINESS OF SURFACES

10.1 Deviations and tolerances of surfaces form and position

Shape of parts applied in machine-building is a combination of simple and double-curvature geometric surfaces. Basically they are cylindrical parts (70 %), flat (12 %), toothed gears (3 %), body-type parts (4 %) and others. It is impossible to obtain perfect form and relative positions of surfaces during manufacturing process because of errors of machine-tools, workholding devices, workpiece, tools, etc., that is, form and position of any real surface will differ from the perfect one.

In movable joints these deviations cause decrease of the parts wear-resistance because of increased contact pressure at the peaks of surface irregularities, violations of operation smoothness, increased noise, etc.

In the fixed joints deviations of form and position of surfaces cause non-uniform interference that result in decrease of a joint strength, pressure-tightness and accuracy of alignment.

In assemblies these deviations cause errors of locating of parts relative each other, unequal clearances that result in violations of normal operation of mechanisms and machines. For example, frictionless bearings are very sensitive to deviations of form and relative position of mating surfaces.

Deviations of surfaces form and position reduce technological indexes of manufacture because they worsen accuracy and increase labour-intensiveness of assembling, enlarge volume of preparatory operations, reduce accuracy of dimensions measurements, worsen accuracy of locating of a part at manufacture and control.

10.1.1 Deviations and tolerances of surfaces form

Form deviation is a deviation of form of real (actual) element (surface, line) from a nominal (perfect) form that is estimated by the largest distance from points of actual element to adjoining element along a normal. Numerical values of tolerances for form are specified depending on ***accuracy degree***.

Surface microirregularities, related to surface roughness, are not included into the form deviations.

Deviations of and tolerances of form, profile and position of surfaces are submitted in Table 10.1.

Selection of tolerances for form depends on design and manufacturing requirements and related with a dimension tolerance. Tolerance band of dimension for mating surfaces limits also any deviations of form along the length of a joint. No one of form deviations must exceed a dimension tolerance. Form tolerances are specified in those cases, when they should be less than dimension tolerance.

Table 10.1 – Deviations and tolerances of form, profile and position of surfaces

Deviation name	Tolerance name	Tolerance symbol
Deviations and tolerances of form		
Deviation of straightness	Tolerance for straightness	⏤
Deviation of flatness	Tolerance for flatness	⏥
Deviation of roundness	Tolerance for roundness	○
Deviation of cylindricity	Tolerance for cylindricity	⌭
Deviation of longitudinal-section profile	Tolerance for longitudinal-section profile	=
Deviations and tolerances of position		
Deviation from parallelism	Tolerance for parallelism	//
Deviation from perpendicularity	Tolerance for perpendicularity	⊥
Deviation from angularity	Tolerance for angularity	∠
Deviation from concentricity	Tolerance for concentricity	◎
Deviation from symmetry	Tolerance for symmetry	⌯
Deviation from true position	Tolerance for true position	⌖
Deviation from axes intersection	Tolerance for axes intersection	×
Total deviations and tolerances of surfaces forms and positions		
Radial runout	Tolerance for radial runout	↗
Face runout	Tolerance for face runout	↗
Motion variation in specified direction	Tolerance for motion variation in specified direction	↗
Total radial runout	Tolerance for total radial runout	⌰
Total face runout	Tolerance for total face runout	⌰
Deviation from form of specified profile	Tolerance for form of specified profile	⌒
Deviation from form of specified surface	Tolerance for form of specified surface	⌓

Several types of form deviations and designations of form tolerances in drawings are shown in Fig. 10.1. Symbol of tolerance and its numerical value are depicted in the standardised frame. Specified tolerance can be applied to whole length ***L*** of element (ref. Fig. 10.1,a and 10.1,c), or it can be spread only along the specified length (ref. Fig. 10.1,b). In this case roundness deviations are checked in any cross section along the length of 50 mm.

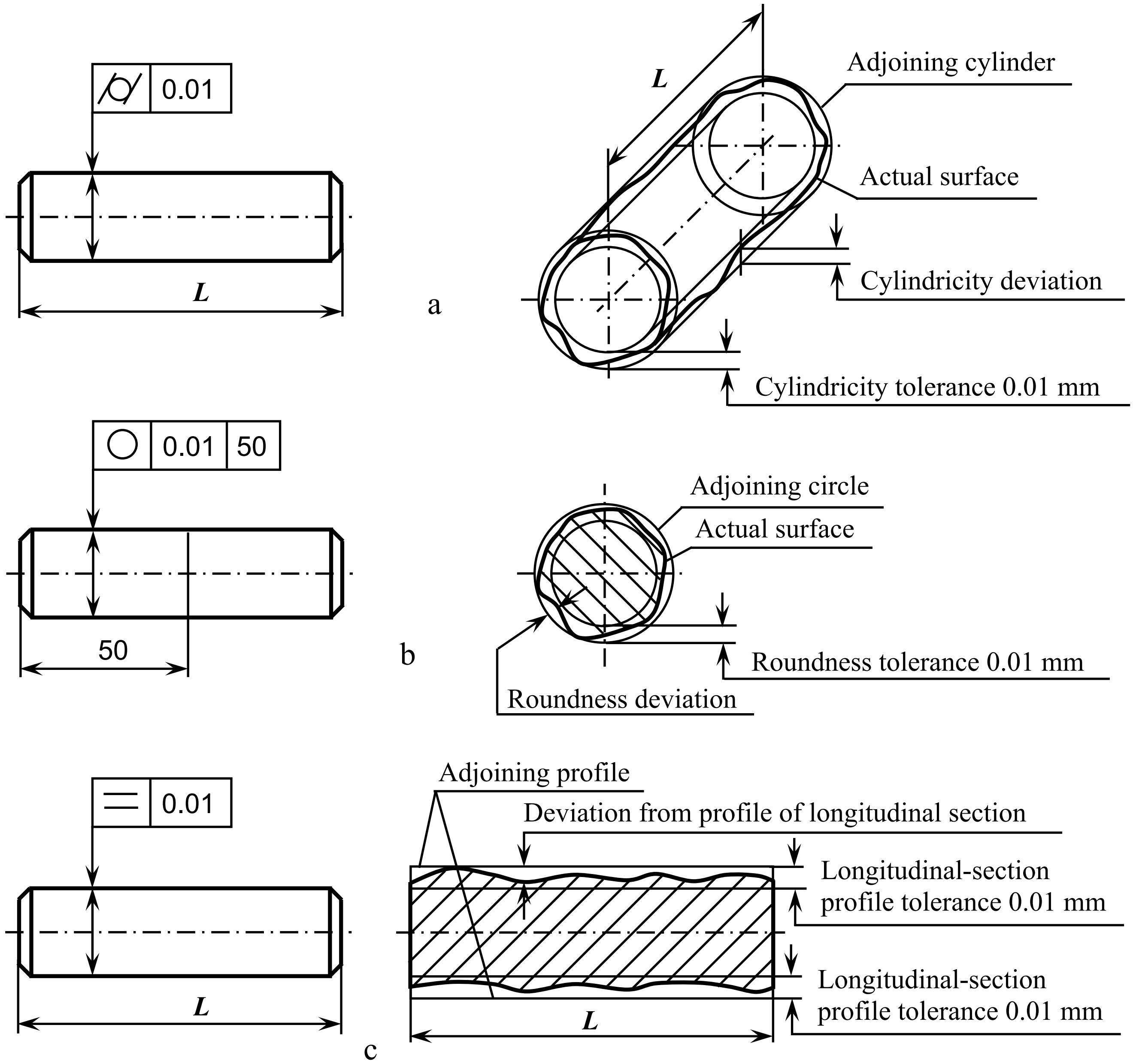


Figure 10.1 – Designations of form tolerances in drawings and schematic diagrams of deviations for: a – cylindricity; b – roundness; c – profile of longitudinal section

Tolerance for cylindricity includes tolerances for roundness and profile of longitudinal section.

