Introduction

Under modern conditions of science and technology, under the organized large scale production, standardization, based on the wide introduction of the interchangeability principles, is one of the most effective means of promoting progress in all areas of economical activity and improving the quality of output items.

In market conditions in all branches of industry, on the one hand, the issue of the quality of manufactured items became acute; on the other hand, modern production can not exist without its constant improvement and development. Basic framework for this is interchangeability at all levels of production. Course objective of “Fundamentals of interchangeability” is to acquaint students with the basics of achieving a projected accuracy in the items quality, as well as getting them the necessary knowledge on the methods of items interchangeability and ways to achieve them.

The learning guide summarizes the theorises necessary for solving examples and problems. The basis is a unified system of fits and limits for slick and cylindrical joints and recommendations on the basis of which standards for the limits of metric threads, plain and thread gauges, cylindrical gear pairs and surface roughness parameters have been developed in recent years.

The main focus of the manual is given to typical examples and tasks common to all engineering specialties.

The learning guide aims are to consolidate the theoretical provisions of the course, presented in lectures, to develope skills in the use of reference materials, as well as to acquaint students with the main types of calculations.

Interchangeability, fits and limits

At modern engineering factories of batch production and mass production, the processes of parts manufacturing and their assembly into separate units and machines are carried out in different shops. In this case, they use standard fasteners, rolling bearings, rubber technical and electrical items obtained through cooperation from other enterprises. In spite of this, assembly of units and machines must be done without fitting the parts, which is possible only when they are interchangeable.

Modern mechanical engineering is characterized by:

- continuous increase in machines capacity and productivity;
- constant improvement of the machines design and other items;
- increasing requirements for the accuracy of machines manufacturing;
- growth of mechanization and automation of production.

For the successful development of mechanical engineering it is a great importance to organize the machines and other items production on the basis of interchangeability and standardization.

Interchangeability of goods (machines, equipment, mechanisms etc.), their parts or other types of products (raw materials, materials, semi-finished items, etc.) is the ability of the item to substitute when using any of many copies of items, their parts or other items for another identical type.

Complete interchangeability is the most widely used, it provides the ability to assemble without fitting (or to replace during repair) any part that is independently manufactured with the specified accuracy into assembly units, and the latter - into items, subject to the requirements for them (to assembly units or items) is the technical requirements of all the quality parameters. The complete interchangeability is only possible when the dimensions and shape, mechanical, electrical and other quantitative and qualitative characteristics of parts and assembly units after manufacturing are within the specified limits and the assembled items meet the technical requirements. Under complete interchangeability, the assembly is performed without finalizing the parts and assembly units. Such a production is called interchangeable. Parts, assembly units and items in general can be interchangeable. First of all, the details of the assembly units should be interchangeable, because reliability and other performance indicators depend on them.

In the case of **incomplete** (limited) interchangeability, group breeding of parts (selective assembly), compensators, position regulation of some parts of machines and devices, fitting and other additional technological measures are applied with mandatory fulfillment of the requirements for the quality of assembly units and items. It can be implemented not to all, but only to certain geometric or other parameters.

Level of interchangeability of production can be characterized by the interchangeability coefficient I_c, equal to the ratio of laboriousness of manufacturing interchangeable parts L_i to the total laboriousness of manufacturing the item L_Σ.

Interchangeability coefficient of I_c ranges within $0 < I_c \leq 1$.

At the same time, the fulfillment of the specified requirements for the accuracy of parts, assemblies, assembled units or their elements is the crucial and defining condition for ensuring interchangeability.

The unification coefficient U_c (standardization) of production is closely related to the interchangeability coefficient I_c, and it is defined as the ratio of the laboriousness of assembling unified (standard) parts L_{st} to the laboriousness of assembling original parts L_{org}

$$Uc = \frac{L_{st}}{L_{org}}; \quad Uc > 0. \tag{1}$$

The implementation of one or another type of interchangeability in production is influenced by many factors, most important of which are the type of production, the type of output, the degree of development of production relations, the culture of production.

In general, with the increase of the I_c, the performance characteristics of the item are improved, as conditions are created for rapid and effective replacement or recondition of parts, assemblies or assembly units of the item during repairs, maintenance and scheduled tasks of the preventive maintenance system. However, in individual and low-batch production, there is a poor relationship due to the limited number of items produced. The I_c greatest influence on the performance characteristics is observed in mass and large batch production. The I_c influence on the performance characteristics depend very heavily on the type of item: the largest – in the electronic industry, slightly less – in instrument-making industry, much less – in machine engineering.

The complete interchangeability is economically reasonable to implement for parts manufactured in mass and batch production and having accuracy not higher than the sixth grade, as well as for assembly units and items consisting of a small number of parts violation of specified clearance or interference during assembling items in machine and instrument making is unacceptable even for some parts.

Modern industry cannot develop without broad cooperation, the basis for which is interchangeability. The design of effective technological processes and their practical implementation is also impossible without taking into account the interchangeability of parts, assemblies and structures. Quality of items and its monitoring is carried out on basis of techniques developed by the practice of using various types of interchangeability. Thus, interchangeability has been identified as an independent research and production field in many industries, which plays a pivotal role in achieving high quality items and ensuring its competitiveness in the world market.

Beside complete and incomplete interchangeability, there are external, internal and functional interchangeability of parts and assembly units.

External interchangeability is the interchangeability of purchased and cooperated parts (mounted in other more complex parts) and assembly units by performance indicators, as well as the dimensions and shape of the joint surfaces. For example, in an electric motor, external interchangeability is ensured by the speed of rotation of the drive shaft, and also by the dimensions of the joint surfaces. In rolling bearings, external interchangeability is ensured by the outer diameter of the outer race installed in the body of the item and the inner diameter of the inner race installed on the shaft, as well as the accuracy of rotation and the load accommodation.

Internal interchangeability extends to parts, assembly units and mechanisms included in the item. For example, for the selective assembly of rolling bearings, the internal group interchangeability has rolling elements and races.

Compatibility is the property of objects to occupy their place in a complex finished item and to perform the required functions at simultaneous and consecutive operations of these objects and a complex item under the given operational characteristics.

Object is an autonomous unit, device, or other item that is part of a complex item.

Functional interchangeability is interchangeability provision of machines and other items for optimal performance indicators, which is the main principle of interchangeability of items and machines in general. Therefore, in a more generalized representation, interchangeability, which ensures the operability of items or their consumer properties with optimal and stable (within given limits) in time performance indicators or optimal performance indicators, is called **functional**.

At the same time, functional parameters include geometric, electrical, mechanical and other parameters that affect the reliability or economic performance of machines and other items, or the service functions of assembly units. For example, the engine power (performance indicator) depends on the limit between the piston and the cylinder (the functional indicator, determined by the limit for the size of parts). These parameters are called functional, as there is a connection with the service functions of the assembly units and the operation of the specified item.

Such a relationship can be both regular and random. To obtain the greatest efficiency of interchangeability, i.e. to achieve a functional interchangeability, it is necessary to take into account a set of scientific and technical reference points in the design, manufacture and operation of machines and items, which are subsumed under the generic term – **fundamental interchangeability** – and which are determined by the «life» cycle of the item.

The reference points used in the design of items:

a) the operational characteristics of machines and other items are determined by the level and characteristics stability of the technological process; dimensions, shape and other geometric parameters of parts and assembly units; the level of mechanical, physical and chemical properties of the materials from which the parts are made, and other factors. Inevitable inaccuracies of parameters and changes in the properties of materials affect the parameters of the technological process and the operational characteristics of machines, so for critical parts and component parts it is necessary to provide interchangeability not only in size, shape and other geometric parameters, parameters of the mechanical properties of the material (especially the surface layer of the parts), but also in electrical, hydraulic, optical, chemical and other functional parameters (depending on the operating principle of the machine);

b) it is very important to ensure uniformity of raw materials, materials, blanks and semi-finished goods in terms of their chemical constitution and structure, equal level and stability of the sizes and shapes;

c) functional interchangeability is provided at the item design stage. First of all, this requires clarifying the nominal values of their performance indicators of items, which they must have at the end of the established period of work. The difference between these indicators for new items and at the end of their service life is their limit. There is another way to solve this problem: generalization of operational experience and carrying out of experimental tests of models, simulators or samples. It is important to establish the basic components of the machine, on which its performance indicators primarily depend; make a list of parts and components that determine the longevity of the item overall. For this category of parts and component parts of the item, then, design shape, materials, manufacturing techniques are selected and surface quality is established, which ensure maximum service life, accuracy and other characteristics;

d) when designing, it is necessary to identify the functional parameters on which the values and allowable range of deviations of the machine performance indicators depend. Theoretically and experimentally on simulators, models and prototypes, it is necessary to establish possible changes in the functional parameters over time (as a result of wear, plastic deformation, thermo cyclic influences, recapitalization and material aging, corrosion, etc.); find the relationship and the degree of influence of these parameters and their deviations on the performance indicators of the new item and in the process of its continuous exploitation. Knowing these relationships and limits on the performance of items, you can determine the limits of the functional parameters and calculate the tolerances for critical connections. Another method is also used: using established relationships, determine the deviations of performance indicators with the selected limits of the functional parameters. When calculating the accuracy of the functional parameters, it is necessary to create a guaranteed space of working capacity of the items, which will ensure the maintenance of performance indicators at the end of their service life within the specified limits. It is also necessary to optimize the limits by setting smaller limits for functional parameters whose inaccuracies most strongly affect the performance of the items. Making connection between performance indicators and functional parameters and independent manufacturing of parts and components according to these parameters with an accuracy determined on the basis of allowable deviations in the performance of items at the end of their service life is one of the main conditions for ensuring the functional interchangeability;

e) when designing products it is necessary to apply more widely the common technical regulations, universal and standardized parts and assembly units, and also should be guided by the principles of preference and aggregation, because in modern conditions without this it is impossible to provide high quality items and economy of production;

f) to ensure the interchangeability of critical parts in terms of roughness, shape and location of their surfaces, these parameters should be chosen so that wear of parts is minimal and performance is optimal;

g) When designing, it is necessary to take into account the technological requirements and allow for the choice of the accuracy parameters of parts, assembly units and an item of such a measurement scheme that does not introduce additional inaccuracies and allows the use of simple and reliable universal or existing special measuring means.

Thus, the generation of drawings and technical requirements, indicating the accuracy of dimensions and other parameters of parts, assembly units and items, ensuring their high quality, is the first component of the interchangeability principle, performed in the process of item design. Technologists and metrologists are usually involved in the development of drawings and technical conditions in order to better align the design, technological and metrological requirements and to ensure the possibility of using advanced technology for manufacturing parts, assembling machines and devices. The engineering drawing, in which the precision requirements are specified, is the initial and directive document on which the technological processes are designed and controlled, and also the accuracy of the parts, components and finished products is checked.

The initial positions in the manufacture of items:

a) to ensure interchangeability in the manufacture of parts and assembly of products, it is necessary to strictly maintain the normalized accuracy of the functional parameters;

b) to create a large stock of machine runnability for critical functional parameters, it is advisable to ensure the fulfillment of the condition that the operating limits of the parameters were higher than technological limits, with the accepted technological process.

The accuracy of equipment, tools and manufacturing jigs, as well as their preventive control has great importance for the interchangeability implementation and the achievement of high quality items. The accuracy of the equipment and tooling should be several times higher than the required accuracy of the parts and components produced, i.e. accuracy stock is required;

c) for critical parts, it is necessary to create an optimal surface quality;

d) to ensure interchangeability and high quality of machines and other items, it is necessary that the technological and measuring bases coincide with the constructive ones, i.e. it is necessary to observe the unity principle of bases. In addition, the measuring diagram must correspond to the working motion diagram of the part in the mechanism. This requirement is met, for example, in the single-flank control of gears.

The initial positions in the item operation:

a) an important part of the interchangeability principle implementation, which determines the continuous and economical operation of items, is the identification of the necessary set of spare parts (components and assembly units). They would ensure a quick replacement during the operation of worn out or broken parts or assembly units, while maintaining the required performance of the machine during long time. For this, an analysis must be carried out and the "weak points" of the item must be identified, i.e. the parts and assembly units that are most affected by wear and that have an impact on the performance;

b) in the process of operation, it is necessary to monitor carefully the machine operation and pay special attention to the «weakest» elements;

c) repair of worn out parts of machines and other items is advisable to produce at special repair plants by replacing them with suitable parts.

So, for practical implementation of the principle of functional interchangeability of items, a clear system of design, process control, metrological and operational documentation is necessary.

It is especially important to ensure the functional interchangeability of parts and items obtained by waste-free technology, in which mechanical processing is minimized. This will increase the efficiency of waste-free technology, not only with respect to material savings, but also a sharp rise in labor productivity and product quality.

In mechanical engineering, the nominal, actual and limiting sizes are distinguished according to GOST (All-Union State Standard) 25346-89.

Size is the numerical value of a linear value (diameter, length, etc.) in the selected units of measure (in engineering usually in millimeters). By designation, there are sizes that determine the magnitude and shape of the part: the coordinating dimensions that determine the mutual position of the critical surfaces and the axes of the parts necessary for the correct operation of the mechanism: assembly and mounting dimensions that characterize the application of joints to the connecting dimensions. In addition, there may be working dimensions that are necessary directly for the manufacture of parts and their control.

Nominal size (D, d, l, etc.) is the size that serves as the start point of the deviations and relative to which the limiting dimensions are determined. Nominal size is obtained as a result of calculation for strength and rigidity and is selected by rounding, as a rule, upward of the standard range of normal sizes in accordance with the instructions of GOST 6636-69. Dimensions in the drawings are in fact different from the dimensions of the specific parts manufactured according to this drawing, that is, from the actual dimensions, which are determined by engineering measurements.

Actual size is the dimension set by the measurement with an allowable inaccuracy.

Limiting dimensions of piece (D_{max}, D_{min} – for the hole, d_{max}, d_{min} for the shaft) are the two allowable limiting dimensions between which the actual size of the work piece must be or can be equal. The larger of them is called the largest limiting dimension, the smaller is the smallest limiting dimension.

In engineering drawings, the nominal and limit dimensions are set in millimeters without specifying the dimension. Other units of measurement are indicated in the appropriate size or technical requirements. Limit deviations in the limit tables are displayed in micrometers.

Maximum material limit is a term applied to that of two limiting dimensions, which corresponds to the maximum amount of material.

Minimum material limit is a term applied to that of two limiting dimensions, which corresponds to the minimum amount of material.

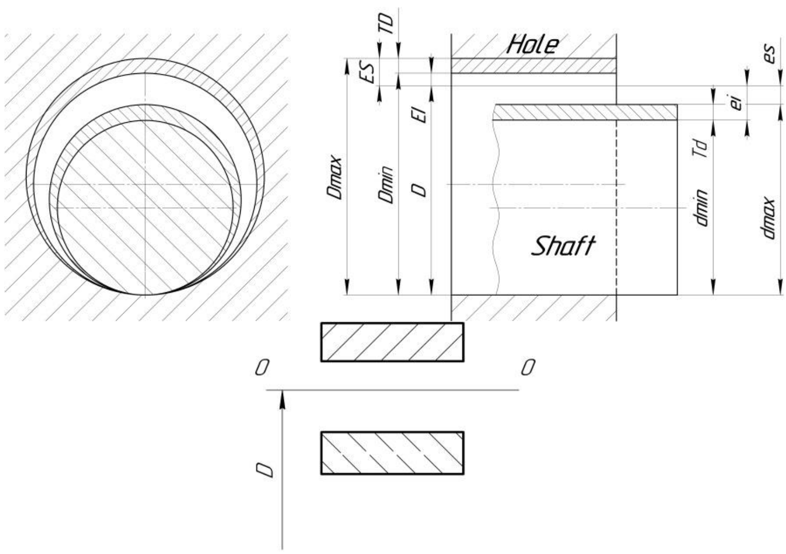
Zero line is a line corresponding to the nominal size, from which dimensional deviations are laid off in the graphical plot of fits and limits.

Upper limit deviation *ES, es* is the algebraic difference between the largest limiting and nominal sizes.

Lower limit deviation *EI, ei* is the algebraic difference between smallest limiting and nominal sizes.

The actual deviation is the algebraic difference between the real and nominal dimensions. The deviation is positive if the limit or actual size is larger than the nominal, and negative if the specified dimensions are less than the nominal size.

Symbols of nominal and limiting dimensions, limiting deviations and limits of holes and shafts are shown in figure 1.1, *a*. For simplicity, the limits can be represented graphically in the form of tolerance limit (figure 1, *b*).



a) *b*)

Figure 1 – Is a layout of tolerance limits of holes and shafts

The relationship between these parameters is expressed by the following formulas

$$D_{\max} = D + ES, \qquad d_{\max} = d + es, \tag{2}$$

$$D_{\min} = D + EI, \qquad d_{\min} = d + ei. \tag{3}$$

Shaft is a term used to refer to external (female) elements (surfaces) of parts.

Hole is a term used to refer to internal (female) elements (surfaces) of parts.

Basic shaft is a shaft which upper deviation is zero (*es = 0*).

Basic hole is a hole which upper deviation is zero (*EI = 0*).

Size limit *T* is the difference between the largest and smallest limiting dimensions or the absolute value of the algebraic difference between the upper and lower deviations. Limit is always positive.

The tolerance limit is a tolerance limited by the upper and lower deviation.

The dimensions limits of the female and male surfaces are abbreviated as the limit of the hole *TD* and the limit of the shaft *Td* and are expressed by formulas

$$TD = D_{\max} - D_{\min}, \qquad Td = d_{\max} - d_{\min}, \tag{4}$$

$$TD = ES - EI, \qquad Td = es - ei. \tag{5}$$

Limits of holes and shafts for different accuracy degrees and nominal sizes, with some exceptions, are calculated by the formula

$$IT = ai \text{ or } IT = aI \tag{6}$$

where a is the number of limit units (accepted for different accuracy degrees according to the table in Annex B);

i (I) is a limit unit, micron (Annex D).

For the dimensions less than 500 mm

$$i = 0{,}45\sqrt[3]{D_m} + 0{,}001D_m. \tag{7}$$

For the dimensions more than 500 mm

$$I = 0{,}004D_m + 2{,}1 \tag{8}$$

where D_m is geometric mean of the extreme interval values D' and D''.

$$D_m = \sqrt{D' \cdot D''}. \tag{9}$$

Error limit is a measure of the accuracy of the size: the smaller the limit, the higher the required precision of the part, the smaller the variation in the actual dimensions of the parts and, consequently, the variation of clearance or interference in the joint. Conversely, low accuracy is characterized by a high limit. Limit directly affects the complexity of manufacturing and the cost of parts: the larger the limit, the easier and cheaper the manufacturing. The choice of equipment and means of control, and the qualification of the personnel depend on the limit.

Using the tolerance unit and the coefficient *a*, for each grade of accuracy there are series of limits of all sizes covered by this dimension-limit system. Currently, there are system of fits and limits for sizes up to 1 mm, from 1 to 500 mm, over 500 mm to 3150 mm, over 3150 to 10000 mm. Simplifying the tolerance of fits and limits, each of the size ranges, in turn, is divided into several intervals and the limits are assumed to be the same for all sizes combined in one interval (for example, over 6 to 10 mm, over 10 to 18 mm, etc.). The diameters along the intervals are distributed in such a way that the limits calculated at the extreme values in each interval differ from the limits calculated by the average value of the diameter in the same interval by no more than 5 %–8 %.

Fit is called the way of parts joint, determined by the size of the clearance or interference. The fit characterizes the freedom of relative parts movement or the degree of resistance to their mutual displacement.

Two or more movable or immovably connected components are called **conjugated**. The surfaces by which the parts are joined are called **interfaced**. The rest surfaces are called **conjugated** (free).

Clearance, *S* is the difference in the size of the hole and shaft, if the size of the hole is larger than the size of the shaft. The clearance provides the possibility of relative movement of the assembled parts.

Interference, *N* is the difference between the dimensions of the shaft and the hole before the assembly, if the shaft dimension is larger than the hole one. The interference provides mutual immovability of parts after their assembly.

Clearance and interference for fits are calculated by the following formulas maximum clearance $S_{\max} = D_{\max} - d_{\min} = ES - ei,$ (10)

minimum clearance $S_{\min} = D_{\min} - d_{\max} = EI - es,$ (11)

maximum interference $N_{\max} = d_{\max} - D_{\min} = es - EI,$ (12)

minimum interference $N_{\min} = d_{\min} - D_{\max} = ei - ES.$ (13)

Clearance fit is a fit**,** which provides a clearance in the joint (the hole tolerance limit is located above the shaft tolerance limit).

Interference fit is a fit**,** which provides an interference in the joint (The hole tolerance limit is located under the shaft tolerance limit).

Transitional fit is a fit, at which it is possible to obtain both clearance and interference (the tolerances of the hole and shaft overlap partially or completely).

Tolerance fit *TN* is equal to the sum of the the hole and shaft tolerances that make up the connection

$$TD + Td = TS = TN \tag{14}$$

where $TS = S_{\max} - S_{\min}$ is a tolerance fit with interference;
$TN = N_{\max} - N_{\min}$ is a tolerance fit with clearance.

Tolerances and deviations are specified in the standard tables for parts whose dimensions are determined at a normal temperature of +20 °C. Such a temperature is accepted as close to the temperature of the working place of machine-building and instrument-making plants. At this temperature graduation and attestation of linear and angular measures of measuring instruments and instruments are carried out and measurements must be made. In the production it is accepted to observe the following conditions of the normal temperature regime:

a) the temperature of the part and the measuring means at the time of inspection should be the same that can be achieved by jointly exposure of the part and the measuring means under the same conditions (for example, on a cast-iron plane);

b) it is desirable that the coefficient of linear expansion of the part material and the measuring means be approximately the same. It should be pointed out that the measurement error also arises from local heating. For example, under the influence of the hand controller's heat for 15 minutes, the size of the staple for checking the shafts with a diameter of 175 mm changes by 8 μm, and the brackets for checking the shaft diameter of 280 mm – by 11 μm. This indicates the need to apply thermal insulation, for example, heat-insulating pads and handles for staples or thermo-insulating gloves.

The measurement error that occurs during the measurement as a result of temperature deformations is determined by the formula

$$\Delta l = l(\alpha_1 \Delta t_1 - \alpha_2 \Delta t_2)$$

where l is a measuring size;

α_1, α_2 are linear expansion coefficient of the material, respectively, to the part and the measuring instrument (Annex B);

$\Delta t_1 = t_1 - t_0$, $\Delta t_2 = t_2 - t_0$ are the difference between the temperature of the part, or the measuring means, and the normal temperature.

The change in the calculated clearance or interference due to temperature deformations is calculated by formulas

$$\Delta s_t = D(\alpha_D \Delta t_D - \alpha_d \Delta t_d),$$

$$\Delta N_t = D(\alpha_D \Delta t_D - \alpha_d \Delta t_d)$$

where D is a nominal joint diameter, mm.

If Δs_t or ΔN_t are the positive values, then there is a clearance or interference in the joint, respectively.

The limit deviation is listed immediately after the nominal dimensions by the following GOST 2.307-2011 methods:

1) conventional signs of fits and limits fields;

2) numerical values;

3) conventional signs and numerical values, which are placed to the right of the conventional sign in parentheses (figure 2).

The third method is used in cases where the limit deviations are assigned:

- to dimensions not included in the series of normal linear dimensions, for example $41{,}5H7\left(^{+0{,}25}\right)$;

- to certain types of items and their elements, for example, to the grooves for the keys (figure 3, *a*);

- to the size of the ledges with an asymmetric tolerance limits (figure 3, *b*);

- to the holes along the shaft system (figure 4, *c* is correct, figure 4, *d* is incorrect).

Limit deviations are placed above the dimension lines within the extension lines (figure 2), above the extension of the dimensional lines (figure 4) or above the extension lines (figure 6, *e*, *f*). When writing numerical values, limit deviations are necessarily indicated with signs.

The upper deviation is placed above the lower one, fiting the number of decimal places in both deviations: $\varnothing\ 60^{+0,051}_{+0,032}$ mm, $\varnothing\ 60^{-0,030}_{-0,049}$ mm ($\varnothing 60^{-0,03}_{-0,049}$mm is incorrect).

Limit deviations equal to zero is not indicated: $\varnothing\ 60^{+0,019}$ mm or $\varnothing 60_{-0,019}$ mm; in the first case $EI = 0,$ in the second – $es = 0.$

With a symmetric tolerance limits, the absolute value of the deviations is indicated once by numbers which height is equal to the height of the values accepted for the nominal dimension: $\varnothing\ 60 \pm 0,06$ mm.

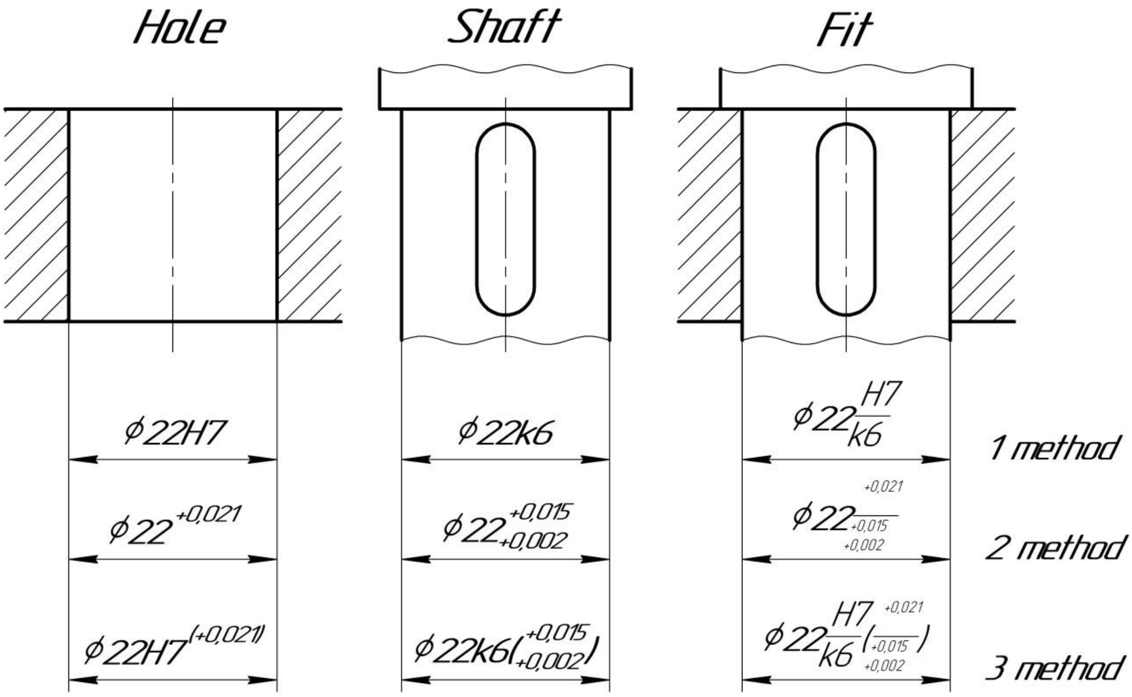
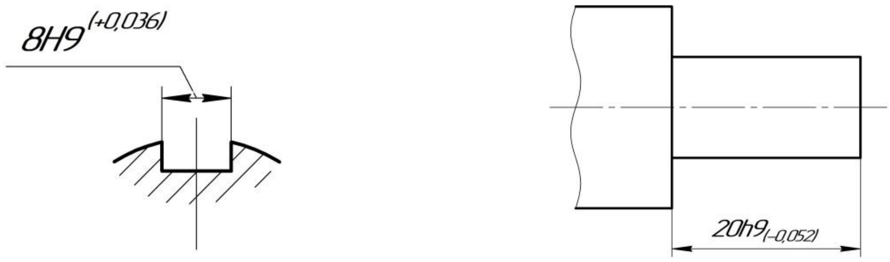


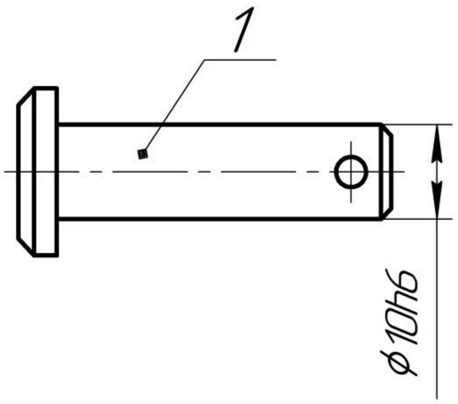
Figure 2 – Is an indication of limit deviations of dimensions on drawings



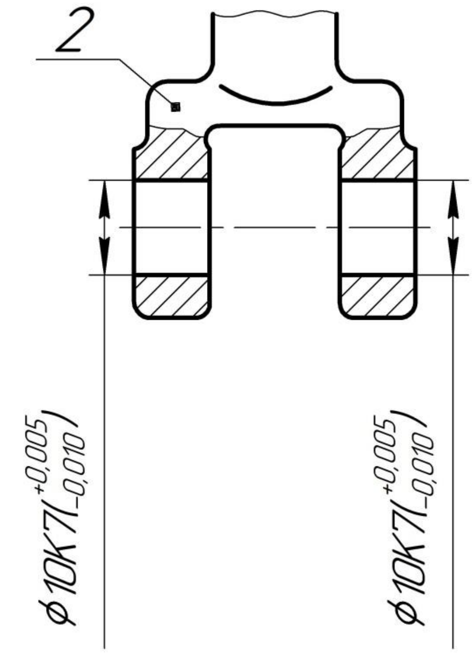
a) *b*)

Figure 3 – Is an example of applying limit deviations on the drawings by a third method

In the general assembly drawings, one of the three indicated methods is indicated by the tolerance limits and the limit deviations of the accepted fit (see figure 2). Above the line (when writing the limit deviations in the line in the first place), the conventional signs of the tolerance limits or the numerical values of the limit deviations of the hole is placed, and below the line or on the second place is the sign or the limit deviations of the shaft. This principle is followed regardless of the system to which the adopted fit applies.



a)



c)

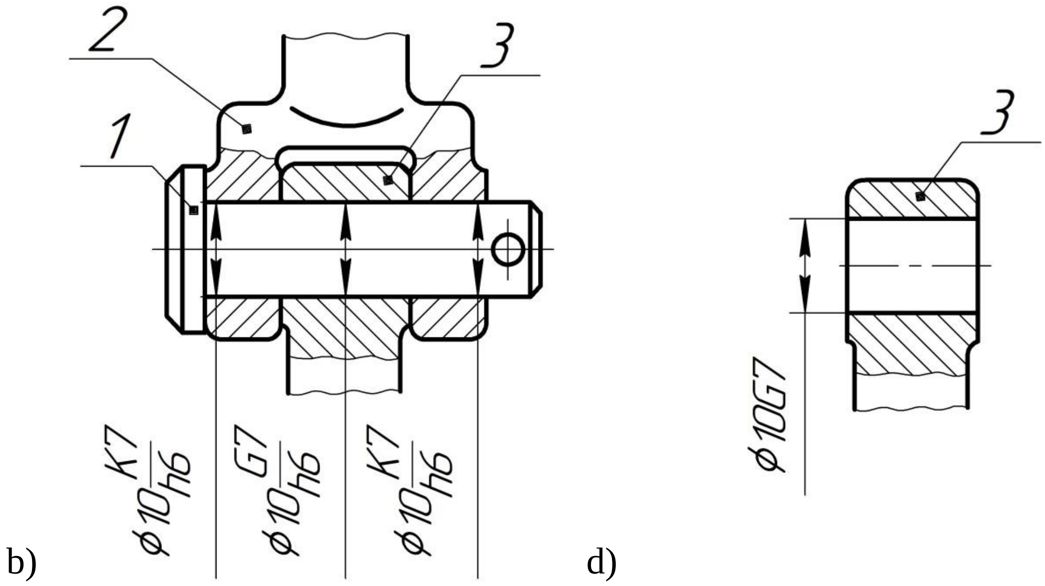


Figure 4 – Is an example of applying limit deviations on the drawings

If the limit deviations referring to only one conjugated component are indicated on the assembly drawing, then specify the details of these deviations (figure 5). If individual sections of the surface with the same nominal dimension are processed according to different limit deviations, then these sections are separated by a thin line, and the nominal dimension is indicated with the corresponding limit deviations for each section separately (figure 6 *a*). If it is necessary to limit the value of the distance error between repeating elements (figure 6, *b*), then these data are indicated in the technical requirements.