Particular types of deviations from roundness are ovality and lobing (faceting). *Ovality* is a deviation from roundness, at which actual cross-section profile has an oval contour with perpendicular maximum and minimum dimensions (Fig. 10.2,a). *Lobing* is a deviation from roundness, at which actual cross-section profile has a form of multi-faceted figure (Fig. 10.2,b).

Particular types of deviations from profile of longitudinal section are taper, barrel and bow. *Taper* is a deviation of longitudinal-section profile, at which generating lines

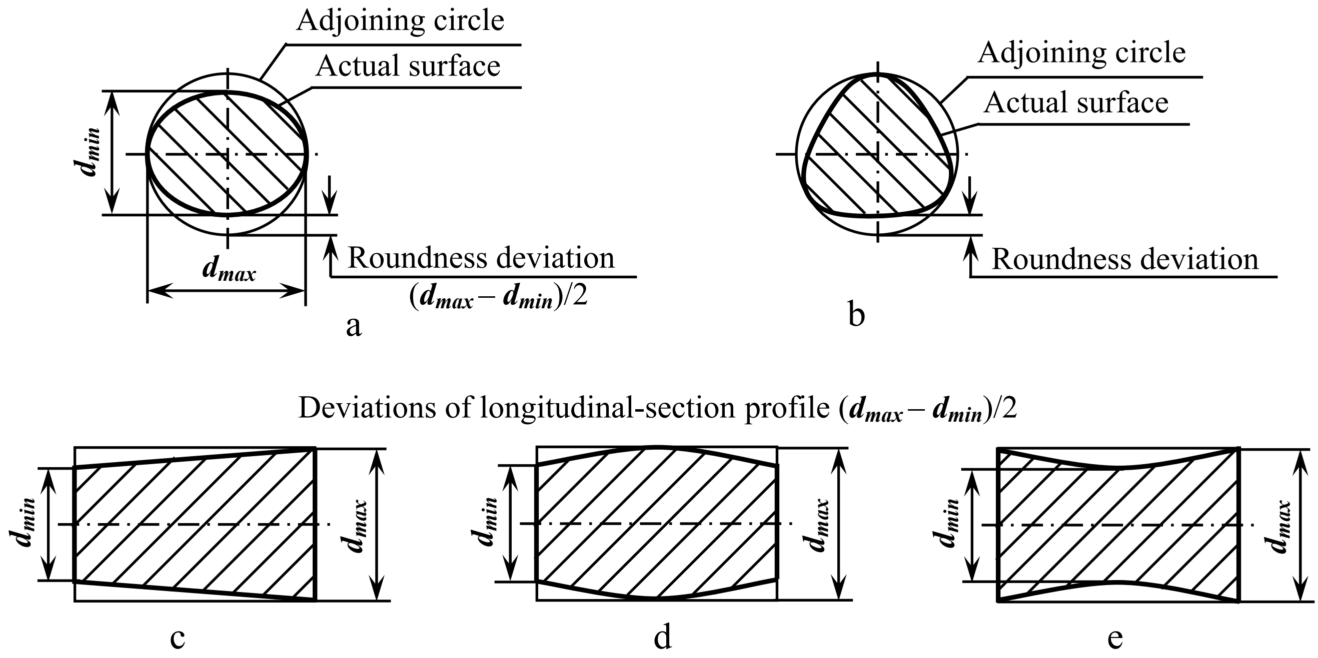


Figure 10.2 – Particular types of form deviations: a – ovality; b – lobing; c – taper; d – barrel; e – bow

are straight, but not parallel (Fig. 10.2,c). *Barrel* is a deviation of longitudinal-section profile, at which generating lines are curved and diameters enlarge from ends to middle of the section (Fig. 10.2,d). *Bow* is a deviation of longitudinal-section profile, at which generating lines are curved and diameters decrease from ends to middle of the section (Fig. 10.2,e).

10.1.2 Deviations and tolerances of surfaces position

Position deviation is a deviation of actual position of considered element from its nominal position (ref. Table 10.1). Nominal position is a position determined by nominal linear and angle dimensions (Fig. 10.3).

For estimation of surfaces position accuracy, as a rule, datums are specified.

Datum (*locating element*) is a part element (point, line, surface), which is used for coordinating a position of other part element. In drawings datums are specified by capital letters in square frames with a triangle and connecting line.

Numerical values of tolerances for position are specified depending on ***accuracy degree***.

Estimation of position deviation value is performed by position of adjoining surface plotted on actual surface; thus deviations of form are excluded from a consideration.

Dimensions in frames (ref. Fig. 10.3,c) are dimensions with tolerances, values of which are determined by position tolerance of the hole axis. Actual deviations Δx and Δy of dimensions x and y appeared because of deviation of actual hole axis position from its nominal position.

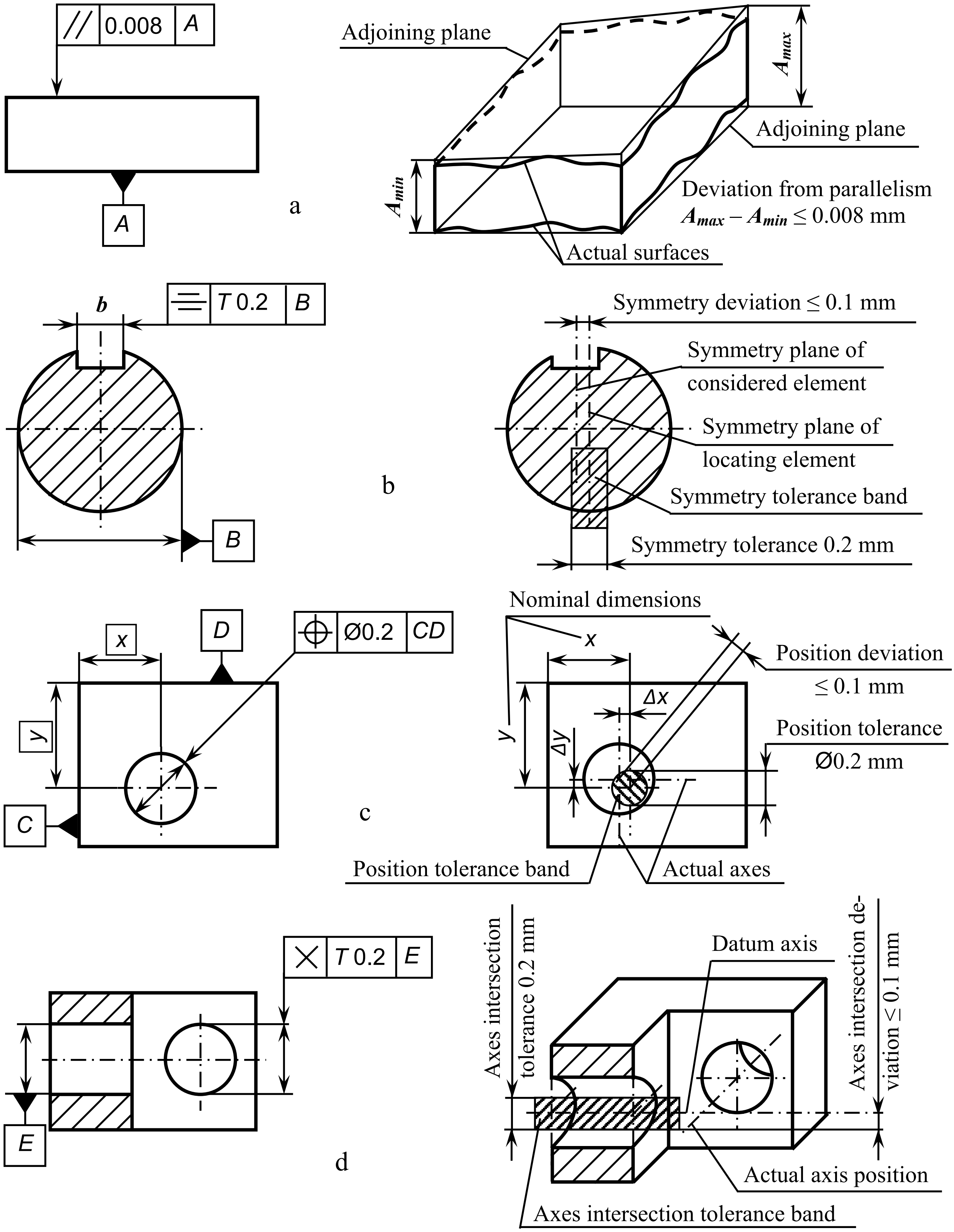


Figure 10.3 – Designations of position tolerances in drawings and schematic diagrams of deviations for: a – parallelism; b – symmetry; c – true position; d – axes intersection

There are two methods for specifying limit deviations for position of holes axes (Fig. 10.4).

1. *Position tolerances of holes axes*. This method is preferable for a quantity of holes more than two. In this case tolerance band is a cylinder of permissible positions of axis of actual hole. Thus, all linear and angular deviations are limited in complex.

2. *Limit deviations of dimensions* that determine position of holes axes in orthogonal or polar coordinate system.

There is a correlation between these methods (Fig. 10.5). Position tolerances $\boldsymbol{T_p}$ can be transformed into limit deviations of coordinate dimensions $\pm\boldsymbol{\Delta L}$, $\pm\boldsymbol{\Delta R}$, $\pm\boldsymbol{T_\alpha}/2$ by means of geometric formulas.

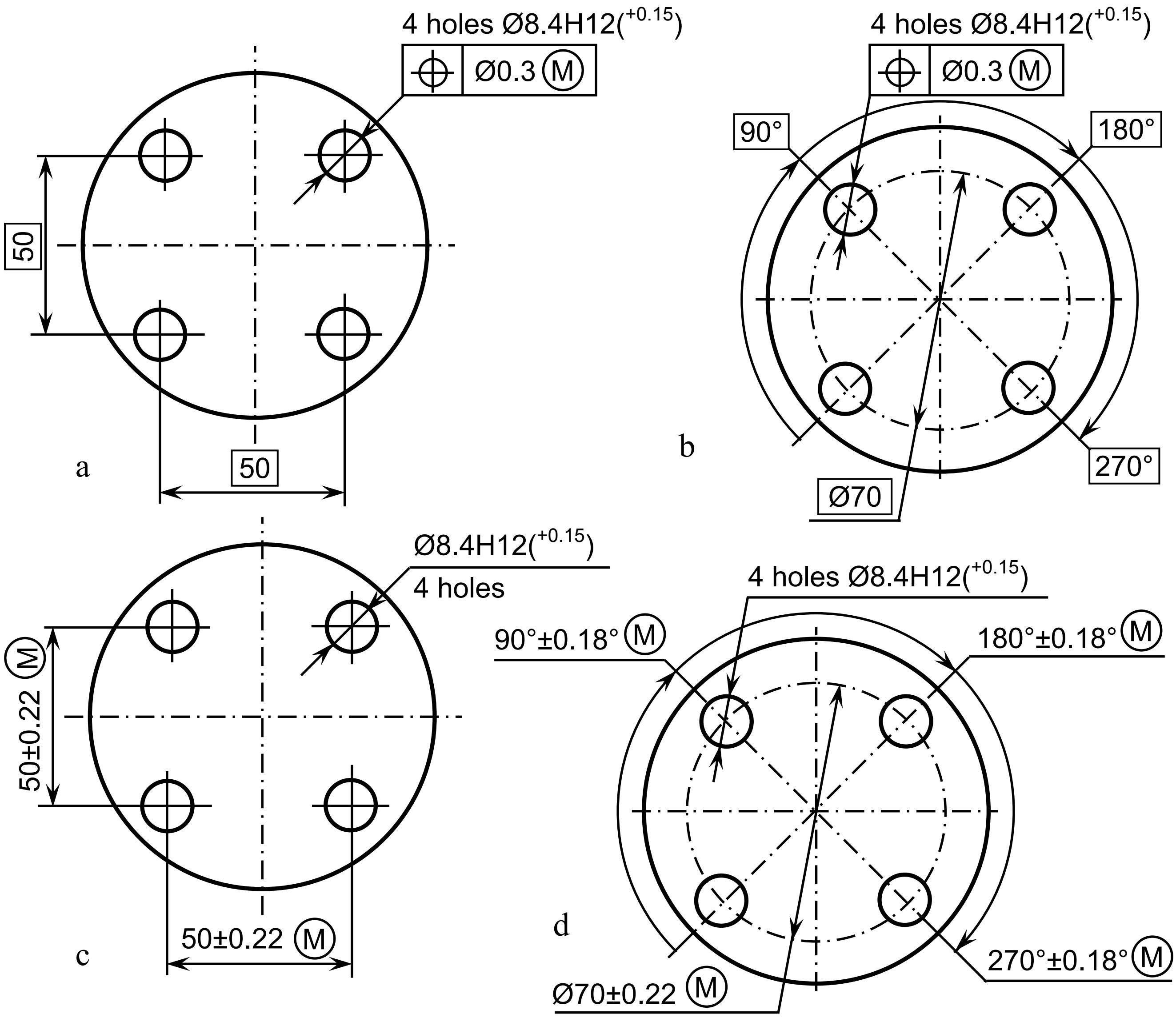


Figure 10.4 – Methods for specifying limit deviations for position of holes axes: a – position tolerances in orthogonal coordinate system; b – position tolerances in polar coordinate system; c – limit deviations of coordinate dimensions in orthogonal system; d – limit deviations of coordinate dimensions in polar system