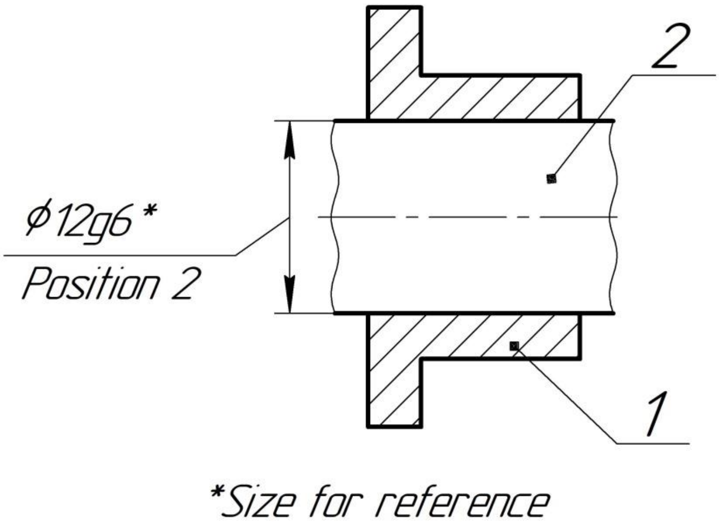


Figure 5 – Is example of limit dimensional deviations on the drawings

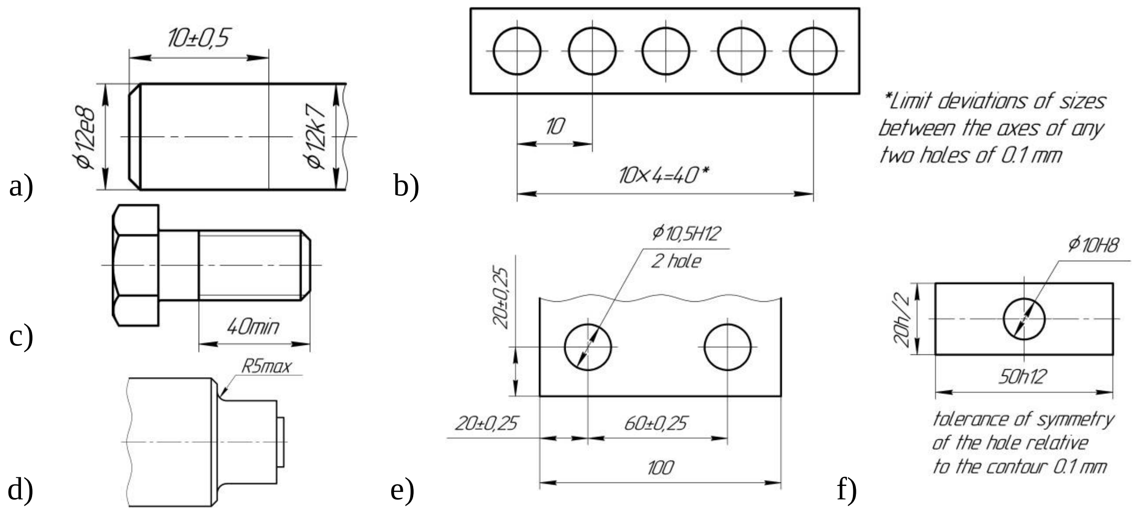


Figure 6 – Is example of limiting dimensional deviations on the drawings

In the case where it is necessary to specify only one limit dimension (the second is limited to increasing or decreasing by some condition), after the nominal size, *max* or *min* are specified, respectively (figure 6, *c*, *d*).

The limiting deviations of the holes axes location can be indicated by the limit dimensional deviations coordinating the axes (figure 6, *e*) or the limit displacement of the axes from the nominal position (figure 6, *f*).

If one part during the machining or assembling is adjusted over the conjugated component or involves the joint processing of the conjugated components, then the corresponding inscriptions are made in the technical specifications on the drawings.

The limiting deviation is not indicated for:

- reference dimensions, these dimensions are marked with a * sign and recorded in technical specifications (figure 5);
- dimensions that define zones of different roughness of the same surface, heat treatment zone, inoxidizing coatings, finishing, knurling, etc.;
- the dimensions that are set with the fitting stock of one part to the other during processing or assembly. Such dimensions are also marked with a * sign, and there are corresponding inscriptions in the technical requirements.

In the drawing repeated deviations of linear dimensional tolerances according to *IT12* and coarser are not indicated after the nominal dimensions, but they are specified in the common record in technical specifications, if this record uniquely determines the value and direction of the limiting deviations. For example: Unspecified dimensional deviations $H14, h14, \pm IT14/2$. If for the constructive, technological and other reasons, any deviation is used for any dimensions that differ from those specified in the common record, then these deviations are placed next to the nominal dimensions.

Fits and limits of slick cylindrical joints

The system of fits and limits is the set of the series of fits and limits that are naturally constructed on the basis of experience, theoretical and experimental studies, and designed in the form of a standard. The system is designed to choose the necessary and practically sufficient options for fits and limits of typical joints of machine parts. The system allows standardization of cutting tools and calibers simplifies the design and achievement of joint interchangeability, improves the quality of products. The systems of fits and limits are built on the same principles for all types of joints.

The unified system of tolerances and fits (USTF) of slick cylindrical joints is set forth in the following standards: GOST 25346-82, GOST 25347-82, GOST 25348-82, GOST 25349-82 Fit system of the basis hole (**hole system** – HS) is the set of fit in which, for the same degree of accuracy and the same nominal size, the limiting deviations of the holes are the same for any fits, and various fits are achieved by varying the limiting deviations of the shafts (Figure 1, *a*).

Fit system of the basis shaft (**shaft system** – SS) is the set of fit in which, for the same degree of accuracy and the same nominal size, the limiting deviations of the shafts are the same for any fits, and various fits are achieved by varying the limiting deviations of the holes (Figure 1, *b*).

For all plantings in the hole system, the lower hole deviation is zero, i.e. the lower boundary of the tolerance field of the hole, called the main hole, always coincides with the zero line. For all fits in the hole system, the lower hole deviation is zero, $es = 0,$ i.e. the lower boundary of the tolerance limit of the hole, called the basis hole, always coincides with the zero line.

The tolerance limit of the basis hole is laid up, and the basis shaft is laid down from the zero line, i.e. in the part material. Such a system is called a unilateral system of tolerances. The nature of the same fit (i.e. the limiting values of clearance and interference) in the hole system and in the shaft system are approximately equal. The choice of this or that system is determined by constructive, technological and economic reasons.

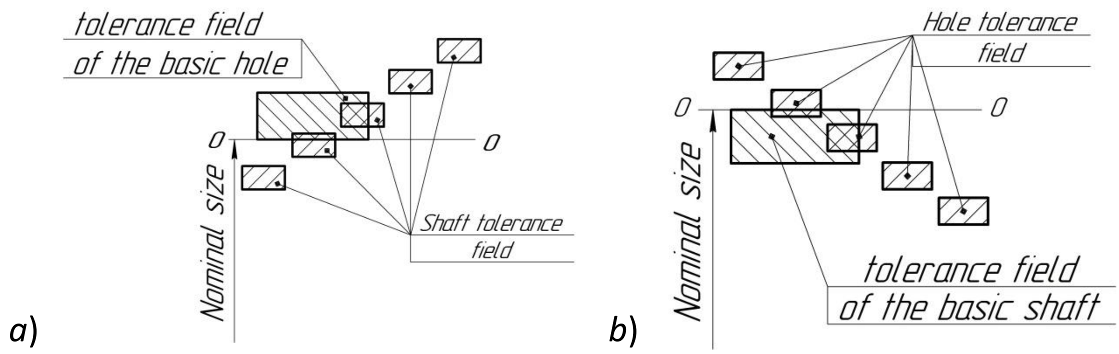


Figure 1 – Is a layout of tolerance limits

In the engineering industry, the hole system got widespread, i.e. the exact holes are treated with an expensive tool (countersinks, reamers, broaches). Each of them is used to process only one size with a specific tolerance limit. The shafts, regardless of their size, are treated with the same tool or grinding wheel. In the hole system there are less holes different in the limiting dimensions. This makes it possible to manufacture the cutting tool at specialized enterprises, with lower costs and high quality.

However, in some cases, for design reasons, it is necessary to use the shaft system when it is necessary to alternate the joints of several holes of the same nominal size, but with different fits on the same shaft.

For example, in the connection shown in Figure 2, *a*, the movable fit of the fork 1 must be provided with the rod 3 and its stationary fit with the fork 2. If this joint is made in the hole system (Figure 2, *b*), the roller will have to be made stepped, with the outermost steps having a larger diameter than the middle one. Installation of such a unit is difficult (the roller, passing by the thickened step through the hole in the fork, will damage the surface of the hole). Therefore, in this case it is advisable to choose the shaft system (Figure 2, *c*). The shaft system is also used when parts such as rods, axles, rollers are made from precise cold-drawn rods without machining their outer surfaces. When selecting a fit system, tolerance must also be made for standard products (for example, rolling bearings).

To construct the dimension-limit system, the tolerance unit *i (I)* is set, which reflects the influence on the accuracy of technological, constructive and metrological factors, expresses the tolerance dependence on the nominal size and acts as a measure of accuracy. Based on the research and systematization of the experience of machining cylindrical metal parts, a tolerance unit is established for sizes from 1 to 500 *mm* and is determined by formulas 7–9.

Grade of accuracy (degree of accuracy for threaded joints, gears, etc.). In each item, parts for various purposes are manufactured with varying accuracy. To qualify the required levels of accuracy, the grades (degrees of accuracy) of the parts and items manufacture are established.

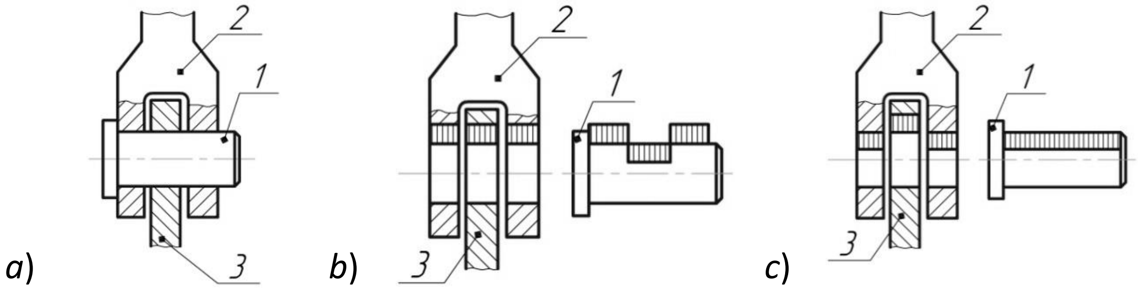


Figure 2 – Is a knuckle fork joint with a rod

Grade of accuracy is understood as a set of tolerances characterized by a constant relative accuracy, determined by the *a* coefficient for all nominal sizes of the given range (for example, from 1 to 500 *mm*). Accuracy within the same grade varies only depending on the nominal size. The number of grades is determined by the needs of various industries, the prospects for improving the accuracy of items, the boundaries of achievable accuracy, functional and technological factors and the accepted value of the denominator ϕ of the geometric progression over which the tolerance varies when moving from one grade to another. The grade determines the manufacturing tolerance allowance, and, consequently, the corresponding methods and means of processing the machine parts.

Numerical values of the upper and lower deviations of tolerance limits for sizes from 1 to 500 *mm* are presented in the tables of GOST 25347-82, which makes it possible to apply the system without resorting to formulas or rules by which they are determined. USTF is one-sided, limiting and allows application, both the hole system and the shaft system. The tolerance limit of the basis hole is denoted by the *H* Latin alphabet letter, and the limit of the basis shaft tolerance is given by the *h* letter.

In USTF for sizes from 1 to 500 *mm*, there are 20 grades:

- 01, 0 and 1 are for end-measuring rod;
- 2, 3 and 4 are for calibers and especially exact items;

- 5, 6...12 are for the formation of fits in engineering;
- 13, 14...19 are for non-coupling (free), size.

USTF determined 13 fundamental and 22 additional intervals of nominal sizes (for sizes from 1 to 500 mm). The fundamental intervals: 1–3, 3–6, 6–10, 10–19, etc. Additional (intermediate) intervals are established in the fundamental: for example, 10–14, 14–18 – in the fundamental interval 10–18.

Starting from the 5th grade, the tolerances are calculated by the formula (6). From the 6th grade, when passing to the next, coarser grade in all intervals of nominal sizes, every five grade tolerances increase by 10 times. This regularity allows calculating the tolerances for any grade.

Annex E provides the USTF tolerances for grades 01...10. Tolerances of grades 11...15 are calculated by the formula

$$IT(N+5) = 10\ ITN$$

Tolerances for grades 16 and more – by the formula

$$IT(N+10) = 100\ ITN$$

where *N* is the number of grades, which is 5 or 10 numbers less than the grade for which the tolerance is calculated.

For example, it is required to determine *IT12* for a size of Ø 20. In this case $N = 12 - 5 = 7$, consequently, for the 18th grade $IT18 = 100 IT8 = 100 \cdot 3300 = 3300$ μm.

For the formation of fits in USTF, for dimensions up to 500 mm, 28 fundamental deviations of the holes and shafts are provided.

The fundamental deviation is one of two deviations (upper or lower) used to determine the position of the tolerance limit relative to the zero line. In USTF, such a deviation is the deviation closest to the zero line. The fundamental deviations of the holes are denoted by uppercase letters of the Latin alphabet, and the shafts by lowercase letters (Figure 3).

Deviations *A-H (a-h)* are intended for formation of tolerance limits in clearance fit; deviations of *Js-N (js-n)* – in transitional fit; deviations of *P-ZC (p-zc)* – in interference fit. For each letter designation, the absolute value and sign of the shaft fundamental deviation (upper *es* for shafts from *a* to *h* or lower *ei* for shafts from *p* to *zc*) are determined by the special formulas given in GOST 25346-82. The value of the fundamental deviation does not depend on the grade, except for the shafts in which there is no fundamental deviation.

The fundamental deviations (Figure 3) and tolerance limits (tables 1 and 2). The values of the fundamental deviations of the shafts are calculated by the empirical formulas given in GOST 25346-82, and the fundamental deviations of the holes are determined by the general and special rules (Figure 4).

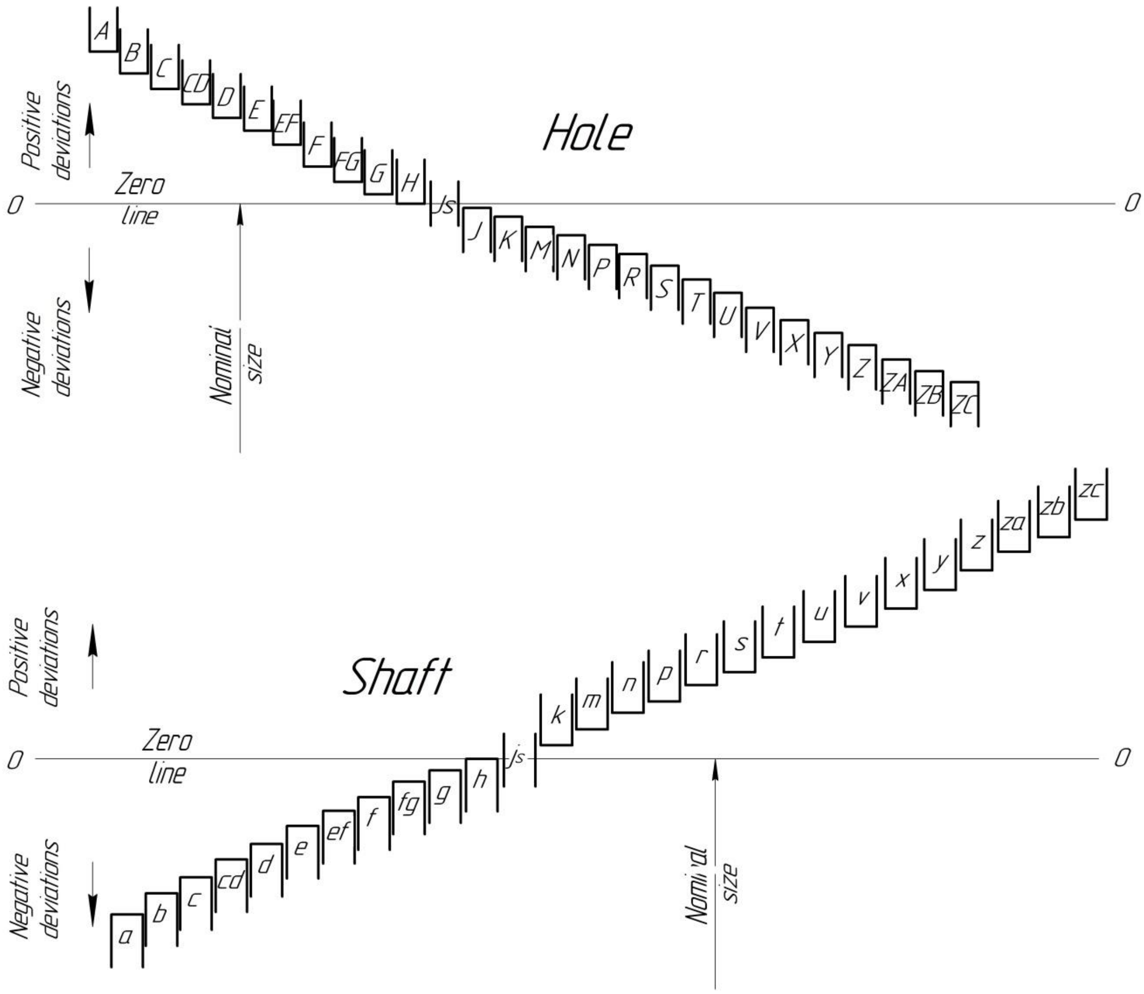


Figure 3 – Is the fundamental deviations of the holes and shafts adopted in the ISO and USTF system

General rule:

- for the holes $A \ldots H$

$$EI = -es \tag{1}$$

- for the holes K…ZC

$$ES = -ei \qquad (2)$$

From the general rule, for sizes over 3 to 500 mm, exceptions are made: the fundamental deviation of hole N is zero, beginning with the 9th grade; for holes J ... N up to the 8th grade and for holes P...ZC up to the 7th grade, the fundamental deviations are determined by a special rule:

$$ES = -ei + \Delta \qquad (3)$$

where $\Delta = IT_n - IT_{n-1}$ is the difference between the tolerance of the grade in question and the tolerance of the nearest more precise grade.

Usually, for transitional fit and interference one, a more accurate tolerance (one grade) is assigned to the shaft than to the hole, i.e. $IT_n = TD$ and $IT_{n-1} = Td$.

The tolerance limits in the USTF are formed by a combination of one of the fundamental deviations with a tolerance for one of the grades. In accordance with this rule, the tolerance limit is denoted by the letter of the fundamental deviation and the grade number: for example, for the shaft *f6, n7*; for the hole – *F7*; *N6*.

In accordance with the ISO recommendation and the practice of engineering, in the standard of all the basic series of tolerance limits for sizes from 1 to 500 mm, the preferred tolerance limits are highlighted. They provide 90–95 % of fits of general use. The use of preferred tolerance limits helps to increase the level of item unification, reduces the range of dimensional cutting tools and calibers, and creates favorable conditions for cooperating and organizing the centralized production of standard cutting tools and gauges at specialized enterprises.

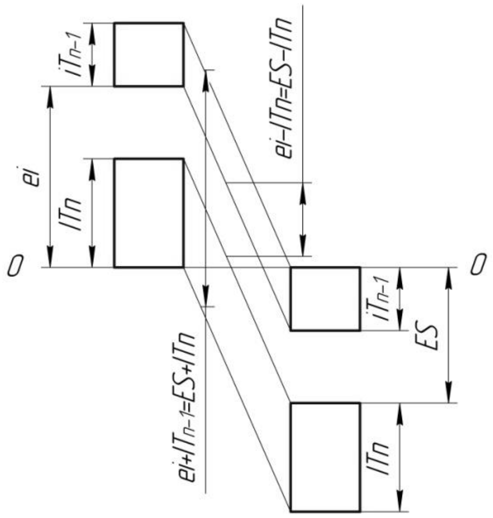


Figure 4 – Is an illustration to the special rule for determining the fundamental deviations of holes

The positive side of the USTF is the establishment in all ranges of sizes of recommended fits with allocation from them for sizes from 1 to 500 mm, preferred fits of priority use. Unification of fits makes work easier for designers when determining fits. At the same time, combining the preferred tolerances of shafts and holes in different options, the system's capabilities to create various fits (in the hole and shaft system) are significantly extended without increasing the set of tools, calibers and other tooling.

Further reduction of the tolerances and fits for one industry can be achieved by introducing a restrictive standard (industry standard or enterprise standard). For economic reasons, in practice, the hole system has a greater application, therefore, in the hole system there are more preferred fits than in the shaft system. GOST 25347-82 in technically justified cases allows the use of fits that differ from those recommended, but formed from the number of tolerance limits of shafts and holes recommended by this standard. However, in these cases, the fit should also refer to the hole or shaft system.

Fits established in the USTF, for sizes 1...3150 mm are indicated in tables 1...5. In GOST 25347-82 there are recommendations for the application of tolerance limits and fit formations for sizes less than 1 mm.

The tolerance limits for plastic parts are given in tables 7; 9. Recommended fits in the hole and shaft systems for connecting plastic parts with metal or plastic parts are given in tables 8; 10.

Currently, three methods of tolerance and fit are used.

1. The method of precedents (the method of analogues) is that the designer searches in similar or other machines, previously designed and in operating, cases of using an assembly unit similar to the one being designed, and assigns the same or similar tolerance and fit.

2. The similarity method is essentially the development of the precedent method. It arose as a result of the classification of machine parts for structural and operational characteristics and the production of reference books with examples of fits. To select the tolerances and fits by this method, an analogy is established between the design features and operating conditions of the designed assembly unit with the characteristics indicated in the reference books. However, in these materials, constructive and operational indicators are often classified by common expressions that do not reflect the quantitative values of the parameters, which make it difficult to choose fits. A common drawback of precedent and similarity methods is the difficulty in determining the features of uniformity and similarity, the possibility of applying erroneous tolerances and fits.

3. The computational method is the most justified method for selecting tolerances and fits. Choosing this method, the grades (degrees of accuracy), tolerances and fits in the design of machines and other items, seek to meet the operational and design requirements for the part, assembly unit and the item as a whole.

When choosing the grade, it is necessary to take into account the technology of parts productio, since certain methods of processing have certain economically justified accuracy (Annex F).

Selection of tolerances (grades) for coupling size and fits. When choosing fits should be guided by the recommendations.

Clearance fit. The H/h fit with zero minimum clearance is set in the grades 4...12. They are characterized by ease of assembly, disassembly, sufficiently high accuracy of centering and moving parts, allow for repeated disassembly and assembly. They are mainly used for static connections, often disassembled or adjustable in the mechanism settings, and the joints are reinforced with accessory mounting.

In dynamic fit joints, *H/h* allows slow axial or lateral movements and even rotational motion with low frequency at light loads during the operation of the mechanisms. They can be used instead of transitional fits.

The *H7/h6 fit* is widely used, for example, for the installation of change gear wheels, friction couplings, caged and housing bearing cups.

The *H8/h7 fit* compared with the *H7/h6* fit facilitates the manufacture and assembly of parts; they are used with low requirements for the centering accuracy and a long joint length.

The *H8/h8, H8/h9, H9/h8, H9/h9* fits are used in various joints with low requirements for centering and alignment accuracy, to facilitate assembly and disassembly; for example, for mounting pulleys, couplings, gears and other parts operating at light and constant loads and rare axial movements on the shafts, for instalation rolling bearing housings and detachable bearing shells in the housing, for fitting easily adjustable parts; for centering parts.

The *H10/h10* fit is usually used instead of the *H9/h9* fit, if the working conditions allow some reduction in accuracy.

The *H11/h11, H12/h12* fits are used in joints of low accuracy: centering of flange covers, connection of pulling chain sprockets with shafts of crawler gears and distance pieces, forks with low accuracy; in connections with structural and technological clearances.

The H/*g, G/h* fits with the minimum backlash clearance provide accurate fixation, smoothness and running accuracy, tightness of joints. They are used in particularly precise movable and reversible joints, in sliding bearings of particularly precise mechanisms at light loads and insignificant heating.

For example, the *H7/g6, G7/h6* fits are used to install sliding gear box in the precise prismatic pair.

The H/f, F/h fits with moderate backlash clearances. Clearances provide semi-liquid or liquid lubrication of the plain bearings for light and medium operation. They are used with relatively low requirements for the running and centering accuracy. They facilitate assembly and disassembly.

The *H7/f7, F8/h6* fits provide good lubrication conditions, they are used for bearings of the light and medium-sized machines and connections with translational movements, for example, gearbox bearings, bushings of freely rotating gears and pulleys, joints of movable coupling halves with shafts.

The *H8/f8*, *P8/h8*, *H8/f9*, *H9/f9*, *F9/h8*, *F8/h9* fits are used, for example, for sliding bearings of two-bearing shafts operating at a considerable speed of rotation, as well as for shafts with widely spaced supports, for large, heavily loaded machines; for long sliding bearings; for supports of freely rotating gears and other parts with low accuracy of centering.

The *H/e, E/h* fits with maximum backlash clearances have clearances about 2 times more than those of the *H/f* fits. They provide free rotation under increased operating conditions (significant load, high rotation speed). These fits are used for sliding bearings of spaced supports, multi-bearing shafts or long shafts, for fixed, controlled joints of low accuracy with large clearances.

The *H6/e7, H7/e7*, *E8/h6* fits are are intended mainly for bearings with liquid lubrication, particular accuracy and durability.

The *H7/e8*, *H8/e8*, *E9/h8*, *E8/h8*; *E8/h7* are the fits of particular and normal grade of accuracy. Often, they are used in bearings with liquid lubrication, operating at high speeds and low pressures, which are sizable (long) or widely spaced.