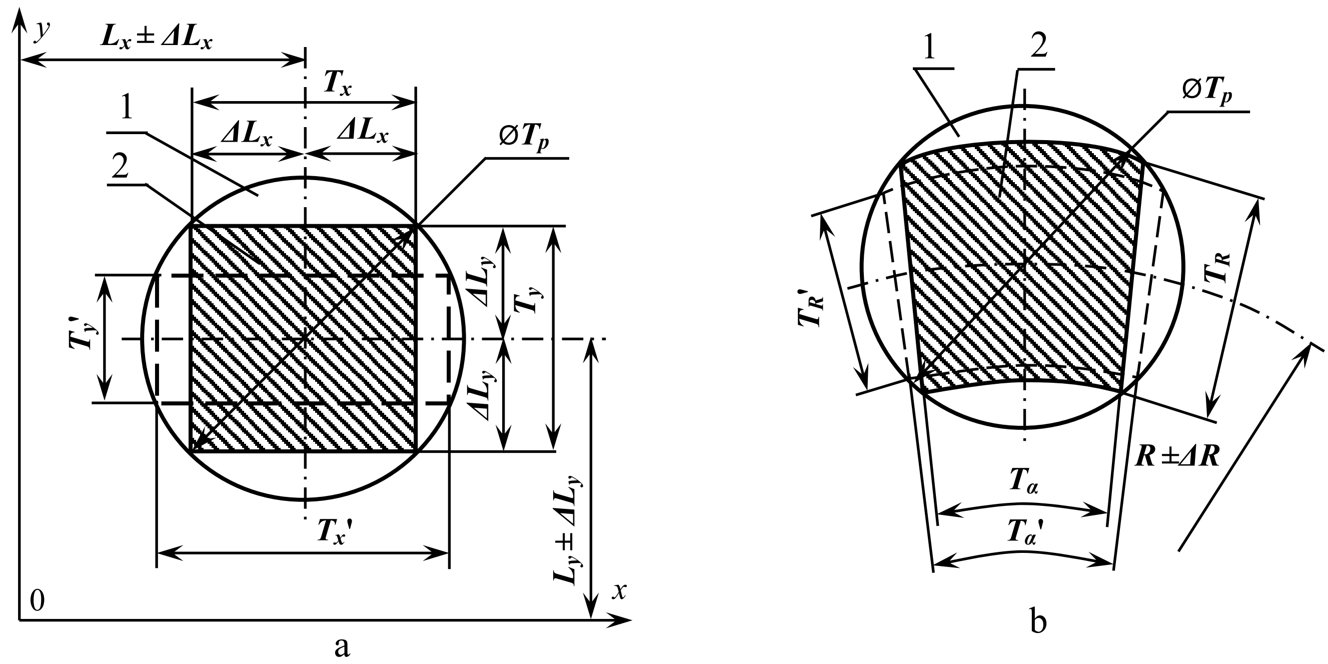


Figure 10.5 – Diagrams for transformation of position tolerances into limit deviations of coordinate dimensions in orthogonal (a) and polar (b) coordinate systems: 1 – round area (projection of cylinder) of possible positions of actual hole axis; 2 – area of possible positions of actual hole axis along coordinate dimensions tolerances

10.1.3 Total deviations and tolerances of surfaces form and position

Total deviation of form and position is a deviation resulting from total action of form deviation and position deviation of a considered surface or considered profile relative to datums (ref. Table 10.1).

Total deviations of surface form and position include (Fig. 10.6):

- *Face runout* (*face motion variation*) is a difference of maximum and minimum distances from the points of actual face surface to a plane perpendicular to datum axis at the specified diameter or at any diameter, if its value is not specified;

- *Total face runout* (*total face motion variation*) is a difference of maximum and minimum distances from the points of whole actual face surface to a plane perpendicular to datum axis. It is a result of deviation from flatness of a considered surface and deviation of its perpendicularity relative to datum axis;

- *Radial runout* (*radial motion variation*) is a difference of maximum and minimum distances from the points of actual surface of revolution to datum axis (to axis of datum surface or common axis) in any cross-section perpendicular to datum axis. It is a result of deviation from roundness of a considered profile and deviation of its centre from datum axis;

- *Total radial runout* (*total radial motion variation*) is a difference of maximum and minimum distances from the points of whole actual surface of revolution to datum axis

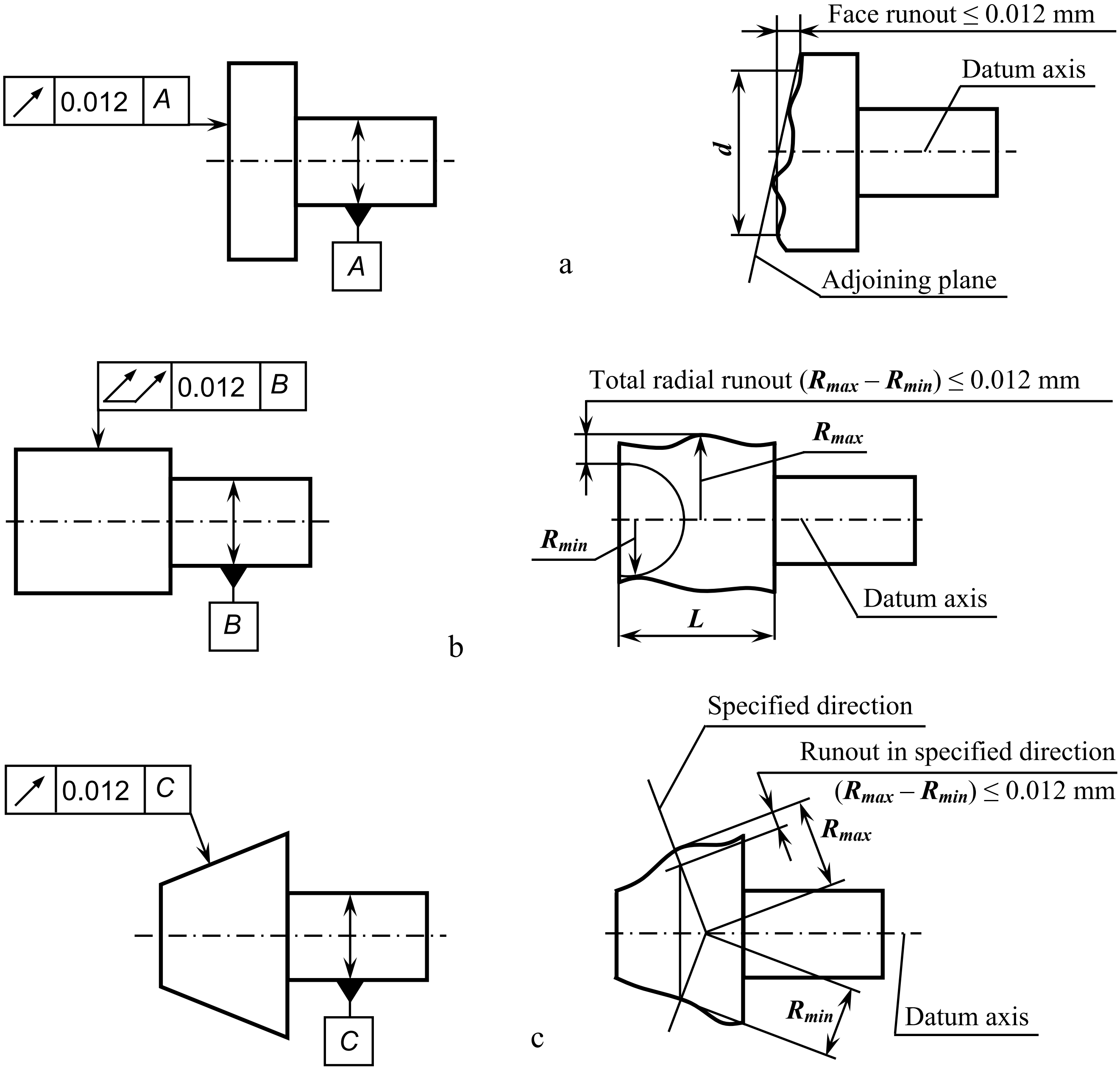


Figure 10.6 – Designations of total form and position tolerances in drawings and schematic diagrams of deviations for: a – face runout; b – total radial runout; c – runout in specified direction

along a specified length of the surface. It is a result of deviation from cylindricity of a surface and deviation of its concentricity relative to datum axis;

- *Tolerance for runout* (*motion variation*) *in specified direction* is a difference of maximum and minimum distances from the points of actual surface of revolution to datum axis (to axis of datum surface or common axis) in any *conical section* to datum axis with a specified cone angle;

- Other total deviations.

The standard states 16 *accuracy degrees* for each type of tolerance for form and surface position. Numerical values of tolerances increase with coefficient 1.6 from one degree to another. The standard also states *levels of relative geometric accuracy* depending on the ratio between a tolerance of dimension and tolerances for form and surface position. These levels are the following: A – *normal* relative geometric accuracy (tolerances for form or surface position are approximately equal to 60 % dimension tolerance); B – *improved* relative geometric accuracy (tolerances for form or surface position approximately equal 40 % dimension tolerance); C – *high* relative geometric accuracy (tolerances for form or surface position approximately equal 25 % dimension tolerance).

Tolerances for form of cylindrical surfaces are approximately equal to 30, 20 and 12 % dimension tolerance respectively to the levels A, B and C, because tolerance of form limits deviations of radius, and tolerance of dimension – deviations of surface diameter.

It is permitted to limit the tolerances for form and surface position by tolerance of dimension.

The tolerances for form and surface position are specified only in those cases, when they should be less than tolerances of dimensions according to functional or manufacturing reasons.

10.1.4 Independent and dependent tolerances

Tolerances of form and position can be independent or dependent.

Independent tolerance is a tolerance, value of which is constant for all items produced by this drawing and does not depend on actual dimensions of a considered element (feature).

Dependent tolerance is a tolerance of position or form, minimum magnitude of which is permitted to exceed by the value depending on deviations from a maximum material condition (shaft maximum limit dimension or hole minimum limit dimension) of actual dimensions of a considered element and/or datum element:

$$\boldsymbol{T}_{\mathbf{dep}} = \boldsymbol{T}_{\mathbf{min}} + \boldsymbol{T}_{\mathbf{ad}},$$

where $\boldsymbol{T}_{\mathbf{min}}$ is a portion of dependent tolerance constant for all parts specified in a drawing (minimum value); $\boldsymbol{T}_{\mathbf{ad}}$ is an additional portion of dependent tolerance depending on actual dimensions of considered elements of an item.

As a rule, dependent tolerances are recommended for those elements of parts, which are assembled with guaranteed clearance.

Minimum value of dependent tolerance $\boldsymbol{T}_{\mathbf{min}}$ is specified in a drawing picture (ref. Fig. 10.4) or technical requirements. It can be enlarged at a value equals a deviation of actual dimension of a part element from maximum material condition.

At any of the methods for specifying limit deviations for position, first the minimum value of position tolerance $\boldsymbol{T}_{\mathbf{min}}$ in diametrical direction is determined (ref. Fig. 10.5). The position tolerance depends on type of joint (A or B), minimum designed clearance $\boldsymbol{S}_{\mathbf{des}}$ for a fastener and degree of use of this clearance for compensation of deviations of axes positions.

For the joint type A (fastening with bolts, rivets, when clearances are in both mating parts) position tolerance is calculated from

$$\boldsymbol{T}_{\mathbf{min}} = \boldsymbol{S}_{\mathbf{des}}.$$

For the joint type B (fastening with screws, studs, pins, when clearance is only in one part) position tolerance is calculated from

$$\boldsymbol{T}_{\mathbf{min}} = 0.5 \cdot \boldsymbol{S}_{\mathbf{des}}.$$

There are two methods for calculations of a designed clearance $\boldsymbol{S}_{\mathbf{des}}$.

The first method applies coefficient $\boldsymbol{K}$ for determination of degree of use of the minimum clearance for compensation of deviations of axes positions

$$\boldsymbol{S}_{\mathbf{des}} = \boldsymbol{K} \cdot \boldsymbol{S}_{\mathbf{min}},$$

where $\boldsymbol{S}_{\mathbf{min}}$ – guaranteed (minimum) diametrical clearance for a fastener stated by the standard.

The $\boldsymbol{K}$ coefficient equals 1.0 or 0.8 for joints that dot not need adjustment of relative position of parts; 0.8 or 0.6 (or even a smaller value) for joints that need some adjustment of relative position of parts.

Another method includes more careful consideration. The designed clearance $\boldsymbol{S}_{\mathbf{des}}$ per diameter under condition of complete interchangeability is calculated from the formula

$$\boldsymbol{S}_{\mathbf{des}} = \boldsymbol{S}_{\mathbf{min}} - \boldsymbol{S}_{\mathbf{adj}} - \boldsymbol{T}_{\mathbf{perp}} - \boldsymbol{T}_{\mathrm{C}},$$

where $\boldsymbol{S}_{\mathbf{adj}}$ – minimal radial clearance between through holes and a fastener for sequent adjustment of relative position of parts or for easier assembling; $\boldsymbol{T}_{\mathbf{perp}}$ – total tolerance for perpendicularity of holes axes to rest planes in all joining parts; $\boldsymbol{T}_{\mathrm{C}}$ – coaxiality tolerance for a stepped fastener or/and stepped hole.