The *H8/e9*, *H9/e9*, *E9/h9* (reduced accuracy fits) are used in less demanding sliding bearings in static connections, if needed to increase the backlash clearance.

The H/d, *D/h* fits with maximum backlash clearances compensate for significant deviations in the location of the joint surfaces and temperature deformation, ensure free movement, adjustment and assembly of parts.

The *H7/d8*, *H8/d8*, *D8/h6*, *D8/h7* fits are intended mainly for precise dynamic joints operating under heavy duty conditions and high heating, for example, turbine plain bearings; support of high-speed idler pulleys and gears, bearings of widely spaced supports.

The *H8/d9, H9/d9, H8/d10, H9/dl0, D9/h8, D9/h9, D10/h9* fits are designed for joints operating with low accuracy requirements. Preference should be given to fit with the *d9* shaft tolerance limit.

H11/d11, *D11/h11* are intended for joints of low running and centering accuracy. The backlash clearance allows compensating for the disalignment of joint faces, dimensional change from the application of protective coatings and ensures the mobility of the mating parts in dusty or contaminated conditions. They are used for unfinished linear motion guides, tie rods and and levers with low running accuracy; joints of bearing caps with housings; installation of spacer sleeves; shaft bearings, as well as gears, pulleys, couplings, freely sitting on the shafts of rough, slow-moving mechanisms.

The *H/a*, *H/b*, *H/c*, *A/h*, *B/h*, *C/h* maximum clearance fits are used mainly in rough grades (11 and 12) for designs of low accuracy: for the high clearance it is necessary for compensation of significant disalignment of the mating surfaces or in terms of operation, for significant dimensional changes of the parts; for joints operating under dust and contamination.

The *H11/c11*, *H11/b11*, *C11/h11*, *B11/H11*, *H11/a11*, *A11/H11* fits are used for flange connections, for solder joints, hinges of low accuracy.

The *H12/a12*, *A12/h12* fits are used for hinges of low accuracy. The fits of this group, if possible, should be replaced by the *H11/d11* fits.

Transitional fits. *The H/j_s, J_s/h* fit with low possible interference. Interference does not exceed half the shaft tolerance limits. Assembly and disassembly requires little effort. These fits are used for static connections, if low clearance is permissible during the centering of parts, repeated frequent assemblages and disassemblies are possible. They are also used for dynamic joints with close clearance. In these cases, the interference is eliminated by a selective assembly. These fits can be replaced by fits of the next group in joints with a relatively large length (3...4 diameters) or with difficult assembly and disassembly.

The $H6/j_s5$, $J_s6/h5$ fits are usually used in bearing unit of high accuracy.

The $H7/j_s6$, $J_s7/h6$ fits are used to install rolling bearings on shafts, sleeves for rolling bearings, as well as thin-walled bushes in the housing, gears, couplings, small pulleys on shafts of small electric machines.

$H8/j_s7$, $J_s8/h7$ are the fits of reduced accuracy, for example, for centering the body parts.

The H/k, K/h fits with equally possible clearance and interference provide good centering, assembly and disassembly with little effort.

H7/k6, *K7/h6* are the fits commonly used in instrument and mechanic engineering; they are used fir connections of gears, pulleys, flywheels, replaceable couplings with shafts, for the installation of cups of bearing assemblies and bushings into the hubs of freely rotating gears.

The *H8/k7, K8/h7* fits are used for the with reduced requirements for the accuracy of connections.

The H/m, M/h fits with high possible clearance. As a rule, clearance is not obvious due to form deviations. They are used for static, rarely disassembled connections with accessory mounting. At light loads and long connection lengths, connections can not be further strengthened.. These fits replace the fit of the next group with a joint length more than (1.5...2) D or when large deformations of the parts are unacceptable.

The *H7/m6, M7/h6* fits are used to fix gears on shafts, to install pins in housings and other parts; to instal static thin-walled bushings or steel bushings in non-ferrous alloy housing.

The H/n, N/h fits with low possible clearance are used only in static, rarely disassembled connections. Assembly and disassembly take place under the press. These fits provide good centering and transmit heavy loads. For light loads of constant magnitude, the joints are secured without accessory mounting.

The *H7/n6, N7/h6* fits are widely used to connect heavily loaded gears, couplings, toggles with shafts; bronze crowns with centers of worm wheels and worm wheels with shafts, bushings of sliding bearings with housings. These fits can be replaced by the *H8/n7, N8/h* fits, if the fit tolerance can vary widely.

Interference fits. *The H/p, P/h* fits with minimum backlash interference are used at light loads; in cases where random drift of connected parts are permissible; for connecting easily deformable thin-walled parts; for centering heavily loaded or fast-rotating large parts with accessory mounting; for connecting parts from non-ferrous metals and light alloys.

The *H6/p5, P6/h5* fits are used, if significant variations in interference are not permissible.

The *H7/p6, P7/h6* fits are used for gears installation on the shafts (connections are reinforced with keys); for the bushes pressing into the housings and the adjusting rings pressing on the shafts.

The H/r, H/s, H/t, R/h, S/h, T/h fit with moderated backlash interference transmit medium loads without additional fastening and heavy loads with accessory mounting.

H7/r6, H7/s6, H8/s7, H7/t6, R7/h6, S7/h6, T7/h6 are the mean accuracy fits [interferences are ≈ (0,0002...0,0006)*D*]. They are used for pressing the bushings of the plain bearings into the housing and gears for heavy and shock loads; for pressing the gears and worm wheels onto shafts under severe operating conditions, as well as bronze gear crowns on cast iron centers.

With accessory mounting, they can replace the fit of the next group.

The H/u, H/x, H/z, U/h fits with maximum backlash interference. Interferenses are (0,001...0,002)*D*. They perceive heavy and shock loads, as a rule, without accessory mounting. To increase the joints strength, a selective assembly is used or a more accurate tolerance is assigned to the basic part.

H7/u7, H8/u8, U8/h7 are the most applicable fits of this group. They are used for pressing the static half couplings onto the shafts ends, as well as bronze gears and steel bandages on cast iron and steel centers; for pressing short hubs into the hubs of freely rotating gears.

The *H8/x8, H8/z8* fits are used in connections subject to variable loads, shocks and vibrations, as well as for connecting thick-walled and solid sections in the parts material of which high power can occur.

Table 1 – Is tolerance limits of universal shaft with nominal sizes from 1 to 500 mm (according to GOST 25347 - 82)

Grades of accuracy	Fundamental deviations																							
	a	b	c	cd	d	e	ef	f	fg	g	h	j_s	j	k	m	n	p	r	s	t	u	v	x	z
01											h01*	j_s01*												
0											h0*	j_s0*												
1											h1*	j_s1*												
2											h2*	j_s2*												
3											h3*	j_s3*												
4								f4	fg4	g4	h4	j_s4		k4	m4	n4	p4							
5						e5	ef5	f5	fg5	g5	h5	j_s5	j5	k5	m5	n5	p5	r5	s5	t5	u5			
6					d6	e6	ef6	f6	fg6	**g6**	**h6**	**j_s6**	j6	**k6**	m6	**n6**	**p6**	**r6**	**s6**	t6	u6	v6		
7					d7	e7	ef7	**f7**		g7	**h7**	j_s7	j7	k7	m7	n7	p7	r7	s7	t7	u7	v7	x7	z7
8			c8		d8	**e8**	ef8	f8			**h8**	j_s8*							s8		u8		x8	z8
9	a9	b9	c9	cd9	**d9**	e9		f9			**h9**	j_s9*												
10					d10						h10	j_s10*												
11	a11	b11	c11		**d11**						h11	j_s11*												
12		b12									h12	j_s12*												
13											h13*	j_s13*												
14											h14*	j_s14*												
15											h15*	j_s15*												
16											h16*	j_s16*												
17											h17*	j_s17*												

Notes
1. Within the table, the preferred tolerance limits are indicated, and additional tolerance limits are underlined.
2. * Tolerance limits, as a rule, are not intended for fitting.
3. In addition to those listed in the table, the za8, zb8 and zc8 tolerance limits are set.

Table 2 – Is tolerance limits for the universal hole with nominal sizes from 1 to 500 mm (according to GOST 25347-82)

Grades of accuracy	Fundamental deviations																				
	A	B	C	CD	D	E	EF	F	FG	G	H	J_s	J	K	M	N	P	R	S	T	U
01											H01*	J_S01*									
0											H0*	J_S0*									
1											H1*	J_S1*									
2											H2*	J_S2*									
3											H3*	J_S3*									
4											H4	J_S4									
5						E5	EF5	F5	FG5	G5	H5	J_S5		K5	M5	N5	P5				
6					D6	E6	EF6	F6	FG6	G6	H6	J_S6	J6	K6	M6	N6	P6	R6	S6	T6	
7					D7	E7	EF7	F7		G7	$\boxed{H7}$	$\boxed{J_S7}$	J7	$\boxed{K7}$	M7	$\boxed{N7}$	$\boxed{P7}$	R7	7S	T7	U7
8			C8		D8	E8	EF8	$\boxed{F8}$			$\boxed{H8}$	J_S8*	J8	K8	M8	N8	P8	R8			U8
9	A9	B9	C9	CD9	D9	$\boxed{E9}$		F9			$\boxed{H9}$	J_S9*				N9	P9				
10					D10	E10					H10	J_S10*									
11	A11	B11	C11		D11						$\boxed{H11}$	J_S11*									
12		B12									H12	J_S12*									
13											H13*	J_S13*									
14											H14*	J_S14*									
15											H15*	J_S15*									
16											H16*	J_S16*									
17											H17*	J_S17*									

Note: see notes 1 and 2 to Table 1. In addition to the above, the Z8 tolerance limit is set in the table.

Table 3 – Is recommended fits in the hole system at nominal sizes from 1 to 500 mm (according to GOST 25347-82)

Basic hole	Fundamental shaft deviations																		
	a	b	c	d	e	f	g	h	j_s	k	m	n	p	r	s	t	u	x	z
	Fits																		
H5							$\frac{H5}{g4}$	$\frac{H5}{h4}$	$\frac{H5}{J_S 4}$	$\frac{H5}{k4}$	$\frac{H5}{m4}$	$\frac{H5}{n4}$							
H6						$\frac{H6}{f6}$	$\frac{H6}{g5}$	$\frac{H6}{h5}$	$\frac{H6}{j_S 5}$	$\frac{H6}{k5}$	$\frac{H6}{m5}$	$\frac{H6}{n5}$	$\frac{H6}{p5}$	$\frac{H6}{r5}$	$\frac{H6}{s5}$				
H7			$\frac{H7}{c8}$	$\frac{H7}{d8}$	$\frac{H7}{e7}$ $\frac{H7}{e8}$	$\frac{H7}{f7}$	$\frac{H7}{g6}$	$\frac{H7}{h6}$	$\frac{H7}{j_S 6}$	$\frac{H7}{k6}$	$\frac{H7}{m6}$	$\frac{H7}{n6}$	$\frac{H7}{p6}$	$\frac{H7}{r6}$	$\frac{H7}{s6}$	$\frac{H7}{t6}$	$\frac{H7}{u7}$		
H8			$\frac{H8}{c8}$	$\frac{H8}{d8}$	$\frac{H8}{e8}$	$\frac{H8}{f7}$ $\frac{H8}{f8}$		$\frac{H8}{h7}$	$\frac{H8}{Js7}$	$\frac{H8}{k7}$	$\frac{H8}{m7}$	$\frac{H8}{n7}$			$\frac{H8}{S7}$		$\frac{H8}{u8}$	$\frac{H8}{x8}$	$\frac{H8}{z8}$
				$\frac{H9}{d9}$	$\frac{H9}{e9}$	$\frac{H8}{f9}$		$\frac{H8}{h9}$											
H9				$\frac{H9}{d9}$	$\frac{H9}{e8}$ $\frac{H9}{e9}$	$\frac{H9}{f8}$ $\frac{H9}{f9}$		$\frac{H9}{h8}$											
H10				$\frac{H10}{d10}$				$\frac{H10}{h9}$											
H11	$\frac{H11}{a11}$	$\frac{H11}{b11}$	$\frac{H11}{c11}$	$\frac{H11}{d11}$				$\frac{H11}{h11}$											
H12		$\frac{H12}{b12}$						$\frac{H12}{h12}$											

Note: there are the preferred fits in the table.

Table 4 – Is recommended fits in the shaft system at nominal sizes from 1 to 500 mm (according to GOST 25347-82)

Basic shaft	Fundamental hole deviations																
	A	B	C	D	E	F	G	H	J_s	K	M	N	P	R	S	T	U
	Посадки																
h4							$\frac{G5}{h4}$	$\frac{H5}{h4}$	$\frac{J_S5}{h4}$	$\frac{K5}{h4}$	$\frac{M5}{h4}$	$\frac{N5}{h4}$					
h5						$\frac{F7}{h5}$	$\frac{G6}{h5}$	$\frac{H6}{h5}$	$\frac{J_S6}{h5}$	$\frac{K6}{h5}$	$\frac{M6}{h5}$	$\frac{N6}{h5}$	$\frac{P6}{h5}$				
h6				$\frac{D8}{h6}$	$\frac{E8}{h6}$	$\frac{F7}{h6};\frac{F8}{h6}$	$\frac{G7}{h6}$	$\frac{H7}{h6}$	$\frac{J_s7}{h6}$	$\frac{K7}{h6}$	$\frac{M7}{h6}$	$\frac{N7}{h6}$	$\frac{P7}{h6}$	$\frac{R7}{h6}$	$\frac{S7}{h6}$	$\frac{T7}{h6}$	
h7				$\frac{D8}{h7}$	$\frac{E8}{h7}$	$\frac{F8}{h7}$		$\frac{H8}{h7}$		$\frac{J_s8}{h7}$	$\frac{K8}{h7}$	$\frac{M8}{h7}$	$\frac{N8}{h7}$				$\frac{U8}{h7}$
h8				$\frac{D8}{h8};\frac{D9}{h8}$	$\frac{E8}{h8};\frac{E9}{h8}$	$\frac{F8}{h8};\frac{F9}{h8}$		$\frac{H8}{h8}\frac{H9}{h8}$									
h9				$\frac{D9}{h9};\frac{D10}{h9}$	$\frac{E9}{h9}$	$\frac{F9}{h9}$		$\frac{H8}{h9};\frac{H9}{h9}$									
h10				$\frac{D10}{h10}$				$\frac{H10}{h10}$									
h11	$\frac{A11}{h11}$	$\frac{B11}{h11}$	$\frac{C11}{h11}$	$\frac{D11}{h11}$				$\frac{H11}{h11}$									
h12		$\frac{B12}{h12}$						$\frac{H12}{h12}$									
Note: there are the preferred fits in the table.																	

Table 5 – Is tolerance limits and recommended fits in the hole system and shaft system at nominal dimensions from 500 to 3150mm (GOST 25347-82)

Basic hole	Fundamental shaft deviations																
	c	cd	d	e	f	g	h	j_S	k	m	n	p	r	s	t	u	v
	Fits																
H6						$(\frac{G6}{h6})$	$(\frac{H6}{h6})$	$(\frac{J_S6}{h6})$	$(\frac{K6}{h6})$	$(\frac{M6}{h6})$	$(\frac{N6}{h6})$						
H7				$\frac{H7}{e7}$ $(\frac{E7}{h6})$	$\frac{H7}{f7}$ $\frac{H7}{f6}$	$\frac{H7}{g7}$ $\frac{H7}{g6}$	$\frac{H7}{h7}$ $\frac{H7}{h6}$	$\frac{H7}{j_S7}$ $\frac{H7}{j_S6}$	$\frac{H7}{k7}$ $\frac{H7}{k6}$	$\frac{H7}{m6}$ $(\frac{M7}{h7})$	$\frac{H7}{n7}$ $\frac{H7}{n6}$	$\frac{H7}{p7}$ $\frac{H7}{p6}$	$\frac{H7}{r7}$ $\frac{H7}{r6}$	$\frac{H7}{s7}$ $\frac{H7}{s6}$	$\frac{H7}{t7}$ $\frac{H7}{t6}$	$\frac{H7}{u7}$ $\frac{H7}{u6}$	$\frac{H7}{v7}$
H8			$\frac{H8}{d8}$ $(\frac{H8}{h7})$	$\frac{H8}{e7}$ $\frac{H8}{e8}$	$\frac{H8}{f8}$ $\frac{H8}{f7}$	$\frac{H8}{g7}$	$\frac{H8}{h8}$ $\frac{H8}{h7}$	$\frac{H8}{j_S7}$	$\frac{H8}{k7}$		$\frac{H8}{n7}$	$\frac{H8}{p7}$	$\frac{H8}{r7}$	$\frac{H8}{s7}$	$\frac{H8}{t7}$ $\frac{H8}{t8}$	$\frac{H8}{u8}$ $\frac{H8}{u7}$	$\frac{H8}{v8}$ $\frac{H8}{v7}$
H9			$\frac{H9}{d8}$ $\frac{H9}{d9}$	$\frac{H9}{e8}$ $\frac{H9}{e9}$	$\frac{H9}{f8}$ $\frac{H9}{f9}$		$\frac{H9}{h8}$ $\frac{H9}{h9}$								$\frac{H9}{t8}$	$\frac{H9}{u8}$	$\frac{H9}{v8}$
H10			$\frac{H10}{d10}$				$\frac{H10}{h10}$										
H11	$\frac{H11}{c11}$	$\frac{H11}{cd11}$	$\frac{H11}{d11}$				$\frac{H11}{h11}$										
H12							$\frac{H12}{h12}$										

Table 6 – Is variants of a combination in one common record of unspecified limit deviations of linear dimensions, except for the radius of curve and bevel radius

Option	Dimensions of shafts		Dimensions of holes		Dimensions of elements not related to holes and shafts
	round (diameters)	other	round (diameters)	other	
	Limit deviations for one common record				
1	–IT (h)		+IT(H)		±t/2
2	–t		+ t		±t/2
3			±t/2		
4	–IT (h)	±t/2	+IT(H)	±42	±t/2

N o t e s

1. Agreed notations: – *IT, + IT* are one-sided limit deviations directed respectively to minus and plus from the nominal size, which are assigned according to the grades of accuracy; correspond to the shaft *h* or hole *H*; – t, +t are one-sided limit deviations directed respectively to minus and plus from the nominal size, which are assigned according to the accuracy class; ± *t/2* are symmetrical limit deviations according to the accuracy class.
2. In the common record, which only applies to dimensions less than 1 mm, it is possible to clerify unspecified symmetrical limit deviations in the grades.
3. Option 1 is preferred; option 2 is not recommended.

Table 7 – Is tolerance limits of shafts for parts made of plastics (in accordance with GOST 25349-82)

Grade	Fundamental deviation														
	a	b	c	d	e	f	h	j_s	k	u	x	z	za	zb	zc
8			c8	d8	e8	f8	h8	j_s8	k8	u8	x8	z8			
9				d9	e9	f9	h9	j_s9	k9						
10				d10			h10	j_s10	k10		x10	z10	za10	zb10	
11	a11	b11	c11	d11			h11	j_s11	k11						zc11
12		b12					h12								
13							h13	j_s12							

Table 8 – Is recommended fit for connecting plastic parts in the hole system (according to GOST 25349-82)

Basic hole	Fundamental shaft deviations																	
	ay	az	a	b	c	d	e	f	h	k	u	x	y	z	za	zb	zc	ze
H8					$\frac{H8}{c8}$	$\frac{H8}{d8}$	$\frac{H8}{e8}$	$\frac{H8}{f8}$	$\frac{H8}{h8}$	$\frac{H8}{k8}$	$\frac{H8}{u8}$	$\frac{H8}{x8}$		$\frac{H8}{z8}$				
H9						$\frac{H9}{d9}$	$\frac{H9}{e9}$	$\frac{H9}{f9}$	$\frac{H8}{h9}$	$\frac{H9}{k9}$		$\frac{H9}{x9}$	$\frac{H9}{y9}$	$\frac{H9}{z9}$	$\frac{H9}{za9}$	$\frac{H9}{zb9}$		
H10						$\frac{H10}{d10}$			$\frac{H10}{h10}$	$\frac{H10}{k10}$			$\frac{H10}{y10}$	$\frac{H10}{z10}$	$\frac{H10}{za10}$	$\frac{H10}{zb10}$	$\frac{H10}{zc10}$ $\frac{H10}{zc11}$	$\frac{H10}{ze11}$
H11	$\frac{H11}{ay11}$	$\frac{H11}{az11}$	$\frac{H11}{a11}$	$\frac{H11}{b11}$	$\frac{H11}{c11}$	$\frac{H11}{d11}$			$\frac{H11}{h11}$	$\frac{H11}{k11}$							$\frac{H11}{zc11}$	$\frac{H11}{ze11}$

Notes

1. In the grades 12 and 13, in addition to the above, the *H12/b12,* H12/h12, H13/h13 and B12/h12 are established.

2. The formation of fits not contained in the table is allowed. For example, to connect plastic parts to each other, you can apply the fits formed by the tolerance limits of the non-basic holes and shafts.

Table 9 – Is tolerance limits of holes for plastic parts (according to GOST 25349-82)

Grade	Fundamental deviations							
	A	*B*	*C*	*D*	*E*	*F*	*H*	J_s
8				*D*8	*E*8	*F*8	*H*8	J_s8*
9				*D*9	*E*9	*F*9	*H*9	J_s9*
10				*D*10			*H*10	J_s10*
11	*A*11	*B*11	*C*11	*D*11			*H*11	J_s11*
12							*H*12	J_s12*
13		*B*12					*H*13	J_s13*
Grade	Fundamental deviations							
	N	*U*	*X*	*Z*	*ZA*	*ZB*	*ZC*	
8	N8	U8						
9	N9**							
10	N10**		X10**	Z10**	ZA10**	ZB10**		
11	N11**						ZC11**	
12								
13								
Note: For footnotes and a note, see table 8.								

Table 10 – Is recommended fits for connecting plastic parts in the shaft system (according to GOST 25349-82) *

Basic shaft	Fundamental hole deviations									
	AY	AZ	A	B	C	D	E	F	H	N
*h*8						$\frac{D8}{h8}$	$\frac{E8}{h8}$	$\frac{F8}{h8}$	$\frac{H8}{h8}$	$\frac{N8}{h8}$
*h*9						$\frac{D9}{h9}$	$\frac{E9}{h9}$	$\frac{F9}{h9}$	$\frac{H9}{h9}$	$\frac{N9}{h9}$
*h*10						$\frac{D10}{h10}$			$\frac{H10}{h10}$	$\frac{N10}{h10}$
*h*11	$\frac{AY11}{h11}$	$\frac{AZ11}{h11}$	$\frac{A11}{h11}$	$\frac{B11}{h11}$	$\frac{C11}{h11}$	$\frac{D11}{h11}$			$\frac{H11}{h11}$	$\frac{N11}{h11}$
Basic shaft	Fundamental hole deviations									
	U	*X*	*Y*	*Z*	*ZA*	*ZB*	*ZC*		*ZE*	
*h*8	$\frac{U8}{h8}$									
*h*9		$\frac{X10}{h9}$	$\frac{Y10}{h9}$	$\frac{Z10}{h9}$	$\frac{ZA10}{h9}$	$\frac{ZB10}{h9}$				
*h*10			$\frac{Y10}{h10}$	$\frac{Z10}{h10}$	$\frac{ZA10}{h10}$	$\frac{ZB10}{h10}$	$\frac{ZC11}{h10}$	$\frac{ZC10}{h10}$;	$\frac{ZE11}{h10}$	
*h*11							$\frac{ZC11}{h11}$		$\frac{ZE11}{h11}$	
*For notes, see table 8										

Normalization of form deviation, surface disalignment and roughnesses

The accuracy of the geometrical parameters of the parts is characterized not only by the dimensional accuracy of its elements, but also by the form accuracy and surface-to-surface accuracy. Form deviations and surface disalignment occur during machining of parts due to inaccuracy and deformation of the machine, tool and device; deformation of the workpiece; unevenness of machining allowance; heterogeneity of the workpiece material and so on. In dynamic connections, these deviations lead to a decrease in the wear resistance of parts due to the increased specific pressure on the ridges of the irregularities, to the disruption of running smoothness, noise, etc.

In static connections, the form deviations and surface disalignment cause unevenness of the interference, resulting in a reduction in the strength of the joint, the tightness and the centering accuracy.

In assemblies, these deviations lead to errors of parts locating relative to each other, deformations, uneven clearances, which cause disruptions in the normal operation of individual assemblies and the mechanism as a whole; for example, rolling bearings are very sensitive to form deviations and relative positioning of the mounting surfaces.

Form deviations and surface disalignment reduce the process parameters of items. So, they significantly affect the accuracy and laboriousness of assembly and increase the volume of fitting operations, reduce the accuracy of measuring dimensions, affect the locating accuracy of the part in the manufacture and control.

When analyzing the accuracy of the geometric parameters of the parts, the following concepts are used.

Nominal surface is the perfect surface, the size and form of which correspond to specified nominal sizes and nominal form.

The actual surface is the surface that delimits the part and separates it from the environment.

Profile is the trace of the surface (there are concepts of actual and nominal profiles, analogous to the concepts of nominal and actual surfaces).

The normalized section L is the surface or line patch to which the form tolerance, position tolerance or the corresponding deviation are applied. If the normalized section is not specified, the tolerance or deviation refers to the entire surface under consideration or the length of the element in question. If the position of the normalized section is not specified, it can occupy any position within the entire element.

Adjoining surface is the surface that has nominal surface form, lies in contact with actual surface and is located outside the material of the part. So the deviation from it of the outermost point of the actual surface within the limits of the normalize section has a minimum value. Adjoining surface is used as a base for determining the form and position deviations.

Instead of an adjoining element for estimating form and position deviations, it is allowed to use as the locating element an average element that has a nominal form and is carried out by the least-squares method with respect to the actual one.

Position is the element of a part or a combination of elements with respect to which the position tolerance of the element in question is specified, and the corresponding deviations are determined.

Basic rules for indications of the tolerances of form and position on the drawings

The tolerances of form and position of surfaces are standardized by the following standards: basic terms and definitions by GOST 24642-81; numerical tolerance values by GOST 24643-81; indication of the tolerances of the form and position on the drawings by GOST 2.308-2011.

The limiting deviations of the form and position of surfaces are indicated on the drawings by the conventional signs in accordance with GOST 2.308-2011. Limiting deviations of the form and position of surfaces are normalized in accordance with GOST 24642-81 and GOST 24643-81 and are assigned if specific requirements arising from working, manufacturing or parts measurement conditions.

The type of tolerance is indicated on the drawing with the symbols given in Table 1. The tolerance data is indicated in a frame divided into two or more parts (Figure 1, *a*), in which:

- the first is the tolerance mark;
- the second is the tolerance value in millimeters;
- the third and subsequent are the letter sign of the *A* base or the surface with which the tolerance of the position is connected.

The frame is drawn with continuous fine lines, arranged horizontally or, as an exception, vertically and connected by a fine arrowhead line, with the element being normalized. The connecting line can be straight or jogged, but the direction of the segment with the arrow must correspond to the measurement direction (Figure 1, *a*, *c*). The connecting line can be drawn from the last part of the frame (Figure 1, *d*) and end with the arrow from the material side (Figure 1, *e*).

If the tolerance refers to the part surface or its profile (Figure 1, *a*) or to the thread flank (Figure 2, *b*), then the frame is connected to the contour line or its extension, the connecting line should not be an extension of the dimension line.

Table 1 – Is a conditional designation of tolerances

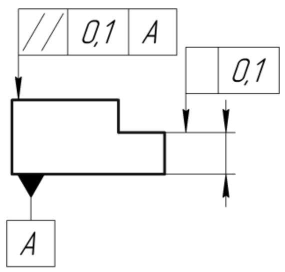
Tolerance group	Tolerance type	Sign

Form tolerance	Straightness tolerance, TFL	⏤
	Flatness tolerance, TFE	⏥
	Circularity tolerance, TFK	○
	Cylindricity tolerance, TFZ	⌭
	Tolerance for longitudinal section profile, TFP	=
Tolerance of position	Parallelity tolerance, TPA	∥
	Perpendicularity tolerance, TPR	⊥
	Angularity tolerance, TPN	∠
	Coaxiality tolerance, TPC	◎
	Symmetry tolerance, TPS	⌯
	Positional tolerance, TPP	⌖
	Intersection tolerance, TPX	×
Cumulative tolerance of form and position	Radial runout tolerance, TCR	↗
	Axial runout tolerance, TCA	
	Tolerance of runout in the intended direction, TCD	
	Total radial runout tolerance, TCTR	⌰
	Total axial runout tolerance, TCTA	
	Profile tolerance of any line, TCL	⌒
	Profile tolerance of any surface, TCE	⌓
Note 1 ∥⏥ is the sign of the cumulative parallelity and flatness tolerance; 2 ⊥⏥ is the sign of the cumulative perpendicularity and flatness tolerance; 3 ∠⏥ is the sign of the cumulative angularity and flatness tolerance.		

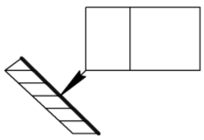
If the tolerance refers to the axis of the workpiece or the plane of symmetry (Figure 2, *c*, *d*) or to the thread axis (Figure 2, *a*), then the connecting line must be a continuation of the dimension line. With a lack of space, the arrow head and connecting lines can be combined (Figure 2, *e*).