Constant portion $\boldsymbol{T}_{\mathbf{min}}$ of dependent tolerance can be equal to zero (specified as zero). In this case total value of position dependent tolerance $\boldsymbol{T}_{\mathbf{dep}}$ is formed only by additional portion $\boldsymbol{T}_{\mathbf{ad}}$, that is, at the expense of dimensions tolerances of specified elements.

The $\boldsymbol{T}_{\mathbf{ad}}$ additional portion of dependent tolerance is calculated from the geometric formulas with the difference between actual dimensions of joined elements (for example, bolts and holes) and their dimensions at maximum material condition (maximum limit dimension for shafts and minimum limit dimension for holes).

Dependent tolerances of position are more economical for manufacturing than independent ones. They allow applying less accurate, but more economical methods of machining and technological equipment.

Dependent tolerances are specified for elements of holes and shafts, as well as for such characteristics as tolerances for position, coaxiality, symmetry, axes intersection, perpendicularity of axes or perpendicularity of axis and plane. Dependent tolerances are depicted in drawings by letter M in a circle (ref. Fig. 10.4) or described by text in technical requirements.

10.2 Surface roughness and waviness

10.2.1 Surface roughness and its influence on operation of machines parts

Operating properties of parts are greatly determined by quality of surface layer, by its geometric, physical and mechanical characteristics.

Real surface of a machined part consists of alternating small projections and depressions as a result of copying a form of cutting tool, plastic deformation of surface layer material, vibrations in technological system and other factors.

Respect to fatigue strength surface roughness is a critical factor, because its role is estimated as 75 %, while residual stresses – 20 %, strain hardening (cold-work strengthening) – 5 %.

Roughness of surface influences:

- Fatigue strength of a part – microrelief is considered as a stress concentrator – stresses in a valley surface are 2–2.5 times higher than average stresses in surface layer;
- Strength and reliability of fixed joints (interference fits);
- Reliability of movable joints under conditions of lubricated friction (Fig. 10.7):
- at larger roughness Ra > 0.32 μm wear occurs at the expense of deformation of microirregularities peaks;
- at smaller roughness Ra < 0.32 μm wear occurs at the expense of pressing out a lubricant from insufficiently deep valleys (“pockets”) and, hence, at the expense of dry friction;
- Capability to absorb (radiate) heat;
- Aerodynamic qualities;
- Corrosion resistance and other parameters.

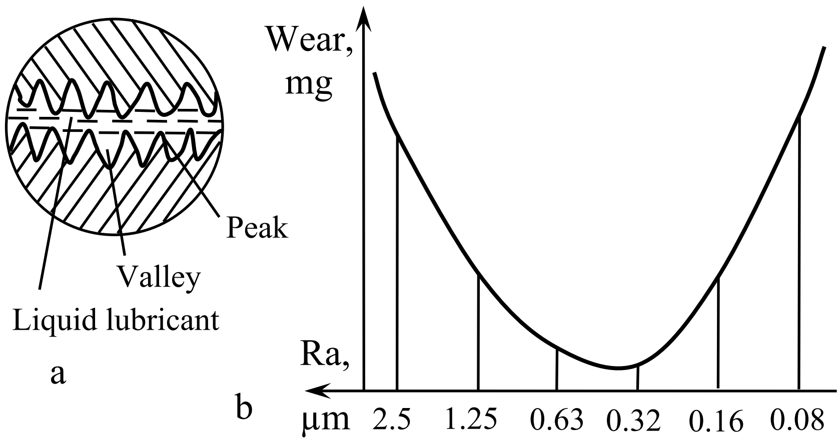


Figure 10.7 – Contact surfaces (a) and material wear removal under conditions of friction with lubricant (b)

10.2.2 Surface roughness parameters

Surface roughness is a totality of microirregularities located on surface with relatively small spacing in the range of sampling length $\boldsymbol{l}$ (Fig. 10.8).

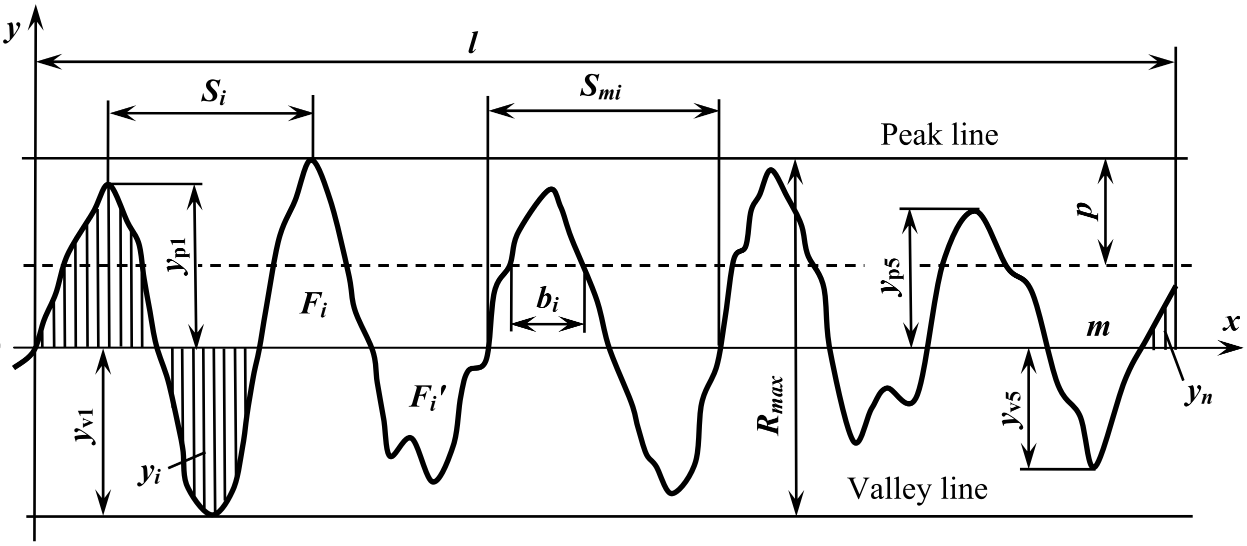


Figure 10.8 – Enlarged surface profile and roughness parameters

Base line, relative to which numerical parameters of roughness are determined, is a mean profile line $\boldsymbol{m}$.

Mean profile line $\boldsymbol{m}$ is a base line, which divides actual profile in such a manner that sum of squares of distances $\boldsymbol{y_i}$ from profile points to this line in the range of sampling length $\boldsymbol{l}$ is minimal, or total area of projections and total area of depressions are equal (ref. Fig. 10.8):

$$\int_0^{l} y^2 dx = min \text{ or } \sum_1^n F_i = \sum_1^n F_i' .$$

Such a system in the international practice is called the *system M*.

For quantitative estimation of roughness the standard introduces 6 parameters:

1) *Maximum height of the profile* $\boldsymbol{R_{max}}$ (ref. Fig. 10.8) is a distance between line of profile peaks and line of profile valleys along sampling length;

2) *Arithmetical mean deviation of profile* $\boldsymbol{R_a}$ is an arithmetical mean from absolute values of profile deviations in a range of sampling length $\boldsymbol{l}$

$$R_a = \frac{1}{l}\int_0^{l} |y(x)| dx \approx \frac{1}{n}\sum_{i=1}^{n} |y_i| ,$$

where $\boldsymbol{y(x)}$ – profile function; $\boldsymbol{y_i}$ – profile discrete deviation value in M system equal to distance between any point of profile and mean line measured along a normal plotted to mean line via this point (ref. Fig. 10.8); $\boldsymbol{n}$ – number of measured discrete deviations along profile sampling length;

3) *Ten-point height of profile irregularities* $\boldsymbol{R_z}$ is a sum of mean absolute values of heights of five largest peaks and depths of five largest valleys in a range of sampling length $\boldsymbol{l}$

$$\boldsymbol{R}_z = \frac{1}{5}\left(\sum_{\boldsymbol{i}=1}^{5}\left|\boldsymbol{y}_{\boldsymbol{pi}}\right| - \sum_{\boldsymbol{i}=1}^{5}\left|\boldsymbol{y}_{\boldsymbol{vi}}\right|\right),$$

where $\boldsymbol{y_{pi}}$ – height of the $\boldsymbol{i}$ largest peak (ref. Fig. 10.8); $\boldsymbol{y_{vi}}$ – depth of the $\boldsymbol{i}$ largest valley;

4) *Mean spacing of profile irregularities* $\boldsymbol{S_m}$ is a mean value of spacing along mean line $\boldsymbol{m}$ between like sides of neighbour irregularities in a range of sampling length $\boldsymbol{l}$

$$\boldsymbol{S_m} = \frac{1}{\boldsymbol{n}}\sum_{\boldsymbol{i}=1}^{\boldsymbol{n}} \boldsymbol{S_{mi}},$$

where $\boldsymbol{S_{mi}}$ – length of mean-line segment between two like sides of neighbour irregularities (ref. Fig. 10.8); $\boldsymbol{n}$ – number of segments along sampling length;

5) *Mean spacing of profile irregularities in crests* $\boldsymbol{S}$ is a mean spacing value of profile local peaks (crests) in a range of sampling length $\boldsymbol{l}$

$$\boldsymbol{S} = \frac{1}{\boldsymbol{n}}\sum_{\boldsymbol{i}=1}^{\boldsymbol{n}} \boldsymbol{S_i},$$

where $\boldsymbol{S_i}$ – length of segment between two neighbour peaks of profile along sampling length (ref. Fig. 10.8);

6) *Relative reference length of profile* $\boldsymbol{t_p}$ is a ratio of profile reference length to sampling length $\boldsymbol{l}$ (Fig. 10.8)

$$\boldsymbol{t_p} = \frac{1}{\boldsymbol{l}}\sum_{\boldsymbol{i}=1}^{\boldsymbol{n}} \boldsymbol{b_i},$$

where $\boldsymbol{p}$ – level of profile section – distance between peak line and line crossing the profile equidistantly expressed in percents of distance between peak line and valley line $\boldsymbol{R_{max}}$; $\boldsymbol{b_i}$ – segment length cut by line at the specified level $\boldsymbol{p}$ in material of peaks (ref. Fig. 10.8). Reference length of profile is a sum of segments lengths $\boldsymbol{b_i}$ along sampling line.

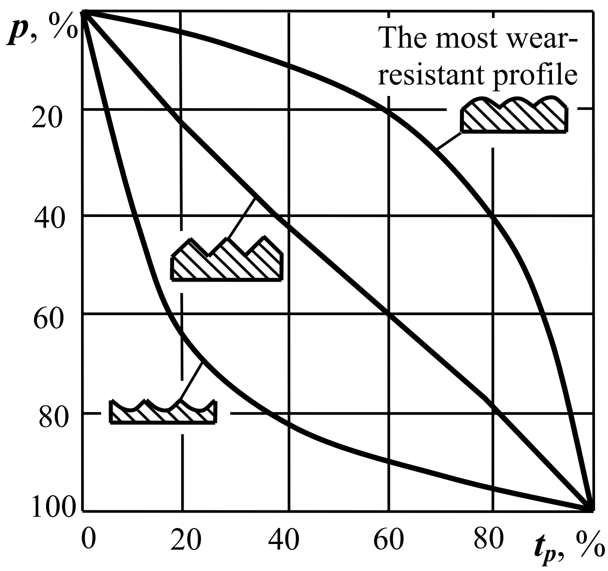


Figure 10.9 – Influence of valleys form on reference length of profile

The $\boldsymbol{t_p}$ value sufficiently describes shape of profile irregularities that allows to standardise many important operating surface parameters (Fig. 10.9). The sharper peaks of profile irregularities, the smaller reference length and wear resistance are.

International Organization for Standardization introduced numerical values of roughness parameters in the standard ISO 1302 (Table 10.2).

Table 10.2 – Preferred and recommended numerical values of surface roughness parameters

$\boxed{\boldsymbol{R_a}}$, µm		14 preferred(total 48)														
	400	100	50	25	12.5	6.3	3.2	1.6	0.8	0.4	0.2	0.100	0.050	0.025	0.012	0.008
$\boldsymbol{R_z}$, $\boldsymbol{R_{max}}$, µm		15 preferred (total 49)														
	1600	400	200	100	50	25	12.5	6.3	3.2	1.6	0.8	0.4	0.2	0.100	0.050	0.025
l, mm		5 recommended														
	25	8			2.5		0.8			0.25				0.08		0.03

10.2.3 Surface roughness specifying in drawings

Surface roughness is specified in a drawing for all surfaces of a part.

Structure of roughness designation is shown in Fig. 10.10. The height ***h*** of roughness symbol should be equal to the height of dimensions digits. The height ***H*** equals (1.5…5.0)***h***. Thickness of lines of roughness symbols approximately equal half-thickness of basic solid lines in a drawing.

Numerical values of roughness parameters are specified after corresponding symbol, for example, $R_a0.8$, R_z20, $R_{max}10$.

Sampling length is not written, if it equals to the standardised recommended value (ref. Table 10.2).

In a designation of surface roughness a designer may use the following symbols.