If the tolerance refers to the axis (plane of symmetry) and from the drawing it is clear for which surfaces the axis (the plane of symmetry) is common, then the frame is connected to the axis (the plane of symmetry) (Figure 2, *f*).

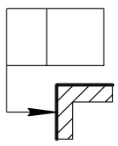
a)



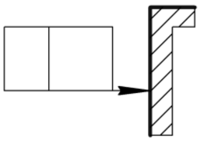
b)



c)



d)



e)

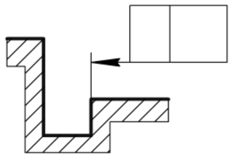
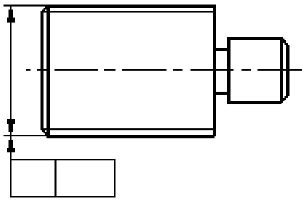
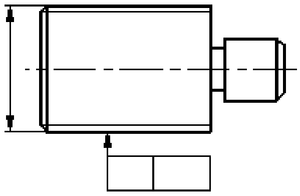


Figure 1 – Is a position of the connecting line with respect to the frame

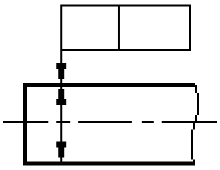
a)



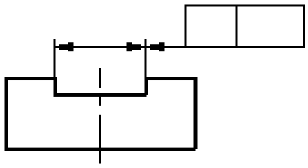
b)



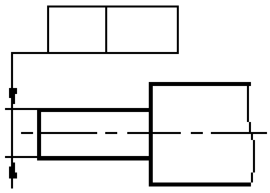
c)



d)



e)



f)

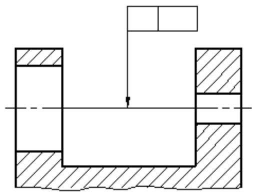


Figure 2 – Is examples of tolerance designation

Before the numerical value of the tolerance, there are:

- ∅ or *R* symbol, if the circularity or cylindricity tolerance is indicated by the diameter or radius, respectively (Figure 2, *a*, *b*);

- *T* symbol, if symmetry tolerance, intersection tolerance, form of profile or surface, and position tolerance (if its field is limited by two parallel straight lines or planes) are indicated in diametrical expression (Figure 2, *c*);

- *T/2* symbol, if the same tolerances are specified in the radius expression;

- the word «round» and the symbol ∅ or R, if the tolerance is spherical (Figure 2, *e*).

If the tolerance of the form and position of the surface refers to:

- the entire length of the profile or surface, then only the tolerance value is placed in the frame (Figure 4, *a*);

- any section of the profile (surface), then the length (Figure 4, *b*) or the size of the normalized section (Figure 4, *c*) is marked next to the numerical value of the tolerance;

- a certain area, the size and position of this section are indicated on the drawing (Figure 4, *d*).

Figure 3 – Is examples of filling the second frame graph

If it is necessary to designate a tolerance for the entire length of the surface and for a given length, the tolerance relating to the specified length is indicated under the tolerance relating to the entire length (Figure 4, *e*).

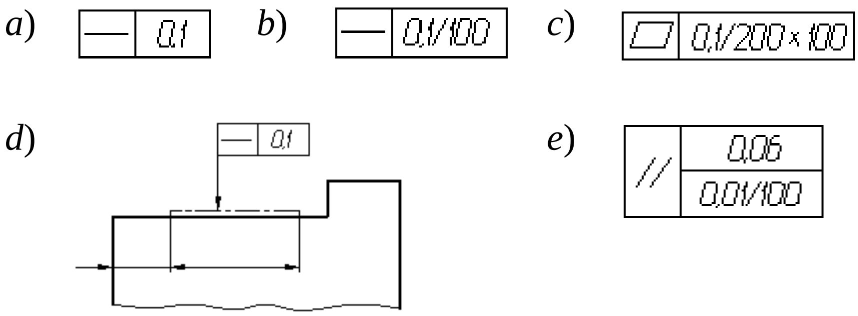


Figure 4 – Is examples of filling the second frame graph depending on the size of the surface covered by the tolerance

An example of the designation of the projecting position field is shown in Figure 5.

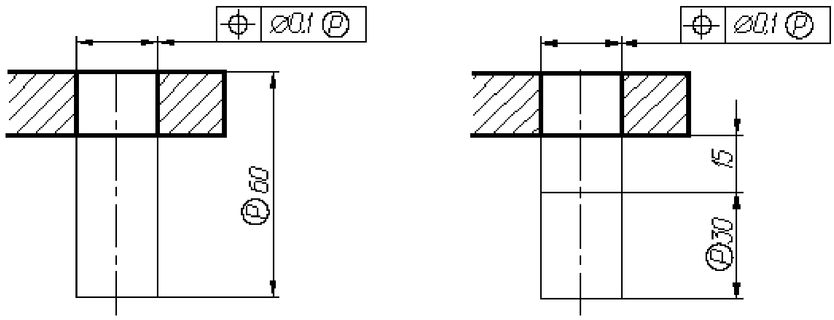


Figure 5 – Is the example of the designation of the projecting position field

Inscriptions supplementing the data shown in the frame are indicated next to the frame (Figure 6, *a*).

Repeated types of tolerances, denoted by the same sign, having the same values and referring to the same bases, are indicated once in the frame (Figure 6, *b*).

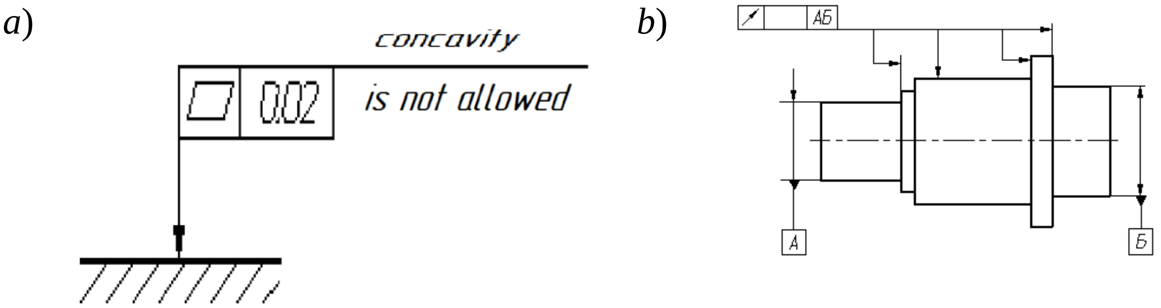


Figure 6 – Is the example of tolerances designation

The bases are denoted by a blackened equilateral triangle, which is connected by a connecting line to the frame (Figure 1, *a*, 6, *b*). If the base is a surface or its profile, then the base of the triangle is located on the contour line (Figure 1, *c*) or on its extension, shifting the triangle relative to the dimension line (Figure 7, *a*). If the base is an axis or a plane of symmetry, then the connecting line must be an extension of the dimension line (Figure 6, *b*). If the base is an axis or a plane of symmetry and it is clear from the drawing, for which surfaces the axis (plane of symmetry) is common, then the triangle is located on the axis (Figure 7, *b*). If the base is the axis of the center holes, then next to the designation of the base axis, the inscription «Axis of centers» is made (Figure 7, *c*). If the base is a certain part of the element or its place, then they are denoted by a dash-dot line and bounded by the dimensions (Figure 7, *d*).

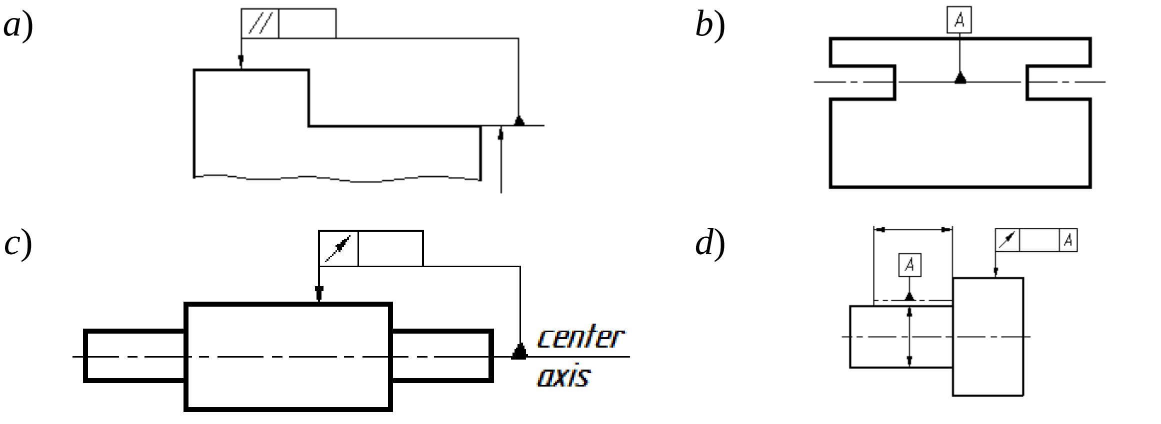


Figure 7 – Is examples of the base designation

If there is no need to select any of the surfaces as a base, then the triangle is replaced by an arrow (Figure 8, *e*).

If the connection of the frame to the base or other surface, to which the deviation of the position relates, is difficult, then the surface is denoted by a capital letter that is inscribed in the third part of the frame. The same letter is inscribed in the frame, which is connected to the base surface by a line ending with a triangle (Figure 1, *a*) or an arrow if the indicated surface is not a base (Figure 8, *b*).

If several elements form an integrated base and their sequence does not matter, then each element is designated independently and all letters are entered consecutively into the third part of the frame (Figure 9, *b*).

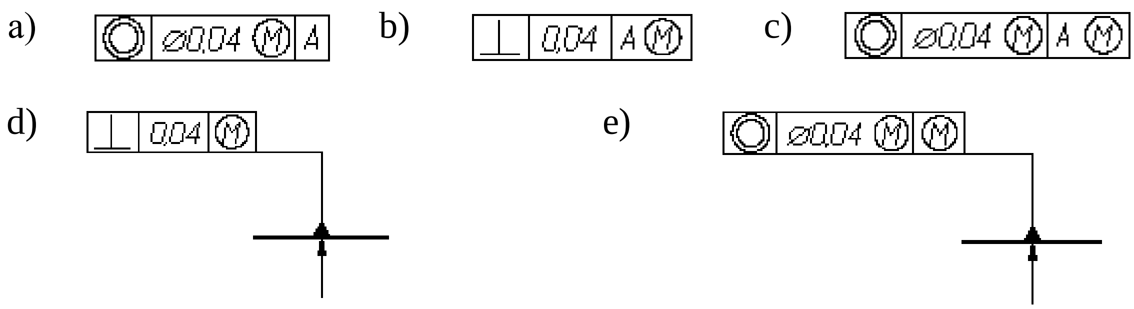


Figure 8 – Is examples of designating qualified tolerances

The linear and angular dimensions defining the nominal position and (or) the nominal form of the elements limited by the tolerance, with the assignment of the positional tolerance, the angularity tolerance, the profile tolerance of specified surface or the specified profile, indicate in the drawings without limiting deviations and are enclosed in rectangular frames (Figure 9, *c*).

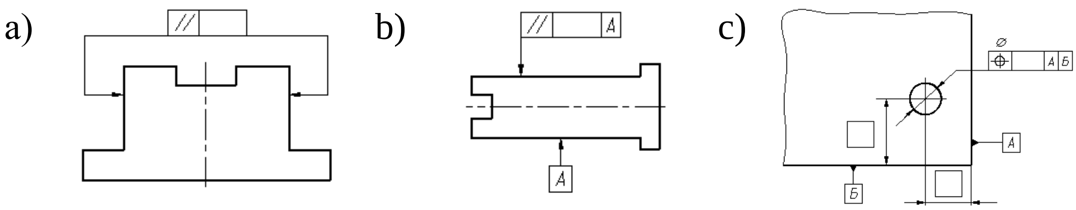


Figure 9 – Is the example of using the third frame graph to indicate the surface, which includes the deviation of the position

The qualified tolerances of the form and position are denoted by a conventional sign, which is placed:

- after the tolerance value, if the qualified tolerance is related to the actual dimensions of the element in question (Figure 8, *a*);
- after the letter designation of the base (Figure 8, *b*) or without the letter designation in the third part of the frame (Figure 8, *d*), if the qualified tolerance is related to the actual dimensions of the base element;
- after the tolerance value and the letter designation of the base (Figure 8, *c*) or without the letter designation (Figure 8, *e*), if the qualified tolerance is related to the actual dimensions of the considered and basic elements.

Surface roughnesses

In the process of forming details on their surface appears roughness – a series of alternating peaks and valleys of relatively small dimensions.

The roughness can be followed by a graver or other cutting tool, a copy of the unevenness of the molds or dies, may result from vibrations arising from cutting, as well as from other factors.

The influence of roughness on the operation of machine parts is manifold:

- surface roughness can disrupt the nature of the part matings due to the crushing or intensive wear of the profile peaks;
- in the butt joints, because of the considerable roughness, the stiffness of the joints decreases;
- the roughness of the shaft surface destroys the seals of various kinds that come into contact with them;
- unevenness, as stress risers, reduces the endurance strength of parts;
- the roughness affects the tightness of joints, the quality of galvanic and paint coatings;
- roughness affects the accuracy of measuring parts;
- corrosion of metal occurs and spreads faster on unfinished surfaces and so on.

A surface roughness is a set of surface irregularities with relatively small spacings on the base length.

The surface roughness is normalized by GOST 2789-73; and the designation for the surface roughness – by GOST 2.309-73.

Basic line is the length of the baseline *l*, the length of the line used to highlight the irregularities.

Mean line is the average profile line (m–m), the line having the form of a nominal profile, with RMS roughness value, all numerical values for roughness are counted from this line.

Surface roughness parameters (Figure 10) are given below:

- arithmetical mean deviation of the profile *Ra,* µm is the arithmetic mean of the absolute values of the profile deviations within the basic line

$$R_a = \frac{1}{n}\sum_{i=1}^{n} y_i$$

- profile ten-point height of irregularities R_z, µm is the sum of the average absolute height values of the five largest profile peaks and the depths of the five largest valleys of the profile within the basic line

$$R_z = \frac{1}{5}\left[\sum_{i=1}^{5}\left|y_{pi}\right| + \sum_{i=1}^{5}\left|y_{vi}\right|\right]$$

- the maximum height of profile *Rmax,* μm is the distance between the line of profile peaks and line of profile valleys within the basic line

- the mean spacing of profile irregularities *Sm*, μm is the average value of the profile unevenness spacing within the basic line

$$Sm = \frac{1}{n}\sum_{i=1}^{n} Sm_i$$

- mean spacing of local peaks of the profile *S*, μm is the average value of the spacing of the local peaks of the profile that are within the basic line

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

- reference length of the profile η_p the sum of the lengths b_i, cut off at a given level *p* in the profile material by a line, an equidistant midline within the basic line

$$\eta_p = \sum_{i=1}^{n} b_i$$

- relative reference length of the profile *tp*, % is the ratio of the reference length of the profile to the base line

$$t_p = \frac{\eta_p}{l}$$

Profile section level values *p* are selected from 5, 10, 15, 20, 25, 30, 40, 50, 60, 70, 80, 90 % from *Rmax*; base line values *l* are selected from 0,01; 0,03; 0,08; 0,25; 0,80; 2,5; 8; 25 mm.

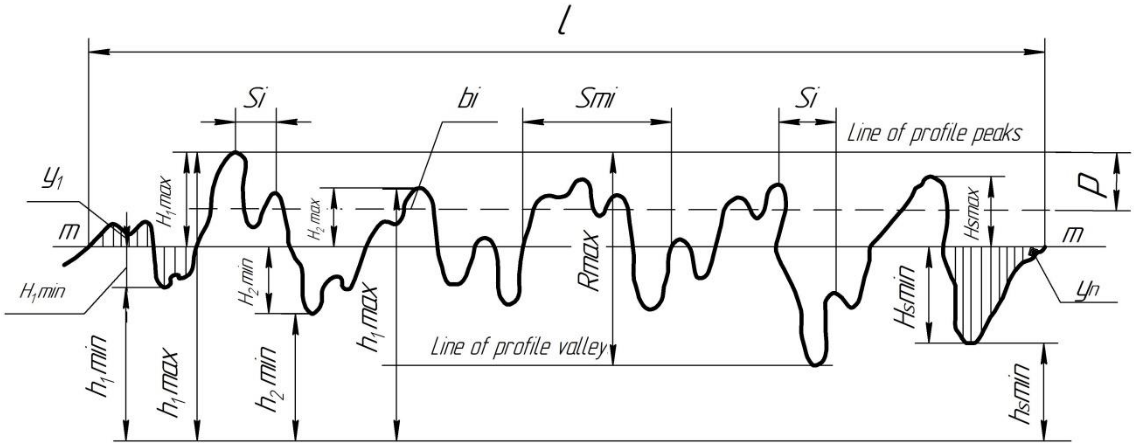


Figure 10 – Is a surface roughness profilogram

H_{imax} is the deviation of the five largest maxima of the profile; H_{imin} is the deviation of the five largest minima of the profile; h_{imax} is the distance from the highest points of the five largest maxima to line parallel to the mean and non-intersecting profile; h_{imin} is the distance from the highest points of the five largest minima to line parallel to the mean and non-intersecting profile; y_i is a profile deviation from the line *m-m*; *p* is a level of the profile section; b_i is lengths, cut off at the *p* level.

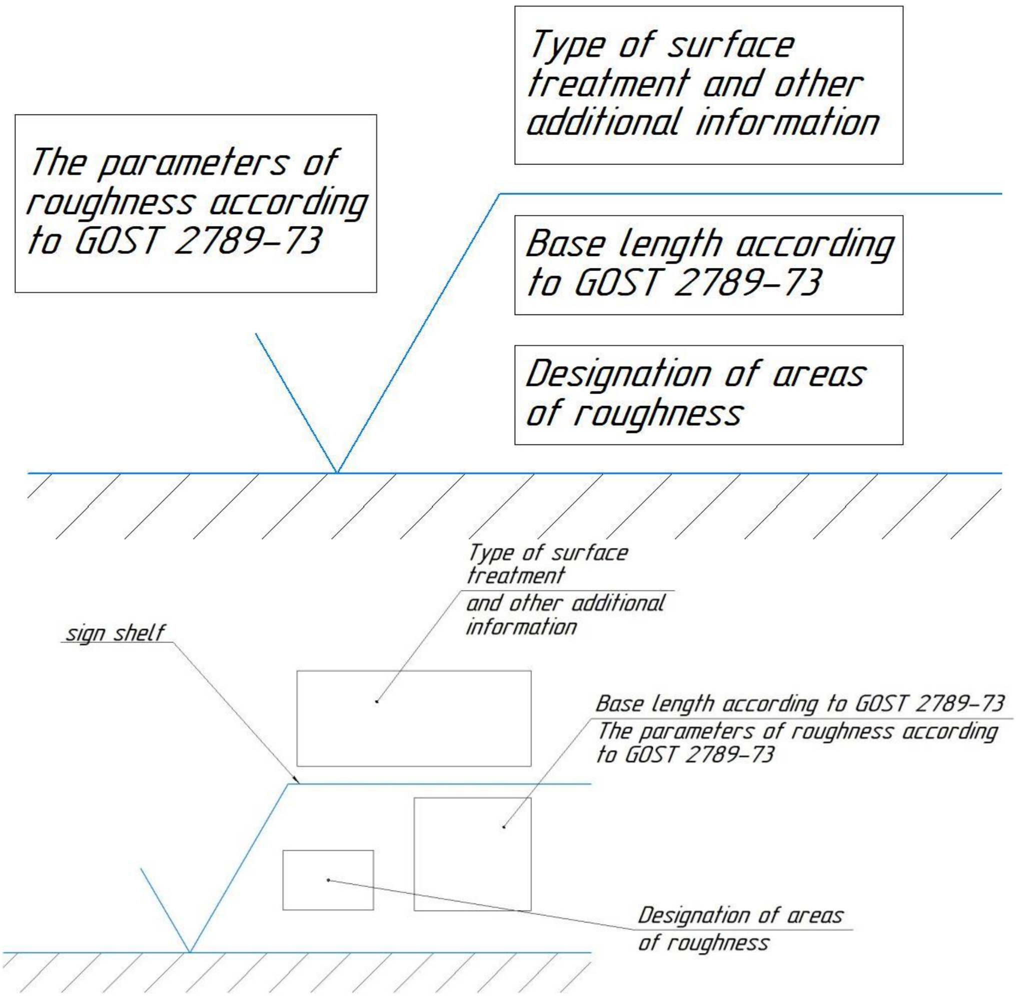
When selecting roughness parameters and the direction of surface irregularities, the application table E can be used, and when assigning values for the parameter *Ra* (taking into account the type of product, the planned accuracy of the dimensions and the processing type of the product), application E. The values of the parameter *Rz* and *Rmax* can be taken in the following way: according to the applications G, K the values of the parameter *Ra* are outlined and the values *Rz* or *Rmax* are chosen according to the above relations *Ra*, *Rz* and *Rmax*.

The relationships between the parameters *Ra*, *Rz* and the basic line *l* are given in Table 2 (using *Ra* is more preferable to using than the parameter *Rz*).

Table 2 – Is relations between the parameters *Ra* and *Rz*

Roughness grade	Basic line *l*, mm	*Ra*, preferable., μm	*Ra*, allowable, μm	Rz, μm
1	8,0	50	80; 63; 40	320; 250; 200; 160
2	8,0	25	40; 32; 20	160; 125; 100; 80
3	8,0	12,5	20;16,0;10,0	80; 63; 50; 40
4	2,5	6,3	10,0;8,0;5,0	40; 32; 25; 20
5	2,5	3,2	5,0; 4,0; 2,5	20; 16; 12,5; 10,0
6	0,8	1,6	2,5; 2,0; 1,25	10,0; 8,0; 6,3
7	0,8	0,80	1,25; 1,00; 0,63	6,3; 5,0, 4,0; 3,2
8	0,8	0,40	0,63; 0,50; 0,32	3,2; 2,5; 2,0; 1,60
9	0,25	0,20	0,32; 0,25; 0,160	1,60; 1,25; 1,00; 0,80
10	0,25	0,10	0,160; 0,125; 0,080	0,80; 0,63; 0,50; 0,40
11	0,25	0,050	0,080; 0,063; 0,040	0,40; 0,32; 0,25; 0,20
12	0,25	0,025	0,040; 0,032; 0,020	0,20; 0,16; 0,125; 0,100
13	0,08	0,012	0,020; 0,016; 0,010	0,100; 0,080; 0,063; 0,050
14	0,08	0,012	0,010; 0,008	0,050; 0,040; 0,032

The structure of the surface roughness designation is shown in Figure 11. If only the values of the parameter (parameters) are indicated in the roughness designation, then a sign without a shelf is used (Figure 12, *a*). The height *h* should be approximately equal to the height of the digits of the dimensional values, $H = (1{,}5...3)h$.



a)

b)

a) in accordance with GOST 2789-73;

b) in accordance with GOST 2789-73 as amended by No. 3 of May 28, 2002

Figure 11 – Is the surface roughness designation structure

To denote the surface roughness, depending on the surface treatment method, the signs shown in Figure 12 are used:

a) – if the method of surface treatment is not established by the designer;

b) – if the surface is obtained by removing the material layer (turning, milling, drilling, grinding, etc.);

c) – if the surface is obtained without removing the material layer (by casting, forging, bulk forming, rolling, etc.), as well as for surfaces not treated according to this drawing.

In the conventional designation for surface roughness, the numerical values of the parameters are indicated, with the following rules:

- the parameter values are indicated in one of three ways: maximum allowable value, for example Rz6,3, S0,32 etc.); limit values, which are placed in two lines (the rough value of this parameter is placed in the upper line);

- nominal values that result with limiting deviations, for example Rmax$40^{+20\%}$, Sm$0{,}32_{-10\%}$;

- the maximum deviations are selected as a percentage of the nominal values of the roughness parameters from the range of 10, 20, and 40%;

- deviations can be one-sided and symmetrical, for example Rz$1{,}00^{+20\%}$; Rz0,80±10%;;

- the values of the parameters *Ra*, *Rz* and *R* are indicated in μm, *Sm*, *S* and *l* – in mm; *tp* and *p* – in percent.

Note. In the example $tp_{50}70$±20 % relative reference length of the profile *tp*=70 % is indicated, with the sectional profile level *p*=50 %; if the ratios of the parameters *Ra*, *Rz* and *Rmax* and basic length *l* are assigned in accordance with GOST 2789-73, then the basic line should not be indicated in the roughness requirements and in the graphical symbol in the drawing; if two or more parameters are specified, then in the designation of roughness, the height parameter of the irregularities is recorded from top to bottom, then the spacing of the irregularities and the relative reference length of the profile; direction of lay is given only if necessary and indicated in the roughness designation by the conventional symbol given in Table 3.

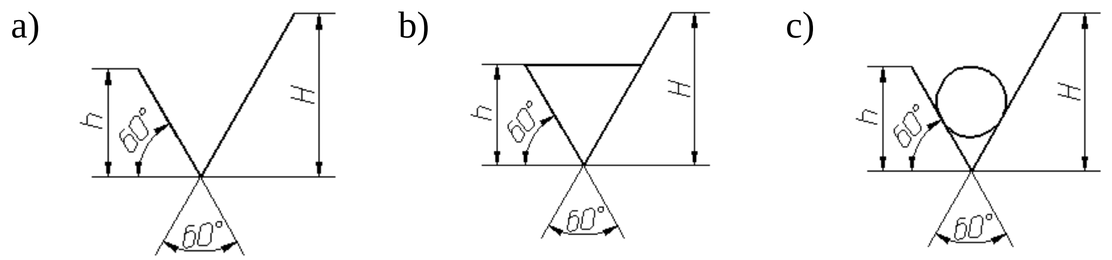
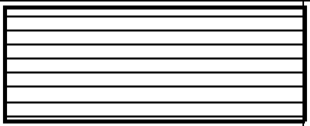
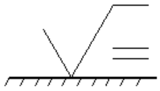
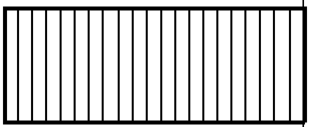
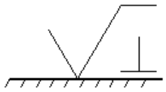
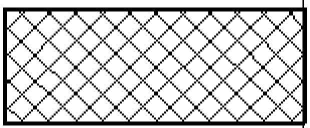
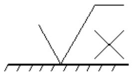
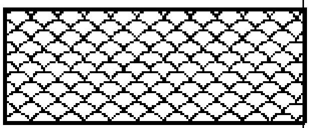
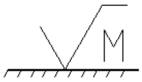
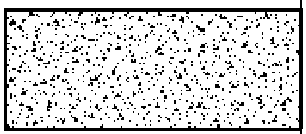
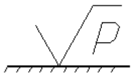
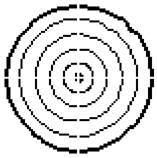
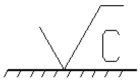
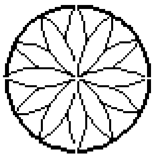
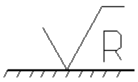


Figure 12 – Is signs used to refer to surface roughness

The processing type of surface (grinding, polishing, etc.) is indicated in the roughness designation, if it is the only one applicable to obtain the required surface quality.

For example, the roughness designation in figure 13 means: the surface is obtained by removing the material layer; final processing is grinding; the roughness of the surfaces is given by the limiting value of the parameter *Ra* = 0,08 μm, the limiting values of the mean spacing of the irregularities *Sm* = (0,063 ... 0,040) μm and the relative reference length of the profile *tp* = (50 ... 70) % for the sectional profile *p*=50 %; basic line *l* = (0,8 ... 0,25) mm; the direction of lay *M* is arbitrary.

Table 3 – Is name and surface lay, view and designation

Ser. No.	Name and surface lay	Layout view	Designation
1	Parallel lay – the irregularities are located parallel to the line depicting the surface in the drawing		
2	Perpendicular lay – the irregularities are perpendicular to the line depicting the corresponding surface in the drawing		
3	Cross-angular lay – when crossing in both directions of irregularities, it is inclined to the line depicting the corresponding surface in the drawing		
4	Multidirectional lay – different directions of irregularities with respect to the line depicting the corresponding surface in the drawing		
5	Point lay – the irregularities are arranged as separate points in relation to the line depicting the surface in the drawing		
6	Circular lay – the irregularities are approximately circular in relation to the center of the surface, to the roughness of which the requirements are established		
7	Radial lay – the irregularities are approximately radial with respect to the center of the corresponding surface.		

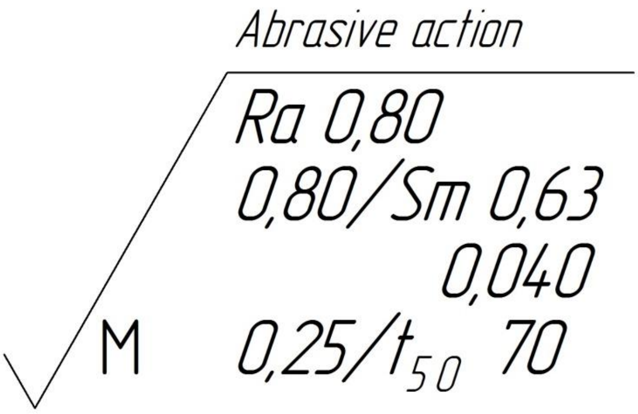


Figure 13 – Is roughness designation

It is allowed to apply a simplified designation of the surface roughness with an explanation of it in the technical specification of the drawing (Figure 14).

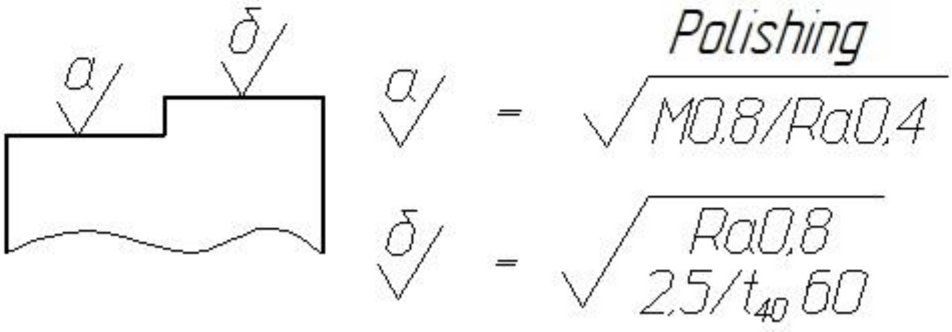


Figure 14 – Is the example of a simplified designation of surface roughness

When applying surface roughnesses designations on the drawings, the following basic rules must be observed.

The designations of the surface roughness in the item image are placed on the contour lines, extension lines (as close as possible to the dimension line), on the shelves of the leader lines, and if there is insufficient space on the dimensional lines or on their extensions and ruptures (Figure 15, *a*). On the invisible line, it is allowed to indicate the roughness if this line is dimensioned.

When specifying a product with a rupture, the roughness designation is applied only on one part of the image, closer to the point of dimensioning.

When specifying the same roughness for all item surfaces, the roughness designation is placed in the upper right corner of the drawing and the image is not applied (Figure 15, *c*). If for some item parts the uniform roughness is indicated, the designation of the same roughness is placed in the right corner of the drawing and the symbol is indicated in parentheses.

For example, for a cylindrical surface *Ra* is set no more than 3.2 μm, and for other surfaces Rz is not more than 50 μm (Figure 15, *b*).

If a part of the surfaces is not processed according to this drawing, then the signs, shown in Figure 15, *d*, are placed in the upper right corner of the drawing. If there are surfaces in the item whose roughness is not indicated, the sign in parentheses or the sign (Figure 15, *c*) cannot be carried to the upper right corner of the drawing.

The surface roughness designation of the repeated elements of the item (holes, grooves, gears, etc.), the number of which is indicated in the drawing, as well as the same surface on different projections, are applied once regardless of the images amount. For example, the surface roughness of 3 holes (Figure 19, *d*) is indicated only at one hole, since the figure indicates the number of holes.

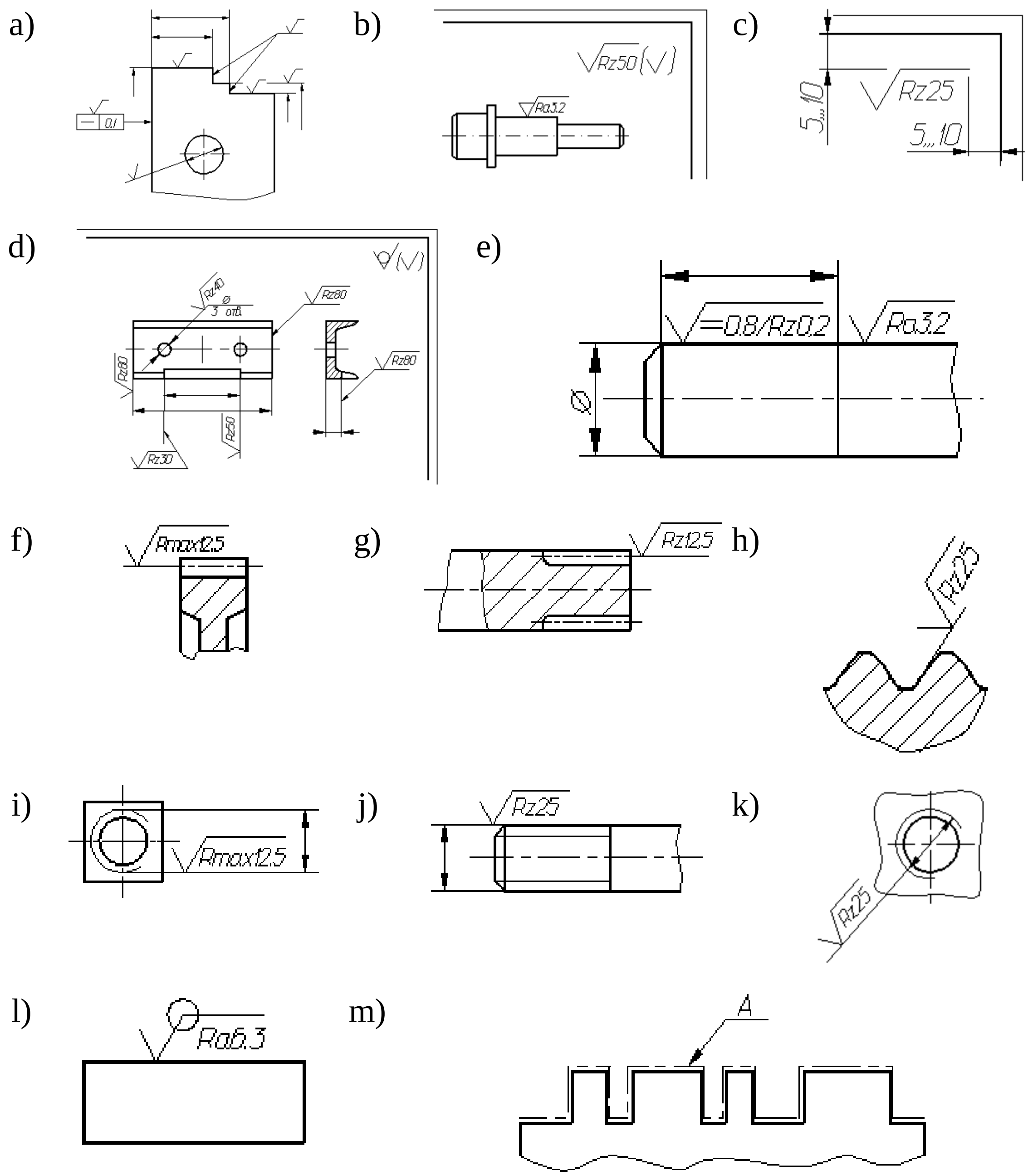


Figure 15 – Is the indication of surface roughness

If different roughnesses are assigned to different parts of the same surface, these sections are delineated by a thin line and are marked with appropriate roughness symbols (Figure 15, *e*).

The designation of the roughness of the working tooth surfaces of gear component, wormwheels, involute splines, etc., if their profile is not shown in the drawing, are conventionally applied on the reference surface line (Figure 15, f, g). If the tooth profile is shown in the drawing, the roughness of their surfaces is indicated, as shown in Figure 15, *i*.

The designation of the roughness of the surface of the thread profile is applied according to the general rules for the profile image (Figure 15, *i*) or conditionally on the extension line for indicating the thread size (Figure 15, *j*, *k*), on the dimension line or on its extension (Figure 15, *l*).

If there is a roughness of the surfaces forming the contour, it must be the same; the roughness designation is applied once in accordance with Figure 15, *m*. The diameter of the auxiliary sign is 4 ... 5 mm.

If the surface of a complicated configuration is marked with the same roughness, then it can be indicated in the technical specifications of the drawing with reference to the letter designation of the surface, which is placed on the shelf of the line-leader, for example, the surface roughness (Figure 15, p).

Interchangeability of plain gauges

Measurement is the finding of a numerical value of a physical quantity by experimental means with the help of special technical means (for example, universal measuring means).

Control is the compliance verification of the product quality indicators with the established requirements, i.e. determining whether the value of the controlled physical quantity is between its limit values or outside them, excluding the measurement process.

Gauges are non-scaled control instruments that allow the control of a part, excluding the measurement process (Figure 1).

Figure 1 – Is general view of gauges

The gauges are used to control the parts dimensions of 6–17 grades at known nominal sizes within 1 ... 500 mm.

There are types of controlled items and parameters:

- plain for cylindrical products;
- thread plug;
- spline;
- keyset, etc.

According to the number of simultaneously controlled components there are:

- elemental is to control individual linear dimensions;
- complex is for simultaneous control of several elements.

According to the conditions for validity evaluation of parts there are:

- normal;
- limiting.

When inspecting by normal gauges, the checked components validity is evaluated on the basis of subjective sensations of the supervising person (since the normal gauges must pass without effort, but without a clearance).

Limit gauges determine the size of parts, distributing them into three groups:

- accepted;

- rejected as a result of crossing over the upper tolerance limit;
- rejected as a result of crossing over the lower tolerance limit.

To work with the limit gauges, a less-qualified worker and controller are required and the objectivity of the control process is increased.

Limit gauges are made in pairs. One of them is called go-on, and the other is "no go-on".

For internal measurements, the nominal size of the go-on gauge is carried out through the nominal, and the "no go-on" through the maximum limiting dimensions.

For external measurements, the nominal size of the go-on gauge is carried out through the maximum, and "no go-on" through the minimum limiting dimensions.

The inspected item is considered as valid if the go-gauge passes over, and the "no go-on" gauge does not pass over the inspected item.

If the go-gauge does not pass, then the part is correctable rejected, if the no go-on gauge passes, then the components is an irreparable reject (Figure 2).

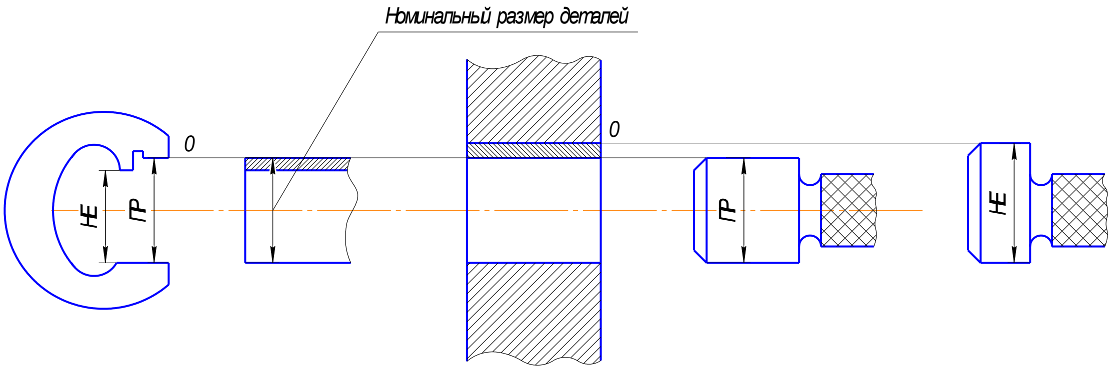


Figure 2 – Is the control scheme of parts with plain gauges

For technological use, the gauges are divided into:
- functional;
- control.

Functional gauges are used to control parts in workplaces during their manufacture. These gauges are used by factory workers and controllers of quality department.

Control gauges are designed to monitor or adjust functional gauges. They are no go-on and serve for the seizure and operation due to the wear and tear of functional gap gauges.

According to the current standards, gauges have the following designations:
- GO-F - go-on functional gauge;
- NO-F - no go-on functional gauge;
- GO-C - control gauge for a go-on functional (new) gauge;
- NO-C - control gauge for a no go-on functional (new) gauge;
- W-K - control gauge for wear control of the go-side of the functional gauge.

For the control of the shafts, the gaps are mainly used. The basic designs of gap gauges are established in accordance with GOST 18355-73 … 18368-73. Single-sided two limits gap gauges are the most common. Adjustable gaps are also applied; they can be adjusted to different sizes.

Adjustable gaps are less accurate and reliable, so they are used to control items of the 8^{th} and coarser grade.

To control the holes, gauge-plugs of the construction are used, which are according to the GOST 14807-69 … 14827-69. Gauge-plugs can be double-sided for sizes up to 50 mm and single-sided for sizes over 50 mm.

To reduce costs, there is a tendency to increase their wear resistance of gauges. For the production of gauges, the 20; 20C; C7; C8; CTZ; BB15 grades are used; for working surfaces of gauges, plates of the TR-6, TR-8 hard alloys are used.

The gauges with alloys have a wear resistance of 50 … 150 times higher than steel.

When constructing the limit gauges, the similarity principle (the Taylor principle) should be followed, according to which the gauges must be the prototype of the mating part with a length equal to the length of the joint (i.e., the gauges for the shafts must be in the form of rings) and control the dimensions along the entire length of the joint taking into account the form errors of the parts.

No-go gauges should have a short measuring length and a contact, approaching a pinpoint, to check the actual size of the part.

In practice, sometimes it is necessary to depart from the similarity principle due to the inconvenience of control. For example, the control of the go-on ring would require the removal of the part fixed in the centers of the machine. Therefore, instead of the rings, go-on gaps with wide measuring surfaces are used.

The gauges are marked in the following way:

- the nominal size of the item, $\varnothing$ 70;
- conventional designation of the limiting deviations of the item, H7;
- values of the limiting deviations of the item in mm, $^{+0,03}$;
- designation of gauge, GO, NO;
- a trade mark of the manufacturer.

Manufacturing tolerance of plain gauges and control gauges are regulated by GOST 24853-81, which provides for the following tolerances: the maximum dimensions of the holes D_{max} and D_{min} are the nominal dimensions of the plug gauges; limiting sizes of shafts d_{max} and d_{min} are the nominal sizes of gap gauges; TD and Td are tolerance limits of the items under the test; H and H_s are the manufacturimg tolerances for plug gauges, respectively, with cylindrical and spherical measuring surfaces; H_1 is manufacturimg tolerance of gap gauges; H_p is the manufacturimg tolerance of control gauges for the plug gauges control; Z is the deviation of the middle of the tolerance limits for the manufacture of a go-on plug-gauge relative to the D_{min} of the monitored hole; Z_1 is the deviation of the middle of the tolerance limits for the manufacture of a go-on plug gauge with respect to d_{max} of the monitored shaft; Y and Y_1 are the permissible output of the dimension of the worn-out plug gauge or the worn-out gap gauge, respectively, beyond the boundary of the admissible article's tolerance limits; α and α_1 are values to compensate for the error in the control of the holes and shafts, respectively, with nominal dimensions exceeding 180 mm.

Figures 3 and 4 show the schemes of the tolerances and deviations limits, respectively, of the gap gauges and plug gauges relative to the tolerance limits of the inspected items.

In the grades 6, 8 … 10, the H_1 tolerances for the gaps are 50 % higher than the H tolerances for plugs, which is explained by the greater complexity of gaps. Tolerances for all types of the H_p control gauges are the same.

For go-gauges that wear out during inspection, wear tolerance is provided. The permissible dimensional drift out of the item tolerance limits for worn-out gauge is regulated by the value of Y for the plugs and the value of Y_1 for the gaps. In grades 9 and coarser Y and $Y_1 = 0$.

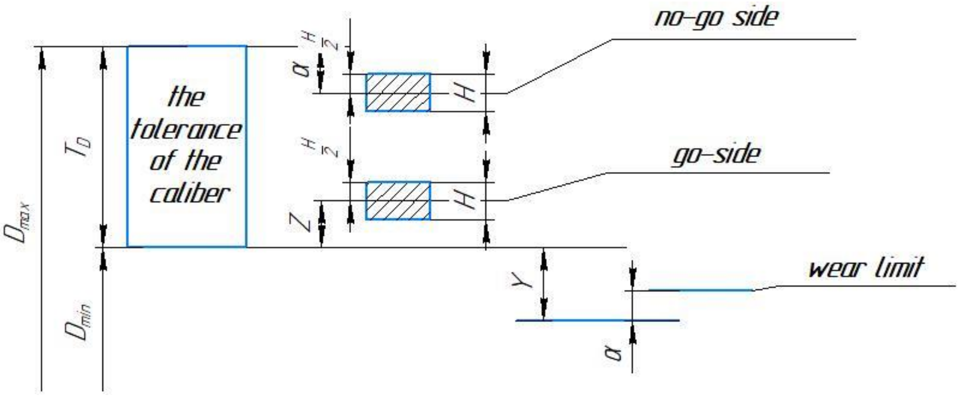


Figure 3 – Is the layout of gauge tolerance limits for hole inspection

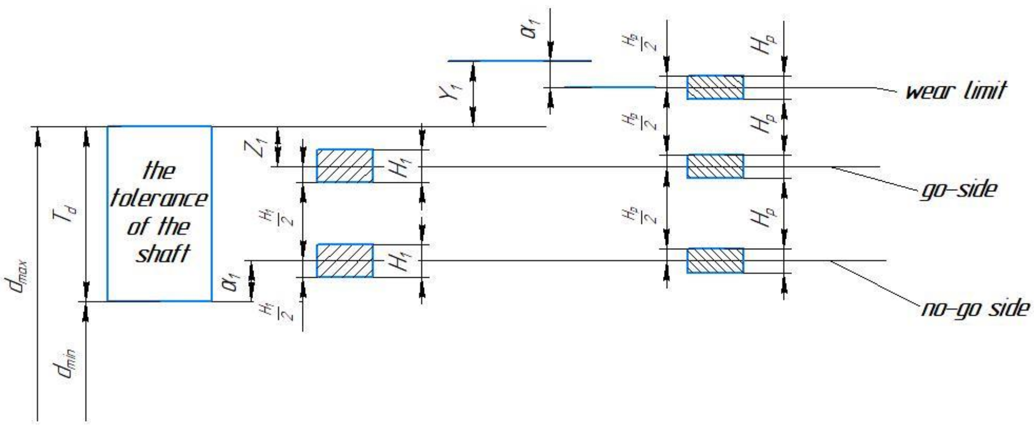


Figure 4 – Is the layout of gauge tolerance limits for the shafts and control gauges inspection

For all go-gauges, the tolerances limits H and H_1 are shifted inward to the tolerance limits of the item by the Z amount for plugs and the Z_1 value for gaps.

At nominal sizes more than 180 mm, the tolerance limits of the no-gauge and the wear limit of the go-gauge are also shifted inside the tolerance limits of the part by the α value for plugs and the α_1 value for the gaps.

Required is called the limiting size of the gauge, on which the new gauge is made.

The required dimensions of the gap is its minimum limiting size with a positive deviation, for the plug and control gauge – their maximum limiting size with a negative deviation. Thus, in the drawing, the deviation is placed in the ”body” of the gauge.

The dimensions of the gauges and their limiting dimensions are calculated using the formulas given in Tables 1 and 2. The tolerances and deviations of the gauges are taken in accordance with Annex L. The required dimensions of the gauges are rounded according to the following rules: dimensions of functional gauges for items of 15 ... 17 grades up to a whole micrometer; for products of grades 6 ... 14 and all control gauges – to values that are multiples of 0,5 μm, in the direction of reducing the production tolerance of the item.

Table 1 – Is formulas for calculating the required dimensions of limit gauges with nominal item dimensions up to 500 mm

Type of the gauge and its designation		Nominal dimensions of the item, mm	
		Up to 180	from 180 to 500
Plug gauge	Go new F-GO_{used}	$(D_{min}+Z+0{,}5H)_{-H}$	$(D_{min}+Z+0{,}5H)_{-H}$
	Go worn-out F-GO_{worn}	$D_{min}-Y$	$D_{min}-Y+\alpha$
	No go F-NO_{used}	$(D_{max}+0{,}5H)_{-H}$	$(D_{max}-\alpha+0{,}5H)_{-H}$
Gap gauge	Go new P- GO_{used}	$(d_{max}-Z_1-0{,}5H_1)^{+H_1}$	$(d_{max}\text{-}z_I\text{-}0{,}5H_I)^{+H_I}$
	Go worn-out F-GO_{worn}	$d_{max}+Y_1$	$d_{max}+Y_1-\alpha_1$
	Not go F-NO_{used}	$(d_{min}-0{,}5H_1)^{+H_1}$	$(d_{min}+\alpha-0{,}5H_1)^{+H_1}$
Control gauge	Go new F-GO_{used}	$(d_{max}-Z_1+0{,}5H_p)_{-H_p}$	$(d_{max}-Z_1+0{,}5H_p)_{-H_p}$
	Go worn-out F-GO_{worn}	$(d_{max}+Y_1+0{,}5H_p)_{-H_p}$	$(d_{max}+Y_1-\alpha_1+0{,}5H_p)_{-H_p}$
	Not go F-NO_{used}	$(d_{min}+0{,}5H_p)_{-H_p}$	$(d_{min}+\alpha_1+0{,}5H_p)_{-H_p}$

Fits and limits of rolling bearings

Bearings serve as supports for shafts and rolling axes. To avoid a loss of efficiency, loss mechanism in bearings should be minimal. The quality of the bearings strongly depends on the runnability and durability of the machines. By the form of friction, bearings are divided into sliding bearings and rolling bearings. This is a special group of fits of plain cylindrical joints.

Rolling bearings are standard assembly units of particular accuracy, which are manufactured at specialized bearing plants with special precision equipment.

The industry of the CIS countries produces bearings with an outer diameter from 1,5 to 2600 mm. Bearings ∅ 20…200 mm are produced in large series.

Bearings have a complete external interchangeability over the bearing seat area, determined by the outer diameter of the outer ring and the inner diameter of the inner ring and incomplete internal interchangeability between the rolling elements and rings.

The rings of the bearings and the rolling elements are chosen by a selective method. The complete external interchangeability allows to mount and replace worn rolling bearings quickly maintaining their good quality.

The quality of bearings, other things being equal, is determined by:

- interface dimension accuracy *d*, *D*, ring width *B*, and for taper roller bearing also by mounting height accuracy *T*;
- form accuracy and interrelational accuracy of the surfaces of the bearing rings and their roughness;
- the form and size accuracy of the rolling elements in a single bearing and the roughness of their surfaces;
- the rotating accuracy, characterized by the radial and axial runout of the bearing races and face rings.

Depending on the specified accuracy indicators according to GOST 520 - 2011, five accuracy classes are established in order of increasing accuracy: 0; 6; 5; 4; 2.

Geometrical and linkage parameters of rolling bearings:

- d, D are the nominal diameters of the hole of the inner ring and the bearing seat area of the outer ring;
- B is the width of the inner and outer rings;
- C is the width of the outer ring if the inner ring has a different width;
- R_i is the radial runout of the bearing race of the inner ring relative to its hole;
- R_a is the radial runout of the bearing race of the outer ring relative to its outer cylindrical surface;
- T is mounting height of single-row conical roller bearing;
- U_p is ring width variation;

- d_m and D_m are the mean bore diameters of the inner and outer ring, which are calculated by formulas

$$d_m = 0{,}5\ (d_{max} + d_{min});\ D_m = 0{,}5\ (D_{max} + D_{min}), \qquad (1)$$

where d_{max} , D_{max} and d_{min} , D_{min} are the maximum and minimum diameters of the bearing seat area of the bearing rings.

Example. The permissible radial runout of the bearing races of inner rings of the 2nd class of accuracy is 10 times less than for 0 class bearings.

For most general-purpose mechanisms, bearings of 0-grade accuracy are used. Bearings of higher accuracy classes are used at high rotational speeds and in cases where high accuracy of shaft rotation is required.

Example:

a) turning spindles are supported by the 5th class bearings;

b) grinding spindles are supported by the 4th class bearings;

c) gyroscopic instruments are supported by the 2nd class bearings.

The accuracy class is indicated through a dash before the bearing designation: 6–312; 312 (0 class).

To ensure a normal life of the rolling bearings, parts, that are joined with them, must have a certain accuracy of the following parameters:

a) dimensions;

b) the surface forms;

c) surface positions;

d) roughness.

Mating parts are performed according to the following grades:

Bearing class	Grade	
	hole	shaft
0; 6	7	6
4; 5	6	5
2	5	4

The form deviation (cylindricity tolerance) of the hole and the shaft must not exceed 1/4 of the size tolerance for the bearings of the 0 and 6th classes.

The roughness of the bearing seat area of the shafts and holes in the housings should not exceed the following values (parameter R_a).

Bearing class	Shafts		Holes	
	d≤80мм	d > 80мм	D≤80mm	D > 80mm
0;	1,25	2,5	1,25	2,5
6; 5	0,63	1,25	0,63	1,25

4	0,32	0,63	0,63	1,25

In addition to these factors, the fit on the shaft and in the housing has a significant impact on the bearings life.

To reduce the nomenclature, bearings are manufactured with variations in the inner and outer diameters, independent of the fit, over which they will be mounted. For all accuracy classes, the upper deviation of the port diameters is assumed to be 0.

Thus, the diameters of the outer ring D_m and inner ring d_m are taken as the diameters of the basic shaft and the basic hole, i.e. a hole system is used to connect the inner ring to the shaft, and the shaft system is used to connect the outer ring to the housing.

The most significant difference between bearing fits from the ordinary ones is the position of the tolerance limits for the inner ring hole of the bearing. This hole is basic, but its tolerance limit is located down from the zero line.

To connect the rolling bearings to the shafts and bodies, GOST 3325-55 set the tolerance limits selected from the USTF (Table 1). Limit deviations for these tolerance limits are taken in accordance with GOST 25346-82.

Bearing operating conditions are varied, therefore the standard recommends a number of fit for the outer and inner rings.

The choice of fits for the rolling bearings on the shaft and in the housing depends on the type and size of the bearing, the conditions of its operation, the value and nature of the operating loads and the type of loading of the rings. There are three main types of loading rings: local, circulation, oscillation.

At local loading, the ring receives a constantly directing resulting radial load F_r only by a limited portion of the circumference of the bearing race and transmits it to the corresponding limited portion of the seat area of the shaft or housing. This loading occurs, for example, when the ring does not rotate relative to the load.

Table 1 – Is tolerance limits of matching sites for housing shafts and holes for rolling bearings (GOST 3325-55)

Grade	Shaft									
	Fundamental deviations									
	f	g	h	j_s	(j)	k	m	n	p	r
4	-	g4	h4	j_s4	-	k4	m4	n4	-	-
5	-	g5	h5	j_s5	(j5)	k5	m5	n5	-	-
6	f6	g6	h6	j_s6	(j6)	k6	m6	n6	p6	r6
7	-	-	h7	-	-	-	-	-	-	r7
8			h8	-	-	-	-	-	-	-
Grade	Housing hole									
	Fundamental deviations									

	E	G	H	J_s	(J)	K	M	N	P
4	-	-	H4	j_s4	-	-	-	-	-
5	-	-	H5	j_s5	-	K5	M5	-	-
6	-	G6	H6	j_s6	(J6)	K6	M6	N6	P6
7	-	G7	H7	j_s7	(J7)	K7	M7	N7	P7
8	E8	-	H8		-	-	-	-	-

At a circulation loading, the ring senses the resulting radial load F_r, successively by the entire bearing race and transmits it to the entire seat area of the shaft or housing. This loading of the ring is obtained by rotating it and the constantly directing load F_r.

Oscillating loading of the ring is the kind of loading in which a fixed bearing ring perceives a resultant that is bounded by section of the bearing race, and the resultant itself does not rotate the entire circle, but oscillates. Such loading occurs in the crank, bearing assemblies of cam gears.

Circularly loaded rings should be installed due to the fits, ensuring their fixity with respect to the mating part. (interference and transitional fits at high load).

Locally loaded rings are installed due to clearance fit at low load. These fits allow the ring to rotate gradually along the seat area, which reduces the wear unevenness of the ring and increases the service life of the rolling bearings.

Oscillatory loaded rings should be installed due to transitional fits.

The most important is the fit of a circulating loaded ring. It is chosen due to the radial load intensity, which is calculated by the formula

$$P_{F_r} = \frac{F_r}{b} \cdot K_1 \cdot K_2 \cdot K_3, \qquad (2)$$

where F_r is a radial force, operating upon the support, H;

b is working width of the bearing ring, b = B - 2r, mm;

B is a bearing width;

r is the bevel size of the bearing ring;

K_1 is a dynamic fit coefficient, depending on the loading conditions: with an overload of up to 150 %, moderate shocks and vibration K_1 = 1; at an overload of up to 300 % of strong impacts and vibrations K_1 = 1,8;

K_2 takes into account the degree of easing of the interference fit with a hollow shaft or thin walled vessel: with a solid shaft K_2 = 1;

K_3 takes into account the distribution unevenness of the radial load F_r between rows of rollers in two-row conical roller bearing or between double ball bearing in the presence of an axial load F_a on support: for one-row bearings K_3 = 1.

On the basis of the calculated intensity of the load, according to the tables of the reference book, the tolerance limit of the shaft or hole is selected.

Fits of locally and oscillatory loaded rings are selected without calculation according to the recommendations of the reference book tables.

Designation of rolling bearings on the drawings: L - Lager bearing.