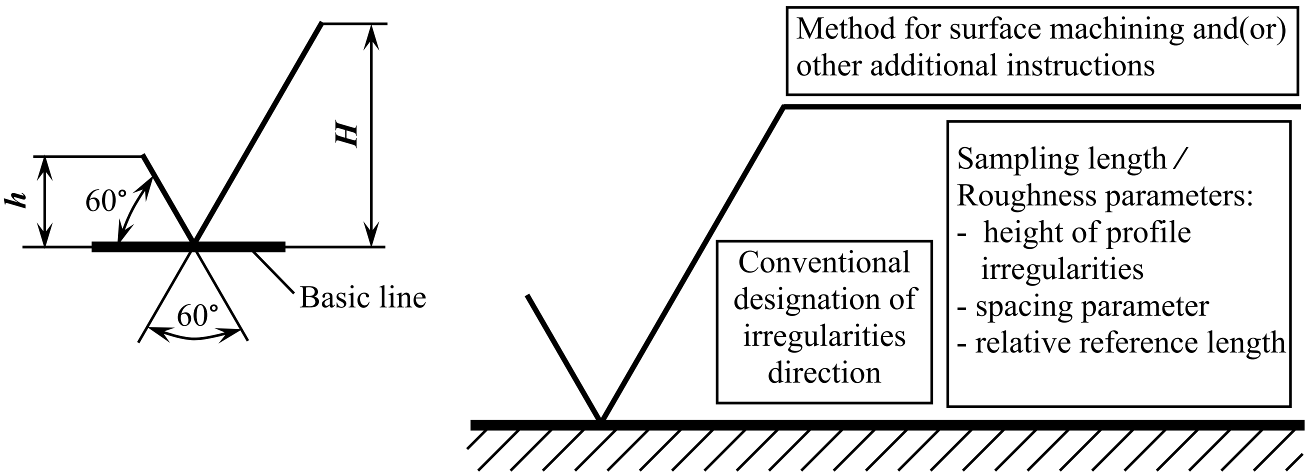


Figure 10.10 – Structure of designation of surface roughness parameters in a drawing

Symbol, shown in Fig. 10.11,a, means that method of processing is not specified by a designer. This symbol is preferred for application, because it does not limit a production engineer in selection of processing method. Another symbol (ref. Fig. 10.11,b) is used, when methods with removal of material from the surface should be applied (metal-cutting processes). When surface should be created without material removal (casting, rolling, forging, etc.), the symbol depicted in Fig. 10.11,c should be used.

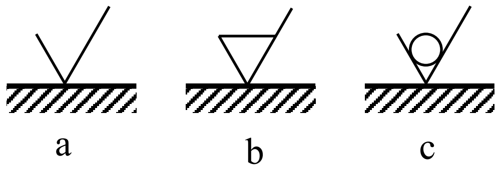


Figure 10.11 – Types of symbols depending on methods of processing

In a drawing designation of critical surface roughness parameters can be submitted in a full form (Fig. 10.12,a) or in less complicated form (Fig. 10.12,b). According to the designation structure (ref. Fig. 10.10) the sampling length is shown before a roughness parameter. If sampling length is equal to the recommended value, it is not written down (compare parameter Ra in Fig. 10.12,a and 10.12,b). Roughness parameters can be specified by two limit values – maximum and minimum (see parameter $\boldsymbol{S}_{\mathbf{m}}$ in Fig. 10.12,a) and by nominal value and deviations (see parameter $\boldsymbol{t}_{\mathbf{50}}$ in Fig. 10.12,a)

Typically surface finish is specified in a simple form (Fig. 10.12,c). Actual surface roughness must not be larger than the specified value.

When specifying similar roughness for several surfaces of a part, a designation of similar roughness and roughness symbol in parentheses are depicted in a right upper corner of a drawing (Fig. 10.12,d). The symbol in parentheses means that all surfaces, which have no designation of roughness in a picture of part, should have the roughness value shown before the symbol in parentheses.

Sizes and lines thickness of the roughness symbols in the right upper corner should be approximately 1.5 times more than in the symbols depicted in the picture of part (ref. Fig. 10.12,d).

When necessary, irregularities direction is shown by special standard symbols (Table 10.3).

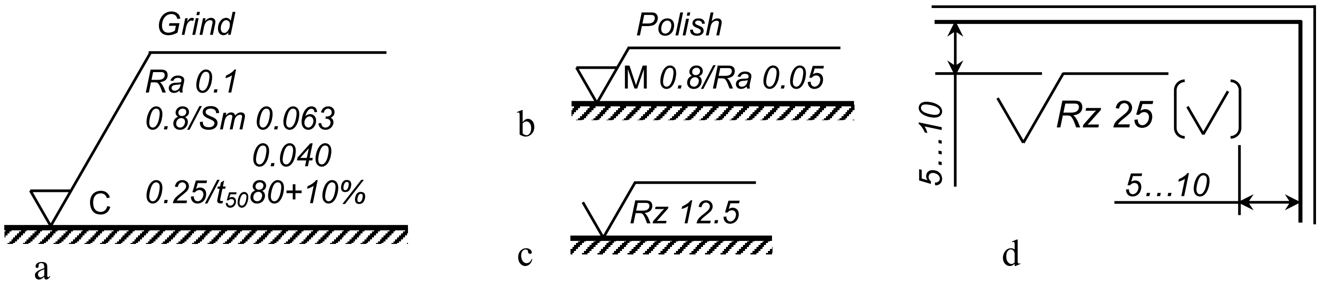
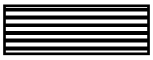
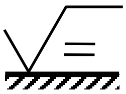
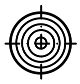
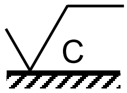
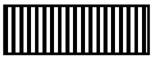
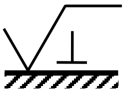
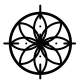
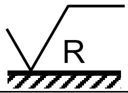
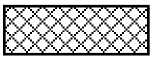
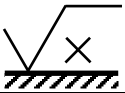
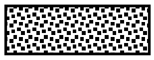
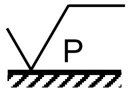
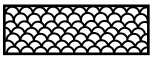
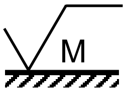


Figure 10.12 – Examples of specifying the surface roughness parameters in drawings

Table 10.3 – Preferred and recommended numerical values of surface roughness parameters

Types of irregularities directions	Symbol	Types of irregularities directions	Symbol
Parallel 		Circular 	
Perpendicular 		Radial 	
Crosswise 		Punctated 	
Arbitrary 			

10.2.4 Surface waviness parameters

Deviations from the perfect smooth surface can be of three types:

- Surface roughness: $\dfrac{\textit{irregularities interval } \boldsymbol{S_m}}{\textit{height of irregularities } \boldsymbol{R_z}} < 50$;

- Surface waviness: $\boldsymbol{S_m}/\boldsymbol{R_z} \approx (50 \ldots 1000)$;

- Form deviations (taper, barrel, bow): $\boldsymbol{S_m}/\boldsymbol{R_z} > 1000$.

Irregularity of cutting forces, presence of unbalanced masses, errors of machine-tools and other factors result in vibrations in technological system “machine-fixture-workpiece-tool”, which are a cause of waviness of machined surface.

Waviness of surface is a combination of periodically repeated irregularities with distance between neighbour peaks or valleys exceeding the sampling length $\boldsymbol{l}$. Waviness are estimated by 2 parameters: waviness height $\boldsymbol{W_z}$ and waviness spacing $\boldsymbol{S_w}$.

Height of waviness $\boldsymbol{W_z}$ is an arithmetic mean from 5 waviness values located in series along measuring length.

Mean spacing of waviness $\boldsymbol{S_w}$ is a mean value of distances between like sides of neighbour waves measured along mean line of profile.

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