Dimension chains

A dimension chain is a set of dimensions forming a closed loop system and directly participating in the solution of the problem posed. For example, using dimension chains, it is possible to determine the accuracy of the relative positioning of the axes and surfaces of a single part (a sub-dimensional chain) or several parts in an assembly unit or mechanism (an assembly dimension chain). Closure of the dimensional contour is a necessary condition for the compilation and analysis of the dimension chain. However, on the working drawing, the dimensions should be shown as an open circuit; the size of the closing link is not specified, since it is not required for processing. The dimensions that form the dimensional chain are called **the links of the dimension chain.**

According the positional relationship of the links, the dimension chains are divided into band and spatial. A dimension chain is called **band** if all its links are located in one or several parallels. **Spatial** is called a dimension chain, the links of which are not parallel to one another and lie in unparalleled planes. Dimension chains, links of which are linear dimensions, are called **linear**. The dimension chains, the links of which are angular dimensions, are called **angular**. When analyzing the accuracy of electrical and electronic elements of machines and devices, the chains are used links of which are the values of resistance, capacitance, inductance, current strength, voltages and other physical parameters.

At designing, the accuracy of items is provided with the help of **engineering dimension chains,** and at manufacturing - with using **manufacturing dimension chains**, that shows the dimension relationships of the workpiece in the process or the size of the MFTW system (machine-fixture-tool-workpiece). When the problem of measuring quantities characterizing the accuracy of a item is solved, **dimension measuring** chains are used, the links of which are the dimensions of the system "measuring means - the measured part".

The dimension chain consists of constituent links and master one. **The closing link** is called the size that is obtained last during the machining of the part, the assembly of the machine unit or the measurement. Its value and accuracy depend on the values and accuracy of the remaining (constituent) dimensions of the circuit. **The constitutional link** is a link of a dimension chain, the change of which causes a change in the closing link (but cannot and should not cause a change in the initial link). The component dimensions are A_1, A_2..., A_{m-1} (for the chain A), B_1, B_2, ..., B_{m-1} (for the chain B), and so on. **The initial link** is the link of the dimension chain, the given nominal size and the limiting deviations which determine the functioning of the mechanism and must be ensured as a circuit result of the dimensional chain. Based on the limiting values of this size, the tolerances and deviations of all other chain sizes are calculated. In the process of assembly, original size as a rule, becomes the closing one. In the sub-dimensional dimension chain, the size, based on the accuracy of which the degree of accuracy of the remaining dimensions is determined, is also called the initial.

Closing dimension $_{\Delta}A$ in three link chain (Figure 1) depends on the size of $A1$, called **increasing** (the more the size, the larger the value of $_{\Delta}A$), and the size $A2$, called **reducing** (if it increases then $_{\Delta}A$ decreases). The closing link can be positive, negative or equal to zero. The dimensional chain can be conditionally represented in the form of a circuit (Figure 1, *b*, *c*). By the scheme it is convenient to identify increasing and decreasing links. Above the letter designations of links an arrow is represented pointing to the right for increasing links and to the left for decreasing links.

Design and analysis of dimension chains allows:

1. To establish a quantitative relationship between the dimensions of the machine parts, to specify the nominal values and tolerances of the interconnected dimensions, based on the operational requirements and the economic accuracy of machining the parts and assembling the machine;

2. To identify the most cost-effective type of interchangeability (complete or incomplete);

3. To achieve the most accurate dimensioning in working drawings;

4. To determine the operating tolerances and convert the design dimensions to working ones (in case of mismatching of technological bases with design ones).

Calculation of dimension chains and their analysis is an obligatory stage of designing machines, contributing to the improvement of quality, ensuring interchangeability and reducing the labor intensity of their manufacture.

The essence of the calculation of the dimension chain is to establish the tolerances and limit deviations of all its links, based on the requirements of the item design and the technology used. In this case, two problems are solved:

1) determination of the nominal size, limiting deviations and the tolerance of the closing link according to the specified nominal dimensions and the limiting deviations of the component links (in cases where it is required to inspect the tolerance conformity of the closing dimension to the tolerances of the component sizes indicated in the drawing, – revised design);

2) determination of the tolerance and limiting deviations of the component sizes from the specified nominal dimensions of all chain sizes and the given limiting size of the original size (for the design calculation of the dimension chain).

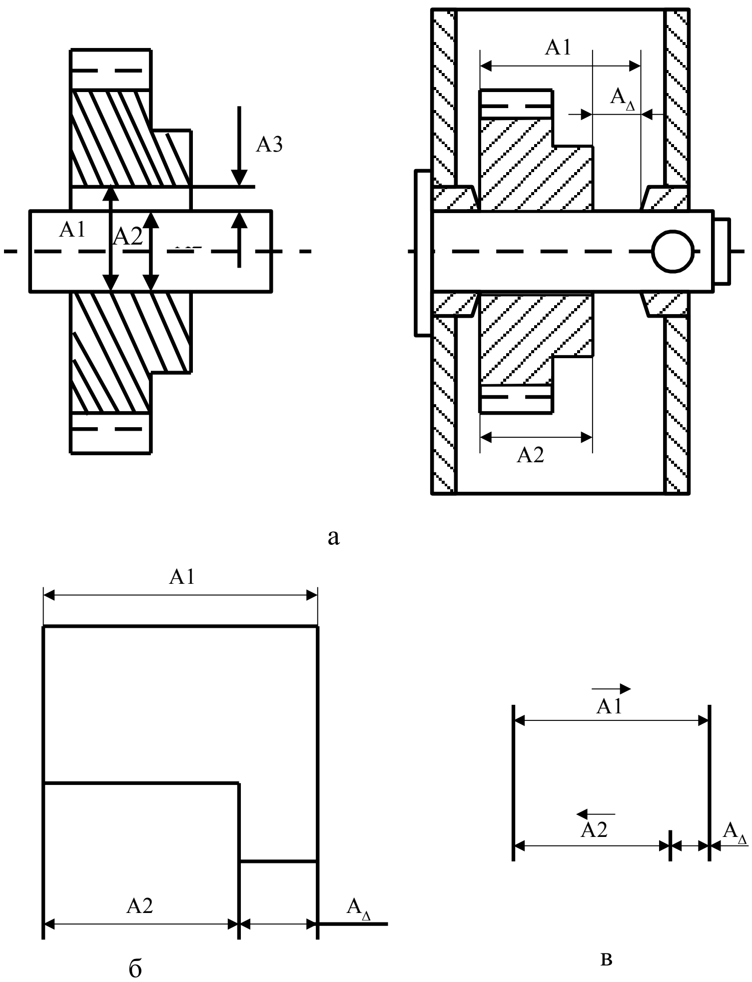


Figure 1 – Is a dimension chain circuits

The analytical solution of the design problems of the dimensional chains allows us to determine the following methods for achieving the specified accuracy of the closing link of the dimensional chain:

1) a design method that ensures complete interchangeability;

2) the method of incomplete interchangeability, performed on technology-based extended tolerances on the parts dimensions;

3) the method of selected interchangeability or selective assembly, consisting of manufacturing parts with relatively wide technologically feasible tolerances, sorting these parts into an equal number of groups with narrower group tolerances and assembling them after setting to specific groups;

4) the adjustment method in which a compensating link is chosen from the set of constituent links, the change of which provides a desired accuracy of the closing link without removing the material from the component links;

5) the fitting method, in which one of the constituent links of the dimension chain is projected by changing the dimensions, and in which a desired accuracy of the closing link is achieved by removing the chips.

For the calculation of dimension chains with parallel links for complete interchangeability the following formulas are used.

Nominal size of the closing link

$$A_{\Delta} = \sum_{j=1}^{n} A_j \qquad (1)$$

or

$$A_{\Delta} = \sum_{j=1}^{n} A_j - \sum_{j=1}^{p} A_j \qquad (2)$$

where m-1 is the total number of constituent links;

n is the number of increasing links;

p is the number of decreasing links.

Limit size of the closing link

$$A_{\Delta\max} = \sum_{j=1}^{n} A_{j\max} - \sum_{j=1}^{p} A_{j\min} \qquad (3)$$

$$A_{\Delta\min} = \sum_{j=1}^{n} A_{j\min} - \sum_{j=1}^{p} A_{j\max} \qquad (4)$$

where $A_{j\max}$ and $A_{j\min}$ are the limiting dimensions of the constituent links are calculated using formulas (2) and (3).

Limit deviations of the closing link

$$ESA_{\Delta} = \sum_{j=1}^{n} ESA_j - \sum_{j=1}^{p} EIA_j \qquad (5)$$

$$EIA_{\Delta} = \sum_{j=1}^{n} EIA_j - \sum_{j=1}^{p} ESA_j \qquad (6)$$

Tolerance of closing link

$$TA_{\Delta} = \sum_{j=1}^{m-1} TA_j \qquad (7)$$

Tolerance of the link, due to which the equality of the link tolerances in the dimension chain is satisfied (this link will be called the adjusting link)

$$TA_y = TA_\Delta - \sum_{j=1}^{m-2} TA_j \tag{8}$$

Mean tolerance of component links

$$T_m = \frac{TA_\Delta}{m-1} \tag{9}$$

The number of tolerance units by which the grade is selected for the tolerances of the constituent links

$$a = \frac{TA_\Delta}{\sum_{j=1}^{m-1} i_j} \tag{10}$$

where a is rounded according to the table in Annex B;

i is the tolerance unit, taken from the annex table D.

The limiting deviations of the links, expressed in terms of the coordinates of the middle of the tolerance limits E_c (Figure 2)

$$ESA_\Delta = E_c A_\Delta + 0{,}5TA_\Delta \,;\, ESA_j = E_c A_j + 0{,}5TA_j \tag{11}$$

$$EIA_\Delta = E_c A_\Delta - 0{,}5TA_\Delta \,;\, EIA_j = E_c A_j - 0{,}5TA_j \tag{12}$$

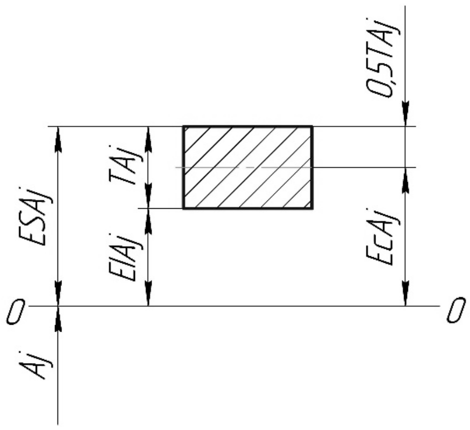


Figure 2 – Is limit deviations of the links

Mean deviation of the closing or initial link

$$E_c A_\Delta = \sum_{j=1}^{n} E_c A_j - \sum_{j=1}^{p} E_c A_j \tag{13}$$

The basic formulas for the probability of the design method of dimension chains are following.

Tolerance of the closing (initial) link

$$TA_{\Delta} = \sqrt{\sum_{j=1}^{m-1} TA_j^2} \tag{14}$$

Mean tolerance

$$T_m = \frac{TA_{\Delta}}{\sqrt{m-1}} \tag{15}$$

Number of tolerance units

$$a = \frac{TA_{\Delta}}{\sqrt{\sum_{j=1}^{m-1} i_j^2}} \tag{16}$$

Formulas (14) ... (16) are valid under the following conditions: random distribution of real deviations obeys the normal distribution law; the arithmetic mean size of the links (deviations) coincides with the middle of the tolerance limit and beyond the distribution field of the random variable is no more than 0,27 % of the actual dimensions of the links.

Example 1. For the batch of pins Ø 40 mm the maximum dimensions are $d_{\max} = 40{,}009$ mm, $d_{\min} = 39{,}984$ mm. From the batch the pins are taken, having dimensions $d_{r1} = 40{,}012$ mm and $d_{r2} = 39{,}976$ mm. Determine the suitability of these pins by comparing the actual dimensions and deviations with the limit dimensions and deviations.

Solution. Compare the actual dimensions with the limiting ones. In the first case, $d_{r1} > d_{\max} : d_{r1} - d_{\max} = 40{,}012 - 40{,}009 = 0{,}003$ mm is a reparable spoilage.

In the second case, $d_{r2} < d_{\min}$: $d_{r2} - d_{\min} = 39{,}976 - 39{,}984 = -0{,}008$ mm is an irreparable spoilage.

We compare the actual deviations with the limit deviations. Limit deviations are determined

$$es = 0{,}009 \text{ mm}; \; ei = -0{,}016 \text{ mm}.$$

Determine the actual deviation of the pins

$$e_{r1} = d_{r1} - D = 40{,}012 - 40 = 0{,}012 \text{ mm},$$

$$e_{r2} = d_{r2} - D = 39{,}976 - 40 = -0{,}024 \text{ mm}.$$

At the first pin $e_{r1} > es$: $e_{r1} - es = 0{,}012 - 0{,}009 = 0{,}003$ mm; at the second pin $e_{r2} < ei$: $e_{r2} - ei = -0{,}024 - (-0{,}016) = -0{,}008$ mm.

For the parts machining and the evaluation of their accuracy, limit dimensions or limit deviations may be specified.

Example 2. Given: $d_{\max} = 44{,}975$ mm, $d_{\min} = 44{,}950$ mm, $d = 45$ mm. Determine the tolerance for the limit dimensions and the limit deviations. Construct a layout of tolerance limits. Record the nominal size with the limiting deviation.

Solution. Determine the tolerance to the limit dimensions: $Td = 44{,}975 - 44{,}950 = 0{,}025$ mm. Find the limit deviations: $es = 44{,}975 - 45 = -0{,}025$ mm; $ei = 44{,}950 - 45 = -0{,}050$ mm. Tolerance of limit deviation: $Td = es - ei = -0{,}025 - (-0{,}050) = 0{,}025$ mm, i.e. for both methods of calculation the tolerance is the same.

The layout of the tolerance limits through the limit dimensions (on a scale) is shown on figure 1, a, the simplified layout is shown on figure 1, b (scale *25 μm* in *1 cm*).

The specified size is recorded in technological normative documents: $45^{-0{,}025}_{-0{,}050}$.

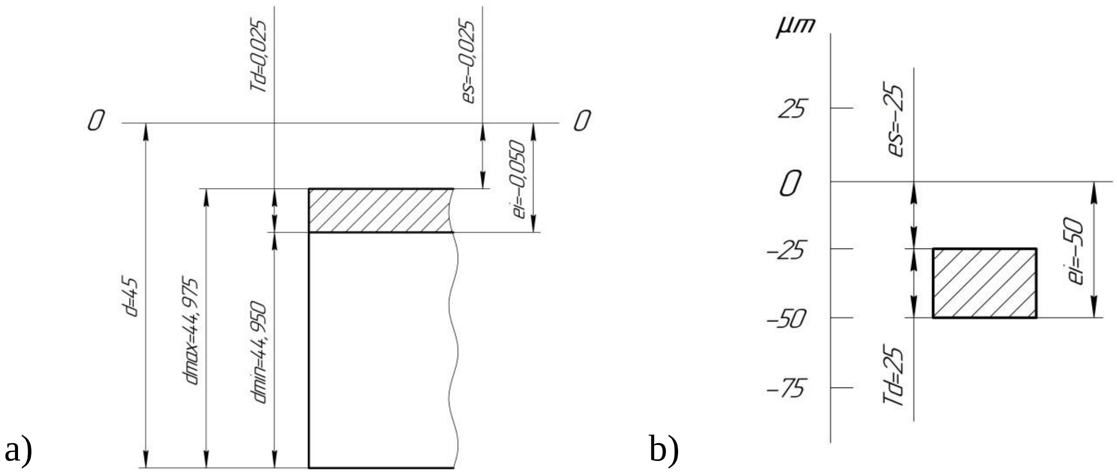


Figure 1 – Is a layout of the tolerance limits of the hole and shaft

Example 3. Tolerance for Ø 28 mm is $Td = 52$ μm. The following tolerances are set for Ø 280 mm: 52,130 and 210 μm. Determine which of the tolerances of Ø 280 mm is more, equal and less than the tolerance of Ø 28 mm.

Solution. According to Annex D, tolerance values for Ø 280 mm are found. According to the dimensions interval $18...30$ mm $i_{28} = 1{,}31$; for Ø 280 mm is $250...315$ mm $i_{280} = 3{,}22$ in the interval of dimensions.

According to the number of tolerance units for Ø 280 mm

$$a_{28} = \frac{Td}{i_{28}} = \frac{52}{1{,}31} = 39{,}7\,.$$

Compare the numbers of tolerance units a $_{280}$ for three cases with a_{28}.

$Td = 52$ μm; $a_{280} = \dfrac{Td}{i_{280}} = \dfrac{52}{3{,}22} = 16{,}3 < a_{28} = 39{,}7$ – for Ø 280 the tolerance is lower than for Ø 28 mm.

$Td = 130$ μm; $a_{280} = \dfrac{Td}{i_{280}} = \dfrac{130}{3{,}22} = 40{,}04 \approx a_{28} = 39{,}7$ – tolerances for both sizes have the same accuracy.

$Td = 210$ μm; $a_{280} = \dfrac{Td}{i_{280}} = \dfrac{210}{3{,}22} = 65{,}3 > a_{28} = 39{,}7$ – Ø 280 has bigger tolerance than at Ø 28 mm.

Example 4. For the hole and the shaft with nominal diameters $D = 20$ mm, given: $ES = +41$ μm, $ei = -61$ μm, $TD = 21$ μm. Give the conventional signs of these dimensions with tolerances, i.e. record nominal sizes with limit deviations, construct a simplified layout of tolerance limits.

Solution. In the condition, only the upper deviation of the hole tolerance limit and the lower deflection of the shaft tolerance limit are specified. Unknown deviations (with known tolerances) are determined by formula (1. 5), solving it with respect to unknown parameters:

- for hole $EI = ES - TD = 41 - 21 = 20$ μm,
- for shaft $es = ei + Td = -61 + 21 = -40$ μm.

Conventional signs of nominal sizes with limit deviations have the form: hole Ø $20^{+0,041}_{+0,020}$; shaft Ø $20^{-0,040}_{-0,061}$ (figure 2). The layout of the tolerance limits is shown on figure 3.

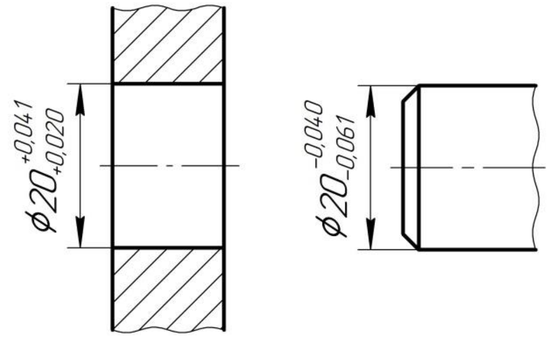


Figure 2 – Is a conventional sign of nominal hole and shaft dimensions

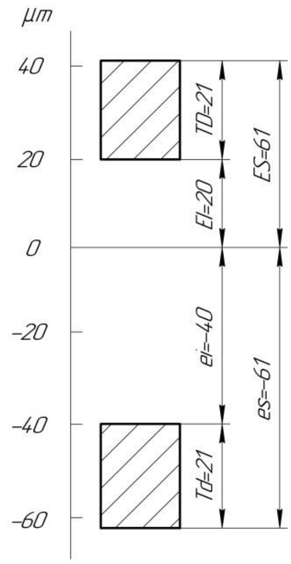


Figure 3 – Is tolerances layout of the hole and shaft

Note. For such problem solving, first it is necessary to draw (not to scale) simplified layouts of tolerance limits, showing the given and required values on them. This will facilitate the compilation of calculation formulas and verification of calculations (figure 4).

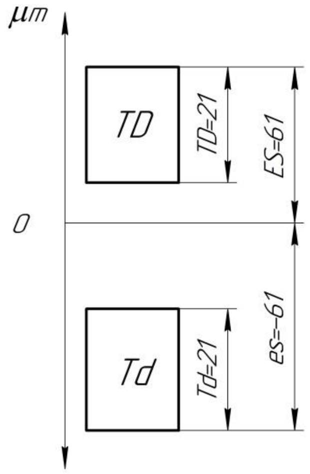


Figure 4 – Is a simplified layout of the tolerances of the hole and shaft

Example 5 Given the hole Ø $20^{+0,021}$ mm; the shaft Ø $20^{-0,007}_{-0,020}$ mm. Calculate the clearance fit: determine the nominal and limit dimensions; limit and average deviations, clearance limit, tolerances of the hole, the shaft and the fit. Construct a layout tolerance limits by limiting size and simplified one.

Solution. The nominal size and deviations are determined.

Nominal size $D = 20$ mm.

the hole deviation $ES = +0{,}021\ mm = +21\ \mu m;\ EI = 0;$

$$E_m = 0{,}5(ES + EI) = 0{,}5(21 + 0) = 10{,}5 \text{ µm};$$

the shaft deviation $es = -0{,}007\ mm = -7\ \mu m;\ \ ei = -0{,}020\ mm = -20\ \mu m;$

$$e_m = 0{,}5(es + ei) = 0{,}5[-7 + (-20)] = -13{,}5 \text{ µm}.$$

Limit sizes are determined for the hole $D_{\max} = 20 + 0{,}021 = 20{,}021$ mm; $D_{\min} = 20 + 0 = 20$ mm; for shaft $d_{\max} = 20 - 0{,}007 = 19{,}993$ mm; $d_{\min} = 20 - 0{,}020 = 19{,}980$ mm.

The maximum clearance is determined $S_{\max} = 21 - (-20) = 41$ µm; the minimum interference is $S_{\min} = 0 - (-7) = 7$ µm. The clearance and the interference can be calculated from the limit dimensions

$$S_{\max} = 20{,}021 - 19{,}980 = 0{,}041\ mm = 41 \text{ µm}$$

$$S_{\min} = 20 - 19{,}993 = 0{,}007\ mm = 7 \text{ µm}.$$

The hole tolerance is determined $TD = 21 - 0 = 21$ µm; the shaft tolerance is $Td = -7 - (-20) = 13$ µm; the fit tolerance is $TП = TD + Td = 21 + 13 = 34$ µm. The tolerance clearance fit is equal to clearance tolerance: $TS = S_{\max} - S_{\min} = 41 - 7 = 34$ µm, i.e. $TS = TП$.

Based on the results of the calculation, we will construct the layout of tolerance limits by limit dimensions without scale (figure 5, *a*) and a simplified layout in scale (figure 5, *b*).

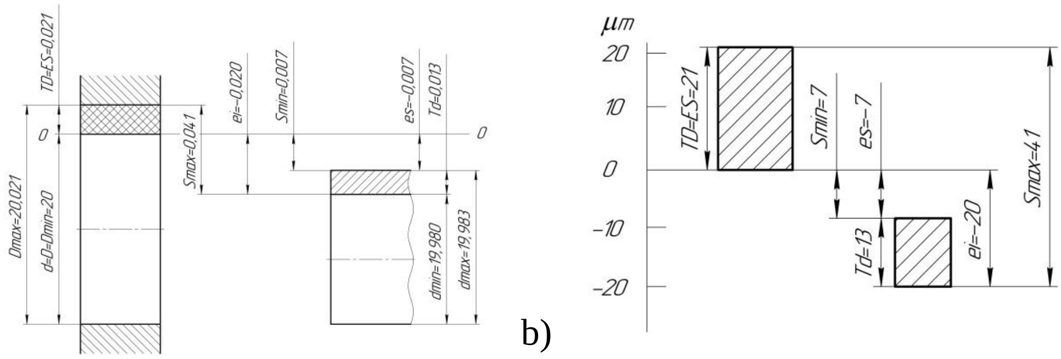


Figure 5 – Is a layout of tolerance limits of clearance fit

Example 6. Given the hole ∅ $20^{+0,021}$ mm, the shaft ∅ $20^{+0,048}_{+0,035}$ mm. Calculate the interference fit: determine the nominal and limit dimensions, the maximum and average deviations, the maximum interference, tolerances of the hole, shaft and fit. Construct a layout tolerance limits by limiting size and simplified one.

Solution. Determine the nominal size and deviation: $D = 20$ mm, $ES = 0{,}021\,mm = 21$ µm, $EI = 0$, $es = +0{,}048\,mm = +48$ µm, $ei = +0{,}035\,mm = +35$ µm, $E_m = 10{,}5$ µm, $e_m = 0{,}5(48 + 35) = 41{,}5$ µm.

The limit dimensions of the hole and shaft are determined mm: $D_{\max} = 20 + 0{,}021 = 20{,}021\,mm$; $D_{\min} = 20 + 0 = 20\,mm$; $d_{\max} = 20 + 0{,}048 = 20{,}048\,mm$; $d_{\min} = 20 + 0{,}035 = 20{,}035\,mm$.

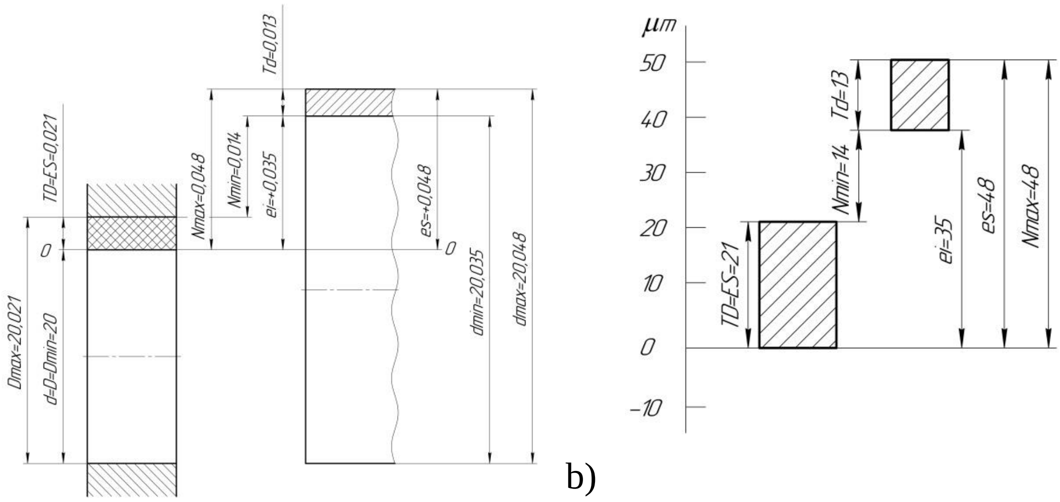
Interference limit is determined: maximum interference is $N_{\max} = 48 - 0 = 48$ µm; minimum interference is $N_{\min} = 35 - 21 = 14$ µm or by limit dimensions:

$$N_{\max} = 20{,}048 - 20 = 0{,}048\,mm = 48 \text{ µm};$$

$$N_{\min} = 20{,}035 - 20{,}021 = 0{,}014\,mm = 14 \text{ µm}.$$

Calculate the hole tolerances $TD = 21 - 0 = 21$ µm; the shaft tolerances $Td = 48 - 35 = 13$ µm; and the fit tolerances $TN = 21 + 13 = 34$ µm. The fit tolerance $TN = N_{\max} - N_{\min} = 48 - 14 = 34$ µm, i.e. it is equal to the tolerance clearance.

Build the layout of tolerance limits by limit dimensions (figure 6, a) and a simplified layout in scale (figure 6, b).



a) b)

Figure 6 – Is a layout of tolerance limits with interference fit

Example 7. Given the hole Ø $20^{+0,021}$ mm, shaft Ø $20^{+0,015}_{+0,002}$. mm. Calculate the transitional fit: determine the nominal and limit dimensions, the maximum and average deviations, the maximum interference and clearance, the tolerances of the hole, shaft and fit. Construct a layout tolerance limits by limiting size and simplified one.

Solution. Determine the nominal size and deviation: $D = 20$ mm, $ES = 0,021\,mm = 21$ µm, $EI = 0,$ $es = +0,015\,mm = +15$ µm, $ei = +0,002\,mm = +2$ µm, $E_m = 10,5$ µm, $e_m = 0,5(15+2) = 8,5$ µm.

The limit dimensions of the hole and shaft are determined: $D_{\max} = 20 + 0,021 = 20,021$ mm; $D_{\min} = 20 + 0 = 20$ mm; $d_{\max} = 20 + 0,015 = 20,015$ mm; $d_{\min} = 20 + 0,002 = 20,002$ mm.

As the negative interference is equal to the positive clearance and vice versa, the interference or clearance is calculated. Interference limit is calculated: maximum interference $N_{\max} = 15 - 0 = 15$ µm; minimum interference $N_{\min} = 2 - 21 = -19$ µm, i.e. $S_{\max} = 19$ µm.

As a check on, define the clearance limit: maximum clearance $S_{\max} = 21 - 2 = 19$ µm; minimum clearance $S_{\min} = 0 - (+15) = -15$ µm, i.e. result is $N_{\max} = 15$ µm.

Recommendation. *It is convenient to make a preliminarily drawing of layout of the tolerance fit limits, observing the ratio of the limit deviations, apply the given and required parameters to it, and then compose the design equations.*

This drawing (figure 7) clearly shows that the combination of the maximum shaft with minimum hole gives maximum interference $N_{\max}$, and the combination of the maximum hole with the minimum shaft gives the maximum clearance that are calculated.

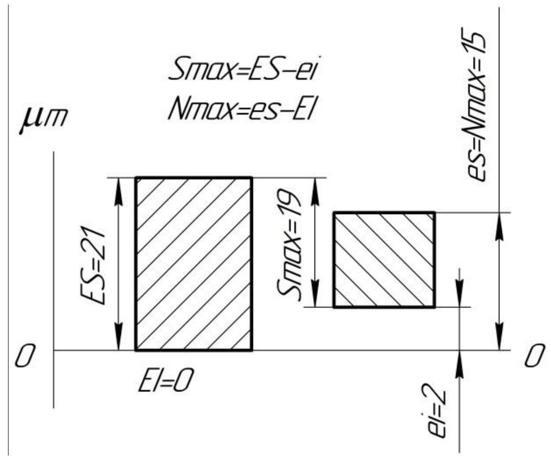


Figure 7 – Is the scheme for determining the maximum interference $N_{\max}$ and maximum clearance $S_{\max}$

Determine the tolerances of the hole, shaft, and fit: $TD = 21$ μm, $Td = 15 - 2 = 13$ μm, $TN = 34$ μm.

*A*s $-S_{\min} = +N_{\max}$, that the tolerance of transitional fit is equal to the sum of $S_{\max}$ and $N_{\max}$, i.e $TN = S_{\max} + N_{\max} = 19 + 15 = 34$ μm.

The layout of the tolerance limits by the limit dimensions is shown on figure 8, *a*; a simplified scheme is on figure 8, *b*.

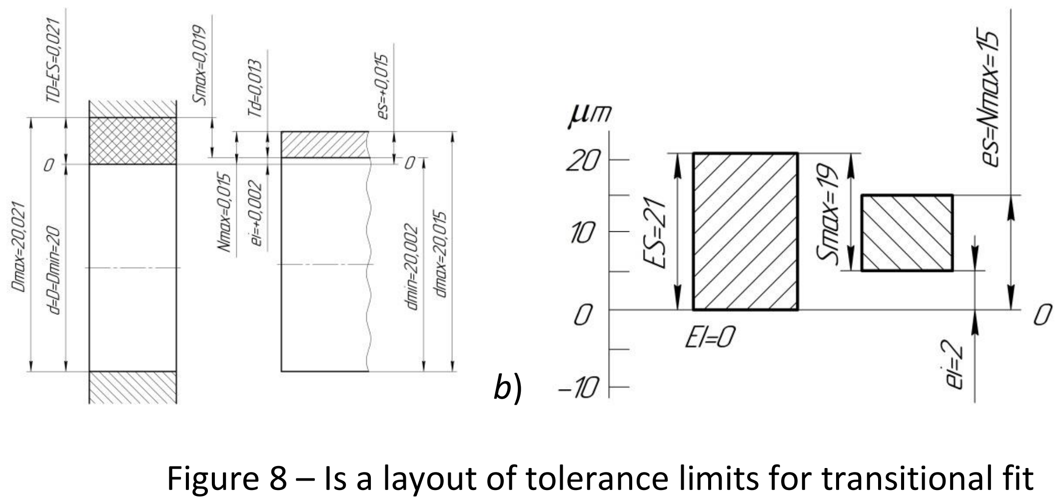
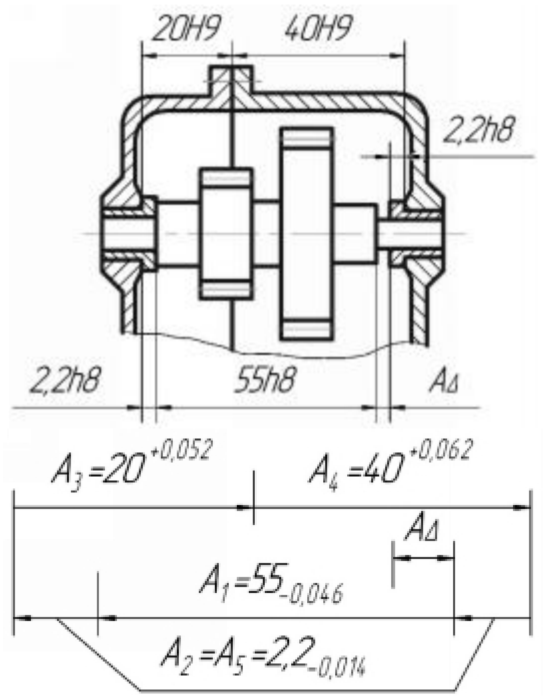


Figure 8 – Is a layout of tolerance limits for transitional fit

Example 1. The dimensions and tolerances of the stud bolt unit of the transfer mechanism are given (Figure 1, a). Determine the nominal value, tolerance and limit deviations, as well as the limiting dimensions of the closing link. Lead the calculation to complete interchangeability.



a)

b)

Figure 1 – is the stud bolt unit of the transfer mechanism

Solution. The links A_1 = 55h8, A_2 = A_5 = 2,2h8 are decreasing; the links A_3 = 20H9 and A_4 = 40H9 are increasing (on the links A_3 and A_4 tolerances are assigned from a coarser grade, taking into account the increased complexity of processing of body parts).

Tolerances of links are written out from the table of Annex E: TA_1 = 46 µm; TA_2 = TA_5 = 14 µm; TA_3 = 52 µm; TA_4 = 62 µm.

Limit deviations of links: ESA_1 = 0; EIA_1 = 46 µm; ESA_2 = ESA_5 = 0, EIA_2 = EIA_5 = - 14 µm; EIA_3 = 0; ESA_3 = 52 µm; EIA_4 = 62 µm. Construct the dimension chain (Figure 1, *b*).

Calculate by the nominal value of the closing link

$$A_\Delta = A_3 + A_4 - (A_1 + A_2 + A_5) = 20 + 40 - (2{,}2 + 55 + 2{,}2) = 0{,}6 \text{ mm.}$$

Tolerance of the closing link by

$$TA_\Delta = TA_1 + TA_2 + TA_3 + TA_4 + TA_5 = 46 + 14 + 52 + 62 + 14 = 188 \text{ µm.}$$

The limiting deviations of the closing link are determined by

$$ESA_{\Delta} = ESA_3 + ESA_4 - (EIA_1 + EIA_2 + EIA_5) = 52 + 62 - (-46 - 14 - 14) = 188 \ \mu\text{m};$$

$$EIA_{\Delta} = EIA_3 + EIA_4 - (ESA_1 + ESA_2 + ESA_5) = 0 \ .$$

Limit sizes of the closing link, if necessary, is convenient to determine by

$$A_{\Delta\max} = A_{\Delta} + ESA_{\Delta} = 0{,}6 + 0{,}188 = 0{,}788 \ \text{mm};$$

$$A_{\Delta\min} = A_{\Delta} + EIA_{\Delta} = 0{,}6 \ \text{mm}.$$

Example 2. The nominal dimensions and the initial size A_{Δ} = 0,6 ± 0,5 mm are specified (deviations are projected based on the operating conditions of the stud bolt of the transmission mechanism (Figure 2, *a*). Determine tolerances and limit deviations in dimensions.

Solution. Since the dimension chain consists of a relatively small number of links having different values, the calculation will be made to the maximum-minimum by the method of one grade tolerances. Draw up a diagram of the dimension chain (Figure 2, b), from which it can be seen that the links A_3 and A_4 are increasing, and the links A_1, A_2 and A_5 *are* decreasing.

Find the tolerances of the constituent links. Tolerance of closing link

$$TA_{\Delta} = ESA_{\Delta} - EIA_{\Delta} = 0{,}05 - (-0{,}05) = 0{,}1$$ mm, which corresponds to *IT*12.

From the table of Annex D, write out the numerical values of the tolerance units of the constituent elements A_1 = 55, i_1 = 1.86; $A_2 = A_5$ = 2,2, $i_2 = i_5$ = 0,55; A_3 = 20, A_3 = 1,31; A_4 = 40, i_4 = 1,56.

The number of tolerance units a is found by

$$a = \frac{100}{1{,}86 + 0{,}55 + 1{,}31 + 1{,}56 + 0{,}55} = 17{,}2 \approx IT = 16 \ .$$

On the constituent links, assign the tolerances for *IT*7: TA_1 = 30 μm; $TA_2 = TA_5$ = 10 μm; TA_3 = 21 μm; TA_4 = 25 μm.

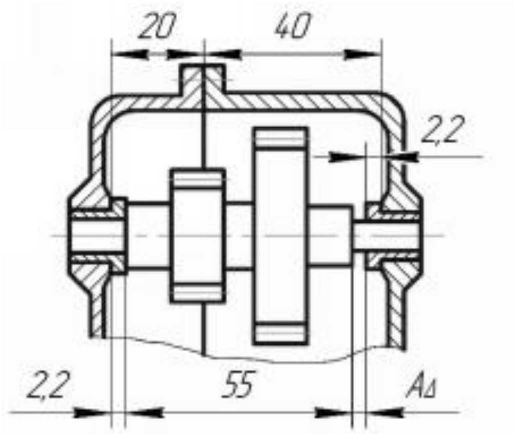
With these tolerances, the tolerance sum of the constituent links is not equal to the initial link tolerance

$$\sum_{j=1}^{m-1} TA_j = 96 < TA_{\Delta} = 100 \ .$$

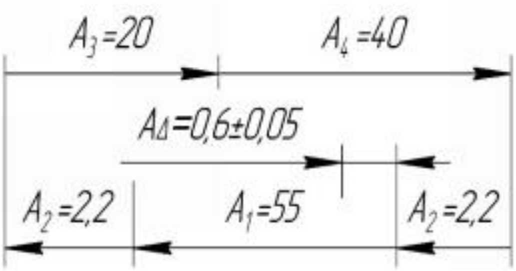
Therefore, take one link, for example A_2, tie down, for it calculate the tolerance by

$$TA_2 = 100 - (30 + 21 + 4 - 25 + 10) = 14 \text{ μm},$$

which corresponds to *IT8*.



a)



b)

Figure 2 – Is the stud bolt unit of the transfer mechanism

Note. In the example considered, the tolerance of the initial link, calculated from the sum of the tolerances of the component links, was slightly less than the specified tolerance. In such cases, it is possible to confine to approximate observance of the equality expressed by find the limiting deviations of the constituent links. Outline the tolerance limits *h7* for dimensions A_1 and A_5, *h8* for size A_2, J_s7 for the dimensions of the housing A_3 and A_4. Then the component dimensions with the limiting deviations will be as follows: $A_1 = 55_{-0,03}$ mm; $A_2 = 2{,}2_{-0,014}$ mm; $A_3 = 20 \pm 0{,}0105$ mm; $A_4 = 40 \pm 0{,}0125$ mm ; $A_5 = 2{,}2_{-0,01}$ mm .

The limiting deviations of the closing size are calculated by

$$ESA_\Delta = 0{,}0105 + 0{,}0125 - (-0{,}03 - 0{,}01 - 0{,}014) = 0{,}077 \text{ mm};$$

$$EIA_\Delta = -0{,}0105 - 0{,}0125 - 0 = -0{,}023 \text{ mm}.$$

As $ESA_\Delta - EIA_\Delta = TA_\Delta$, the calculation is correct. However, the tolerance limit of the initial link is located asymmetrically, as was specified in the condition of the problem. If it is necessary to comply with the stipulated condition, it will be necessary to assign all the dimensions of the tolerance limit to all components J_5**.**

The disadvantage of the performed calculation may be a rather high accuracy of the component links. Tolerances of these links can be increased by increasing the tolerance of the initial link (this is not always possible or desirable) or by calculating the tolerances in the probability method.

Example 3. Determine the tolerance of the closing link for the dimension chain (Figure 1, a) probability method.

Solution. Determination of tolerances and limiting deviations of the component links, as well as the nominal value of the closing link A_Δ, is described in the solution of Example 1. The scheme of the dimension chain is shown in Figure 1, *b*.

The tolerance of the closing link is found by

$$TA_\Delta = \sqrt{\mathrm{TA}_1^2 + \mathrm{TA}_2^2 + \mathrm{TA}_3^2 + \mathrm{TA}_4^2 + \mathrm{TA}_5^2} = \sqrt{46^2 + 14^2 + 52^2 + 62^2 + 14^2} = 97 \text{ μm},$$

which corresponds to *IT*12.

To determine the mean deviations of the constituent links, make the formula using the figure 1.

$$E_c A_j = 0{,}5(ESA_j + EIA_j).$$

From here $E_c A_1 = -23$ μm; $E_c A_2 = E_c A_5 = -7$ μm; $E_c A_3 = 26$ μm ; $E_c A_4 = 31$ m.

The mean deviation of the closing link is calculated by

$$E_c A_\Delta = E_c A_3 + E_c A_4 - (E_c A_1 + E_c A_2 + E_c A_5) = 26 + 31 - (-23 - 7 - 7) = 94 \text{ μm}.$$

Find the limiting deviations of the closing link according to the formulas

$$ESA_\Delta = 94 + 0{,}5 \times 100 = 144 \text{ μm;}$$

$$EIA_\Delta = 94 - 0{,}5 \times 100 = 44 \text{ μm}.$$

Then $A_\Delta = 0{,}6^{+0{,}144}_{+0{,}044}$.

The application of the probability method for calculating the dimension chain made it possible to increase the accuracy of the closing link by reducing the tolerance by almost 2 times.

Example 4. Determine the tolerances and limiting deviations of the dimensions of the unit (Figure 2, a) by the probability method. The initial size A_Δ = 0,6; TA_Δ = 100 μm.

Solution. Compose the scheme of the dimension chain (Figure 1, *b*). From it establish that the links A_3 and A_4 are increasing, and the links A_1, A_2 and A_5 are decreasing. Calculation is made by the method of one grade tolerances.

Find the tolerances of the constituent links. From the table of Annex D, write out the numerical values of the tolerance units of the constituent links; calculate the number of tolerance units *a* for a given dimension chain by formula

$$a = \frac{100}{\sqrt{1{,}86^2 + 0{,}55^2 + 1{,}31^2 + 1{,}56^2 + 0{,}55^2}} = 35{,}2$$

The value of *a* is between the eighth and ninth grades: *a* = 25 for *IT8; a = 40* for *IT9.* Tolerances for all links will be assigned preliminary to the nearest larger value of *a*: TA_1 = 74; $TA_2 = TA_5$ = 25; TA_3 = 52; TA_5 = 62 μm.

With a probability calculation method, the relationship between the initial link tolerance and the constituent link tolerance is expressed by the formula. According to this formula, the tolerances of the constituent links are corrected, if they are assigned according to the grade, in which *a* is not equal to the calculated one. For this, represent the formula in the following form: $TA_\Delta^2 = TA_1^2 + TA_2^2 + TA_3^2 + TA_4^2 + TA_5^2$.

Link A_1 will be tied down, then its tolerance

$$TA_1 = \sqrt{TA_\Delta^2 - [TA_2^2 + TA_3^2 + TA_4^2 + TA_5^2]} = \sqrt{100^2 - [25^2 + 52^2 + 62^2 + 25^2]} \approx 48 \text{ μm},$$

which is approximately equal to IT8 = 46 μm.

Find the mean and limiting deviations of the initial link. For the enlarging links, designate the tolerance limits H9 (Es = + TD), EI = 0); for decreasing A_2 and A_5 h9, for A_1 h8 (es = 0, ei = -Td).

The mean deviation of the constituent links is determined by the formula

$$E_cA_1 = -23 \text{ μm}; \quad E_cA_2 = -12{,}5 \text{ μm}; \quad E_cA_3 = 26 \text{ μm}; \quad E_cA_4 = 31 \text{ μm}.$$

The mean deviation of the initial link is calculated from formula

$$E_cA_\Delta = 26 + 31 - (-23 - 12{,}5 - 12{,}5) = 105 \text{ μm}.$$

The limiting deviations of the initial link can be found by formulas

$$ESA_\Delta = 105 + 0{,}5 \times 100 = 155 \text{ μm};$$

$$EIA_{\Delta} = 105 - 0{,}5 \times 100 = 55 \ \mu\text{m}.$$

Obtain $A_{\Delta} = 0{,}6^{+0{,}115}_{+0{,}055}$.

The application of a probability calculation method with the same allowance for the initial link allowed the application of the ninth grade for processing the constituent links instead of the seventh.

For the initial link, by the condition of the problem, it was required to assign a symmetric tolerance limit, for example $A_{\Delta} = 0{,}6 \pm 0{,}05$ mm. In order to withstand this condition, it is necessary to assign the tolerance limits J_s9 for the increasing ones, j_s9 for the decreasing ones, and j_s8 for the dependent link A_1.

Example 5. For turned part machining the shaft (Figure 2, *a*), a dimension of $\varnothing$ $80.5_{-0.12}$ mm is set .For the final processing of the shaft, the size is set by grinding $\varnothing 80_{-0.74}$ mm. Determine the depth of milling of the transverse slot, if the slot is milled before grinding, and its final depth is set equal to $8^{+0.15}$ mm.

Solution. Compose a dimension chain, guided by the following considerations: the shaft is completely processed in the centers; and the form deviation can be ignored; since the slot is milled after grinding, then 'the given size of the slot depth is obtained last, in fact, this dimension is the closing link of the composed dimension chain; in the problem the measuring base is replaced, at the final machined shaft the slot depth is measured from point *b* (Figure 3, a) in the problem such a base is the point *a*. In the dimension chain it is necessary to introduce an additional link A_3, which is the desired quantity.

Begin the construction of the dimension chain (Figure 3, *b*) from the point *b*. Pit – the constituent links in the direction of the arrows. The increasing links are the links $A_1 = 80_{-0.74}/2 = 40_{-0.37\ and}$ A_3; reducing the link $A_2 = 80{,}5_{-0.12}/2 = 40.25_{-0.06}$; the closing link $A_{\Delta} = 8^{+0.15}$.

Nominal size of the link A_3 find by the formula

$$A_3 = A_{\Delta} - A_1 + A_2 = 8 - 40 + 40{,}25 = 8{,}25 \ \text{mm}.$$

Limit sizes $A_{3\max}$ and $A_{3\min}$ is determined by transforming the formulas

$$A_{3\max} = A_{\Delta\max} - A_{1\max} + A_{2\min} = 8{,}15 - 40 + 40{,}19 = 8{,}34 \ \text{mm};$$

$$A_{3\min} = A_{\Delta\min} - A_{1\min} + A_{2\max\,in} = 8 - 39{,}963 + 40{,}25 = 8{,}287 \ \text{mm}.$$

Limit deviation of size A_3 is convenient to calculate by formulas, mm

$$ESA_3 = A_{3\max} - A_3 = 8{,}34 - 8{,}25 = 0{,}09 \ \mu\text{m};$$

$$EIA_3 = A_{3\min} - A_3 = 8{,}287 - 8{,}25 = 0{,}037 \ \mu\text{m}.$$

Tolerance of the link A_3:

$$TA_3 = ESA_3 - EIA_3 = A_{3\max} - A_{3\min} = 0{,}053 \text{ mm.}$$

Obtain $A_3 = 8{,}25^{+0{,}090}_{+0{,}037}$.

Calculate ESA_3, EIA_3, TA_3 by the formulas

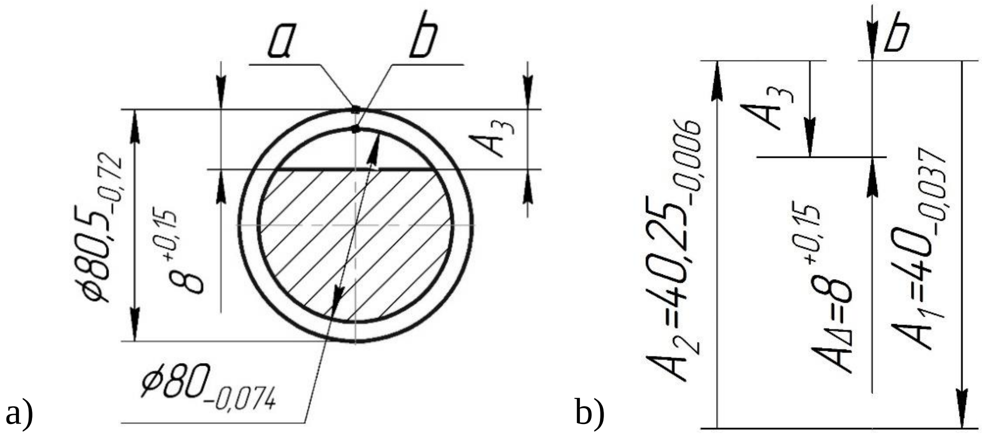
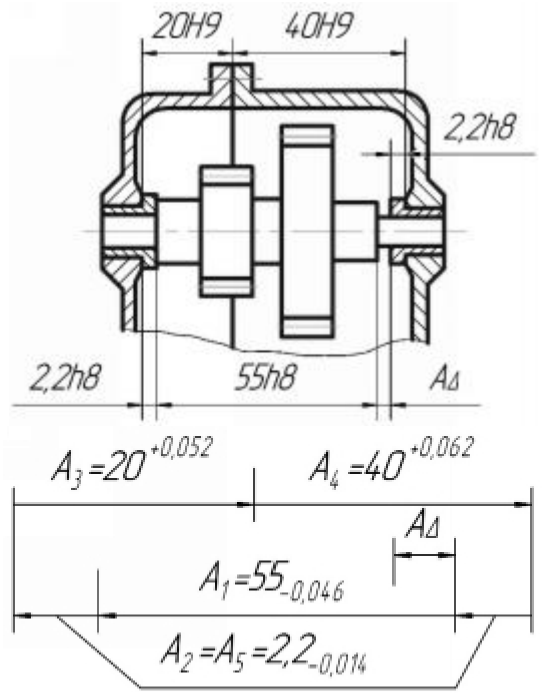


Figure 3 – Is a shaft

Example 1. The dimensions and tolerances of the stud bolt unit of the transfer mechanism are given (Figure 1, a). Determine the nominal value, tolerance and limit deviations, as well as the limiting dimensions of the closing link. Lead the calculation to complete interchangeability.



a)

b)

Figure 1 – is the stud bolt unit of the transfer mechanism

Solution. The links A_1 = 55h8, A_2 = A_5 = 2,2h8 are decreasing; the links A_3 = 20H9 and A_4 = 40H9 are increasing (on the links A_3 and A_4 tolerances are assigned from a coarser grade, taking into account the increased complexity of processing of body parts).

Tolerances of links are written out from the table of Annex E: TA_1 = 46 μm; $TA_2 = TA_5$ = 14 μm; TA_3 = 52 μm; TA_4 = 62 μm.

Limit deviations of links: ESA_1 = 0; EIA_1 = 46 μm; $ESA_2 = ESA_5$ = 0, $EIA_2 = EIA_5$ = - 14 μm; EIA_3 = 0; ESA_3 = 52 μm; EIA_4 = 62 μm. Construct the dimension chain (Figure 1, *b*).

Calculate by the nominal value of the closing link

$$A_\Delta = A_3 + A_4 - (A_1 + A_2 + A_5) = 20 + 40 - (2{,}2 + 55 + 2{,}2) = 0{,}6 \text{ mm.}$$

Tolerance of the closing link by

$$TA_\Delta = TA_1 + TA_2 + TA_3 + TA_4 + TA_5 = 46 + 14 + 52 + 62 + 14 = 188 \text{ μm.}$$

The limiting deviations of the closing link are determined by

$$ESA_\Delta = ESA_3 + ESA_4 - (EIA_1 + EIA_2 + EIA_5) = 52 + 62 - (-46 - 14 - 14) = 188 \text{ µm;}$$

$$EIA_\Delta = EIA_3 + EIA_4 - (ESA_1 + ESA_2 + ESA_5) = 0 .$$

Limit sizes of the closing link, if necessary, is convenient to determine by

$$A_{\Delta\max} = A_\Delta + ESA_\Delta = 0{,}6 + 0{,}188 = 0{,}788 \text{ mm;}$$

$$A_{\Delta\min} = A_\Delta + EIA_\Delta = 0{,}6 \text{ mm.}$$

Example 2. The nominal dimensions and the initial size $A_\Delta = 0{,}6 \pm 0{,}5$ mm are specified (deviations are projected based on the operating conditions of the stud bolt of the transmission mechanism (Figure 2, *a*). Determine tolerances and limit deviations in dimensions.

Solution. Since the dimension chain consists of a relatively small number of links having different values, the calculation will be made to the maximum-minimum by the method of one grade tolerances. Draw up a diagram of the dimension chain (Figure 2, b), from which it can be seen that the links A_3 and A_4 are increasing, and the links A_1, A_2 and A_5 *are* decreasing.

Find the tolerances of the constituent links. Tolerance of closing link

$$TA_\Delta = ESA_\Delta - EIA_\Delta = 0{,}05 - (-0{,}05) = 0{,}1 \text{ mm, which corresponds to } IT12.$$

From the table of Annex D, write out the numerical values of the tolerance units of the constituent elements A_1 = 55, i_1 = 1.86; $A_2 = A_5$ = 2,2, $i_2 = i_5$ = 0,55; A_3 = 20, A_3 = 1,31; A_4 = 40, i_4 = 1,56.

The number of tolerance units a is found by

$$a = \frac{100}{1{,}86 + 0{,}55 + 1{,}31 + 1{,}56 + 0{,}55} = 17{,}2 \approx IT = 16 .$$

On the constituent links, assign the tolerances for *IT*7: TA_1 = 30 µm; $TA_2 = TA_5$ = 10 µm; TA_3 = 21 µm; TA_4 = 25 µm.

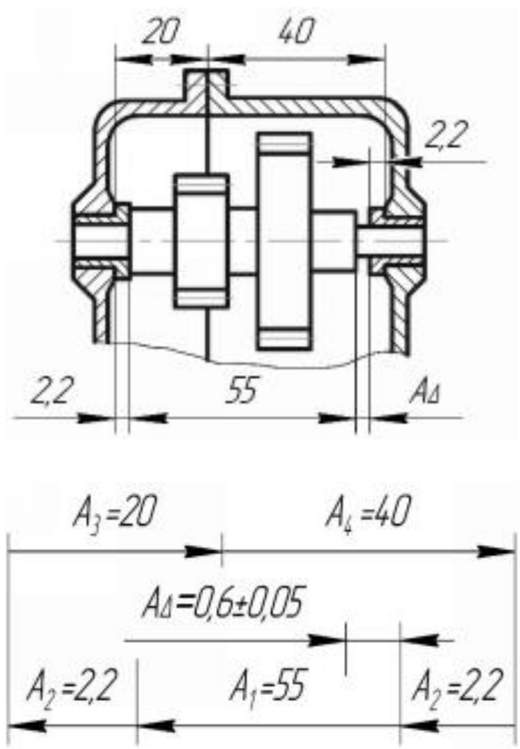
With these tolerances, the tolerance sum of the constituent links is not equal to the initial link tolerance

$$\sum_{j=1}^{m-1} TA_j = 96 < TA_\Delta = 100 .$$

Therefore, take one link, for example A_2, tie down, for it calculate the tolerance by

$$TA_2 = 100 - (30 + 21 + 4 - 25 + 10) = 14 \text{ µm},$$

which corresponds to *IT8*.



a)

b)

Figure 2 – Is the stud bolt unit of the transfer mechanism

Note. In the example considered, the tolerance of the initial link, calculated from the sum of the tolerances of the component links, was slightly less than the specified tolerance. In such cases, it is possible to confine to approximate observance of the equality expressed by find the limiting deviations of the constituent links. Outline the tolerance limits *h7* for dimensions A_1 and A_5, *h8* for size A_2, J_s7 for the dimensions of the housing A_3 and A_4. Then the component dimensions with the limiting deviations will be as follows: $A_1 = 55_{-0,03}$ mm; $A_2 = 2{,}2_{-0,014}$ mm; $A_3 = 20 \pm 0{,}0105$ mm; $A_4 = 40 \pm 0{,}0125$ mm ; $A_5 = 2{,}2_{-0,01}$ mm .

The limiting deviations of the closing size are calculated by

$$ESA_\Delta = 0{,}0105 + 0{,}0125 - (-0{,}03 - 0{,}01 - 0{,}014) = 0{,}077 \text{ mm;}$$

$$EIA_\Delta = -0{,}0105 - 0{,}0125 - 0 = -0{,}023 \text{ mm.}$$

As $ESA_\Delta - EIA_\Delta = TA_\Delta$, the calculation is correct. However, the tolerance limit of the initial link is located asymmetrically, as was specified in the condition of the problem. If it is necessary to comply with the stipulated condition, it will be necessary to assign all the dimensions of the tolerance limit to all components J_5**.**

The disadvantage of the performed calculation may be a rather high accuracy of the component links. Tolerances of these links can be increased by increasing the tolerance of the initial link (this is not always possible or desirable) or by calculating the tolerances in the probability method.

Example 3. Determine the tolerance of the closing link for the dimension chain (Figure 1, a) probability method.

Solution. Determination of tolerances and limiting deviations of the component links, as well as the nominal value of the closing link $A_{\Delta,}$ is described in the solution of Example 1. The scheme of the dimension chain is shown in Figure 1, *b*.

The tolerance of the closing link is found by

$$TA_\Delta = \sqrt{\mathrm{TA}_1^2 + \mathrm{TA}_2^2 + \mathrm{TA}_3^2 + \mathrm{TA}_4^2 + \mathrm{TA}_5^2} = \sqrt{46^2 + 14^2 + 52^2 + 62^2 + 14^2} = 97 \ \mu\text{m},$$

which corresponds to *IT*12.

To determine the mean deviations of the constituent links, make the formula using the figure 1.

$$E_c A_j = 0{,}5(ESA_j + EIA_j).$$

From here $E_cA_1 = -23$ μm; $E_cA_2 = E_cA_5 = -7$ μm; $E_cA_3 = 26$ μm ; $E_cA_4 = 31$ m.
The mean deviation of the closing link is calculated by

$$E_cA_\Delta = E_cA_3 + E_cA_4 - (E_cA_1 + E_cA_2 + E_cA_5) = 26 + 31 - (-23 - 7 - 7) = 94 \ \mu\text{m}.$$

Find the limiting deviations of the closing link according to the formulas

$$ESA_\Delta = 94 + 0{,}5 \times 100 = 144 \ \mu\text{m};$$

$$EIA_\Delta = 94 - 0{,}5 \times 100 = 44 \ \mu\text{m}.$$

Then $A_\Delta = 0{,}6^{+0{,}144}_{+0{,}044}$.

The application of the probability method for calculating the dimension chain made it possible to increase the accuracy of the closing link by reducing the tolerance by almost 2 times.

Example 4. Determine the tolerances and limiting deviations of the dimensions of the unit (Figure 2, a) by the probability method. The initial size A_Δ = 0,6; TA_Δ = 100 μm.

Solution. Compose the scheme of the dimension chain (Figure 1, *b*). From it establish that the links A_3 and A_4 are increasing, and the links A_1, A_2 and A_5 are decreasing. Calculation is made by the method of one grade tolerances.

Find the tolerances of the constituent links. From the table of Annex D, write out the numerical values of the tolerance units of the constituent links; calculate the number of tolerance units *a* for a given dimension chain by formula

$$a = \frac{100}{\sqrt{1{,}86^2 + 0{,}55^2 + 1{,}31^2 + 1{,}56^2 + 0{,}55^2}} = 35{,}2$$

The value of *a* is between the eighth and ninth grades: *a* = 25 for *IT8; a = 40* for *IT9.* Tolerances for all links will be assigned preliminary to the nearest larger value of *a*: TA_1 = 74; TA_2 = TA_5 = 25; TA_3 = 52; TA_5 = 62 μm.

With a probability calculation method, the relationship between the initial link tolerance and the constituent link tolerance is expressed by the formula. According to this formula, the tolerances of the constituent links are corrected, if they are assigned according to the grade, in which *a* is not equal to the calculated one. For this, represent the formula in the following form: $TA_\Delta^2 = TA_1^2 + TA_2^2 + TA_3^2 + TA_4^2 + TA_5^2$.

Link A_1 will be tied down, then its tolerance

$$TA_1 = \sqrt{TA_\Delta^2 - [TA_2^2 + TA_3^2 + TA_4^2 + TA_5^2]} = \sqrt{100^2 - [25^2 + 52^2 + 62^2 + 25^2]} \approx 48 \text{ μm},$$

which is approximately equal to IT8 = 46 μm.

Find the mean and limiting deviations of the initial link. For the enlarging links, designate the tolerance limits H9 (Es = + TD), EI = 0); for decreasing A_2 and A_5 h9, for A_1 h8 (es = 0, ei = -Td).

The mean deviation of the constituent links is determined by the formula

$$E_cA_1 = -23 \text{ μm}; \; E_cA_2 = -12{,}5 \text{ μm}; \; E_cA_3 = 26 \text{ μm}; \; E_cA_4 = 31 \text{ μm}.$$

The mean deviation of the initial link is calculated from formula

$$E_cA_\Delta = 26 + 31 - (-23 - 12{,}5 - 12{,}5) = 105 \text{ μm}.$$

The limiting deviations of the initial link can be found by formulas

$$ESA_\Delta = 105 + 0{,}5 \times 100 = 155 \text{ μm};$$

$$EIA_{\Delta} = 105 - 0{,}5 \times 100 = 55 \ \mu\text{m}.$$

Obtain $A_{\Delta} = 0{,}6^{+0{,}115}_{+0{,}055}$.

The application of a probability calculation method with the same allowance for the initial link allowed the application of the ninth grade for processing the constituent links instead of the seventh.

For the initial link, by the condition of the problem, it was required to assign a symmetric tolerance limit, for example $A_{\Delta} = 0{,}6 \pm 0{,}05$ mm. In order to withstand this condition, it is necessary to assign the tolerance limits J_s9 for the increasing ones, j_s9 for the decreasing ones, and j_s8 for the dependent link A_1.

Example 5. For turned part machining the shaft (Figure 2, *a*), a dimension of $\varnothing$ $80.5_{-0.12}$ mm is set .For the final processing of the shaft, the size is set by grinding $\varnothing 80_{-0.74}$ mm. Determine the depth of milling of the transverse slot, if the slot is milled before grinding, and its final depth is set equal to $8^{+0.15}$ mm.

Solution. Compose a dimension chain, guided by the following considerations: the shaft is completely processed in the centers; and the form deviation can be ignored; since the slot is milled after grinding, then 'the given size of the slot depth is obtained last, in fact, this dimension is the closing link of the composed dimension chain; in the problem the measuring base is replaced, at the final machined shaft the slot depth is measured from point *b* (Figure 3, a) in the problem such a base is the point *a*. In the dimension chain it is necessary to introduce an additional link A_3, which is the desired quantity.

Begin the construction of the dimension chain (Figure 3, *b*) from the point *b*. Pit – the constituent links in the direction of the arrows. The increasing links are the links $A_1 = 80_{-0.74}/2 = 40_{-0.37 \ and}$ A_3; reducing the link $A_2 = 80{,}5_{-0.12}/2 = 40.25_{-0.06}$; the closing link $A_{\Delta} = 8^{+0.15}$.

Nominal size of the link A_3 find by the formula

$$A_3 = A_{\Delta} - A_1 + A_2 = 8 - 40 + 40{,}25 = 8{,}25 \ \text{mm}.$$

Limit sizes $A_{3\max}$ and $A_{3\min}$ is determined by transforming the formulas

$$A_{3\max} = A_{\Delta\max} - A_{1\max} + A_{2\min} = 8{,}15 - 40 + 40{,}19 = 8{,}34 \ \text{mm};$$

$$A_{3\min} = A_{\Delta\min} - A_{1\min} + A_{2\max\, in} = 8 - 39{,}963 + 40{,}25 = 8{,}287 \ \text{mm}.$$

Limit deviation of size A_3 is convenient to calculate by formulas, mm

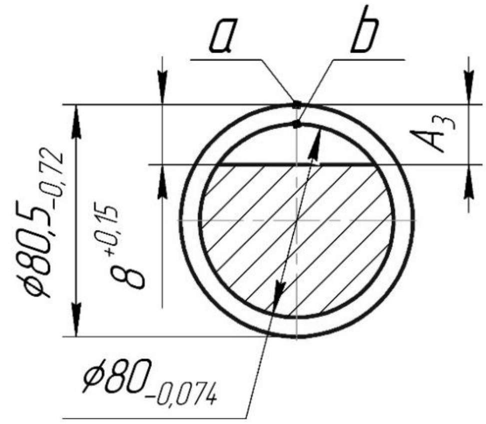
$$ESA_3 = A_{3\max} - A_3 = 8{,}34 - 8{,}25 = 0{,}09 \ \mu\text{m};$$

$$EIA_3 = A_{3\min} - A_3 = 8{,}287 - 8{,}25 = 0{,}037 \ \mu\text{m}.$$

Tolerance of the link A_3:

$$TA_3 = ESA_3 - EIA_3 = A_{3\max} - A_{3\min} = 0{,}053 \text{ mm.}$$

Obtain $A_3 = 8{,}25^{+0{,}090}_{+0{,}037}$.
Calculate ESA_3, EIA_3, TA_3 by the formulas

a) 

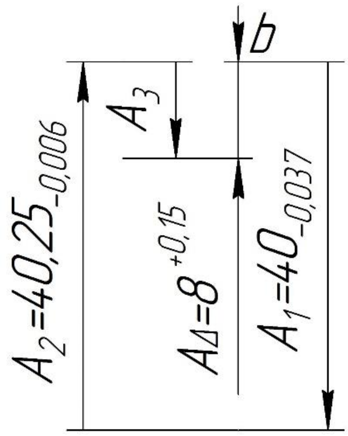
b) 

Figure 3 – Is a shaft

Example1. Determine the required and limiting dimensions of gap gauge to monitor the *Ø48d9* shaft and control gauges to it. Draw a diagram of the tolerances limits of these gauges and their sketches.

Solution. Define the limiting deviations, tolerance and limiting dimensions of the *Ø45d9* shaft: es = -0,080 mm; ei = -0,142 mm; Td= 0,062 mm; d_{max} = 44,920 mm; d_{min} = 44,858 mm.

Table 1 – Is formulas for limiting dimensions of limit gauges

Type of the gauge		Nominal dimensions of the item, mm					
		up to 180		from 180 to 500		from 500	
		nominal size of the gauge				gauge tolerances	
		functional	control	functional	control	functional	control
Plug gauge	Go-on new	$D_{min}+Z$	-	$D_{min}+Z_1$	-	$\pm H(H_s)$	-
	Go-on worn	$D_{min}-Y$	-	$D_{min}-Y+\alpha$	-	-	-
	Not go	D_{max}	-	$D_{max}-\alpha$	-	$\pm H(H_s)$	-
Gap gauge	Go-on new	$D_{max}-Z_1$				$\pm 0,5H_1$	-
	Go-on worn	$D_{max}+Y_1$		$D_{max}+Y_1-\alpha_1$		-	$\pm 0,5H_p$
	Not go	D_{min}		$D_{min}+\alpha_1$		$\pm 0,5H_1$	-

Limit deviations and tolerances of gauges are found from Annex L: Z_l = 11 µm, H_l = 7 µm, У_l= 0 µm, H_p = 25 µm. Then, according to Figure 1, construct the scheme of tolerances limits of the controlled shaft, go and no go sides of the gap gauge and control gauges (Figure 1, *a, b*).

Using the formulas of Table 1, calculate the required dimensions of the gap gauge.

Go-on of new side

$$F-Go_{used}=44{,}920-0{,}011-0{,}5\times 0{,}007=44{,}9055^{+0{,}007}\text{ mm;}$$

No go side

$$F-No_{used}=44{,}858-0{,}5\times 0{,}007=44{,}8545^{+0{,}007}\text{ mm;}$$

Go-on worn

$$F-Goworn=44{,}920+0=44{,}920\text{ mm.}$$

Using the formulas in Table 4.1, find the required dimensions of the control gauge.

To control the go-on new side

$$C - Go_{used} = 44{,}920 - 0{,}011 - 0{,}5 \times 0{,}0025 = 44{,}91025_{-0{,}0025} \text{ mm;}$$

To control the no go-on side

$$C - No_{used} = 44{,}858 + 0{,}5 \times 0{,}0025 = 44{,}85925_{-0{,}0025} \text{ mm;}$$

To control the wear of the go-on side

$$C - W_{used} = 44{,}920 - 0 - 0{,}5 \times 0{,}0025 = 44{,}921_{-0{,}0025} \text{ mm.}$$

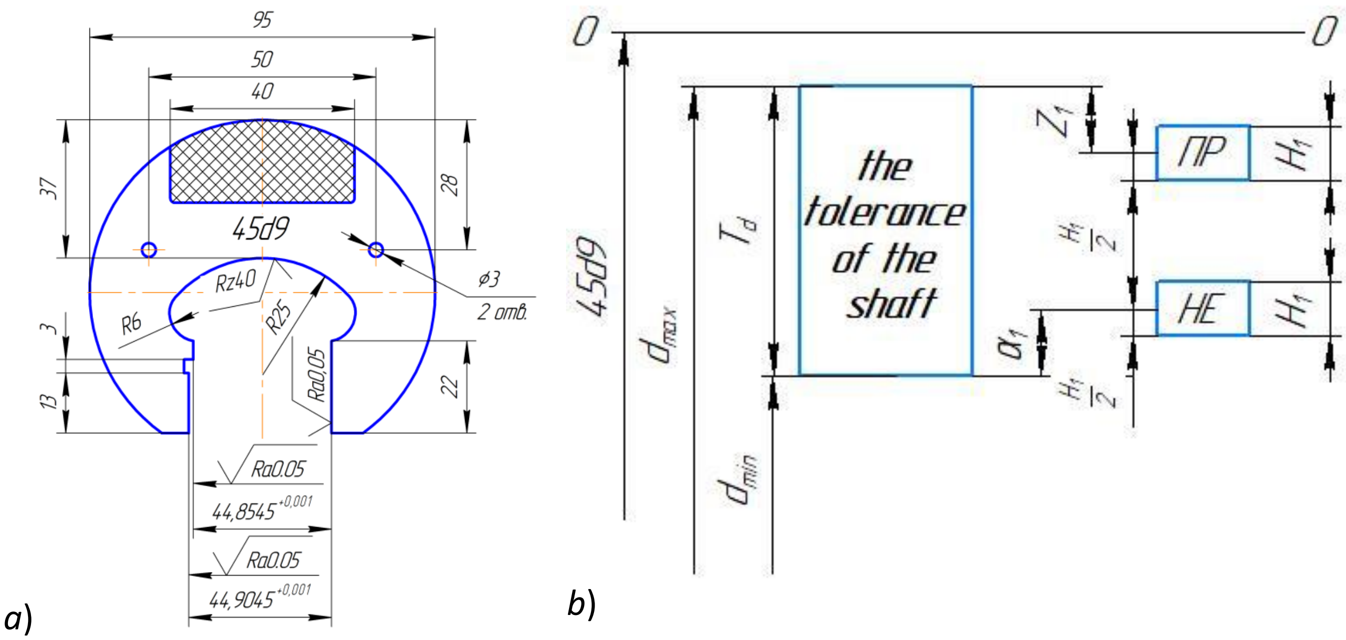


Figure 1 – Is a gap gauge and the layout of the tolerance limits

Determine limiting size of the gauges by the formulas in Table 2 and accumulate them in the table.

Side of gauge	Functional gauge	Control gauge
Go-side	$F\text{-}GO_{max}$ = 44,9125 мм $F\text{-}GO_{min}$ = 44,9055 мм	$C\text{-}GO_{max}$ = 44,91025 мм $C\text{-}GO_{min}$ = 44,90775 мм
"No-go"-side	$F\text{-}NO_{max}$ = 44,8165 мм $F\text{-}NO_{min}$ = 44,8545 мм	$C\text{-}NO_{max}$ = 44,85925 мм $C\text{-}NO_{min}$ = 44,85675 мм
Go worn out-side	$F\text{-}GO_{worn}$ = 44,920 мм	$C\text{-}W_{max}$ = 44,92125 мм $C\text{-}W_{min}$ = 44,91875 мм

Example 2. Determine the required and limiting dimensions of the plug-gauge for monitoring the Ø45H7 hole. Draw a scheme of tolerance limits and a sketch of the gauge.

Solution. Determine the limiting deviations, tolerance and limiting dimensions of the Ø45H7 hole: EI = 0 mm; **TD =** 0,025 mm; **ES =** 0,025 mm; D_{max} = 45,025 mm; D_{min} = 45,0 mm.

Limit deviations and tolerances of the plug-gauge are found from Annex L: Z =3,5 µm, *Y*= 0 µm, *α* = 0 µm, *H* = 4 µm, H_p = 1,5 µm. Construct the scheme of tolerances limits of the plug-gauge (figure 2).

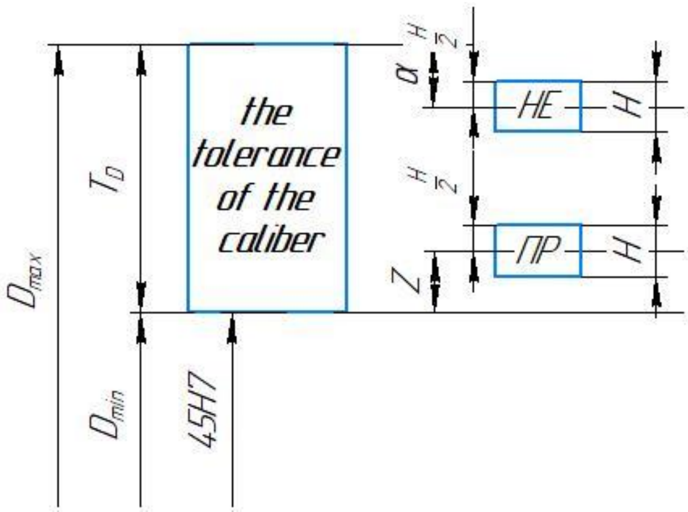


Figure 2 – Is the layout of the 45H7 tolerance limits

Using the formulas in Table 1, calculate the required dimensions of the plug gauge.

The go-on new side

$$F-GO_{used}=45{,}0+0{,}0035+0{,}5\times0{,}004=45{,}0055_{-0{,}004}\text{ mm}$$

The no go-on side

$$F-NO_{used}=45{,}025+0{,}5\times0{,}004=45{,}027_{-0{,}004}\text{ mm}$$

The go-on worn out

$$F-NO_{worn}=45{,}0-0=45{,}0\text{ mm}$$

Limiting sizes of the plug gauge: go-side $F\text{-}GO_{max}$ = 45,0055 mm, $F\text{-}GO_{min}$ = 45,0015 mm; "no-go" side $F\text{-}NO_{max}$ = 45,027 mm, $F\text{-}NO_{min}$ = 45,023 mm. The sketch of the plug gauge is shown in Figure 3.

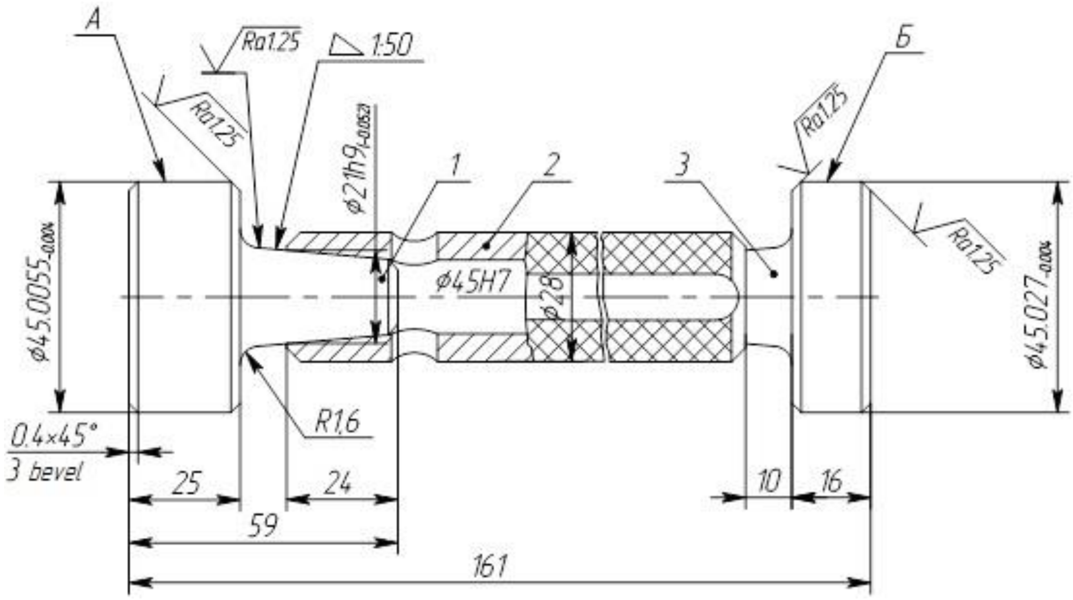


Figure 3 – Is the plug gauge

Example 1. When measuring, the following values of the inside diameter of the rolling bearing are obtained: lager d_{lg} = 99,998 mm, smaller d_{sm} = 99,976 mm. Determine the validity of the bearing ring if the nominal diameter d = 100 mm, accuracy class 0.

Solution. According to the table in Annex B, find the limiting deviations of the inside diameter d, mm: ES = 0,005, EI = -0,025, and also the mean diameter d_m , mm: ES = 0, EI = -0,020 mm.

Determine the maximum dimensions, mm: d_{max} = 100,005; d_{min} = 99,975; $d_{m\ max}$ = 100,000; $d_{m\ min}$ = 99,980.

To establish the validity of the ring, compare the measured values of d and the average value of d_m with the corresponding limiting dimensions, mm: d_{lg} = 99,998 < d_{max} = 100,005; d_{sm} = 99.976> d_{min} = 99,975; the actual value of the mean diameter d_m = 0.5 (99,998 + 99,976) = 99,987, is also within the acceptable limits.

Example 2. When measuring the inner ring of the bearing (nominal and limiting dimensions are given in Example 1), it turned out that d_{max} = 100,004 and d_{min} = 99,998 mm. Set the validity of the ring.

Solution. The measured values of d (mm) are within the acceptable limits: d_{lg} = 100,004 < d_{max} = 100,005; d_{sm} = 99,998> d_{min} = 99,975.

Find d_{ma} = 0,5 (100,004 + 99,998) = 100,001 mm; d_m > $d_{m\ max}$ = 100 mm. Consequently, the bearing ring is defective, despite the fact that the measured values of its inside diameter do not exceed the acceptable limits.

Example 3. To connect the outer ring of the rolling bearing to the body, the tolerance limit N7, D = 80 mm, class of bearing accuracy is 0. Compare the fit of the outer ring of the bearing in the body N7/hB, with a similar fit from the USTF N7/h6.

Note. hB is conventional designation of the tolerance limit for the outer bearing ring; ThB is tolerance of the same ring.

Solution. In accordance with GOST 25346-82, find the maximum deviations: for the tolerance limit N7 ES = -9 µm, TD = 30 µm and EI = -39 µm; for the tolerance limit h6 es = 0, ei = -Td = -19 µm; for the tolerance limit hB es = 0, ei = -ThB = -13 µm; for D_m (Annex M).

Based on the deviations found, draw the tolerance limits (Figure 1). Calculate the limiting interferences, clearances and tolerances of both fits (Table 1).

Both fits have very close parameters and provide approximately the same nature of the connection. However, the fit of the bearing ring in the housing is characterized by a slightly smaller uncertainty of interference and greater reliability of the connection.

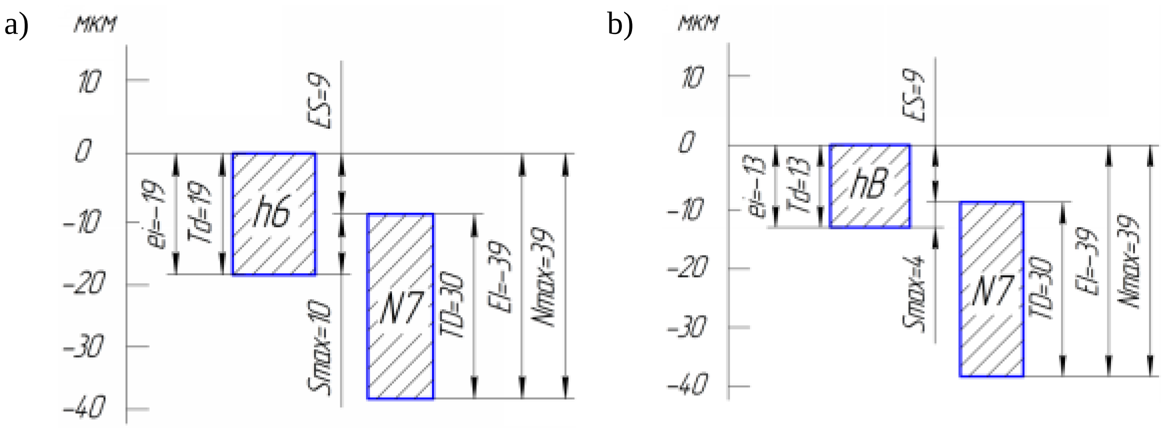


Figure 1 – Is a layout of tolerance limits

Table 1 – Is results of calculation

Quantity	USTF fit $N7/h6$, μm	Bearing fit in the body $N7/hB$, μm
$N_{\max} = es - TI$	$0-(-0.39)=39$	$0-(-39)=39$
$N_{\min} = ei - ES$	$-19-(-9)=-10==-S_{\max};S_{\max}=10$	$-13-(-9)=-4=-S_{\max}$
$N_m = 0{,}5(N_{\max}+N_{\min})$	$0{,}5[39+(-10)]=14{,}5$	$0{,}5[39+(-4)]=17{,}5$
$T\Pi = N_{\max} - N_{\min}$	$39-(-10)=49$	$39-(-4)=43$

Example 4. To connect the inner race of the rolling bearing, *d* = 40 mm and accuracy class is 0, the tolerance limit g6 applied to the shaft. Compare the fit of the inner bearing ring on the shaft KB/g6 with the H7/g6 fit.

Solution. In accordance with GOST 25346-82, find the limiting deviations: for the tolerance limit H7 EI = 0, ES = TD = 25 μm; for the tolerance limit g6 es = -9 μm, Td = 16 μm, ei = -25 μm, for the tolerance limit for the inner bearing ring KB (find from d_m in the annex table M) ES = 0 and EI = -TKB = -12 (TKB is inner ring tolerance).

Construct the schemes of tolerance limits (Figure 2) and calculate the limiting clearances and interferences and fit tolerances (Table 2).

Fit in the connection of the rolling bearing with the shaft was transitional with a very low probability of obtaining interference. Therefore, conditionally, both fits belong to clearance fits. However, the fits differ significantly: due to the position of the tolerance limit below the zero line and the smaller value of the TKB in the bearing fit on the shaft, the fit tolerance decreased by 1,5 times, the maximum clearance by 2 times and the average clearance by 2,7 times, which means that the accuracy reliability of the connection increased.

a) b)

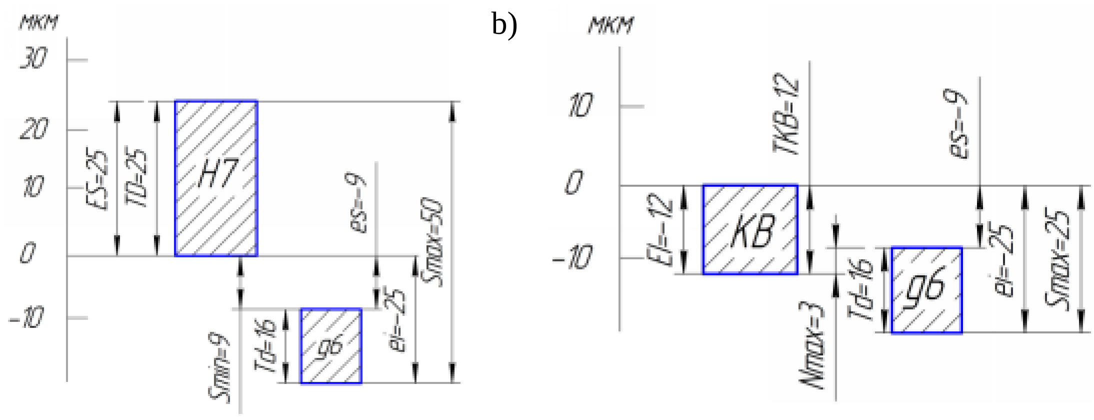


Figure 2 – Is a layout of tolerance limits

Table 2 – Is results of calculation

Calculation formulas	USTF fit $H7/g6$, μm	Bearing fit on shaft $KB/g6$, μm
$N_{\max} = ES - ei$	$25-(-25)=50$	$0-(-25)=25$
$N_{\min} = EI - es$	$0-(-9)=9$	$-12-(-96)=-3=-N_{\max}, N_{\max}=3$
$N_m = 0{,}5(S_{\max} + S_{\min})$	$0{,}5(50+9)=29{,}5$	$0{,}5[25+(-3)]=11$
$TП = S_{\max} - S_{\min}$	$50-9=41$	$25-(-3)=28$

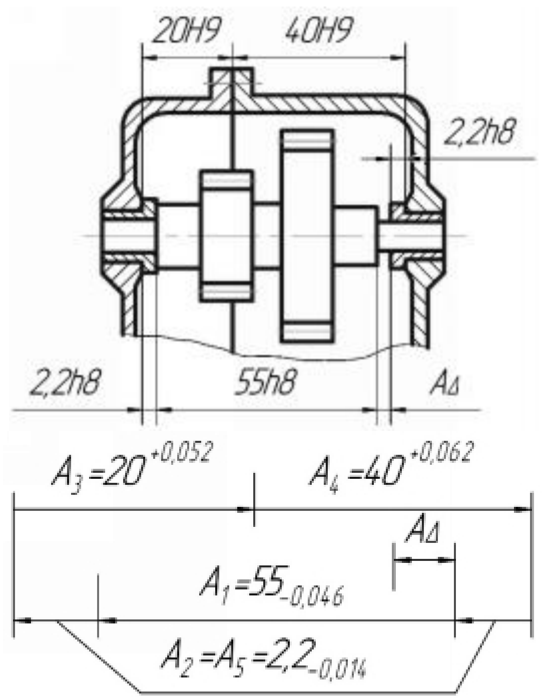
Example 5. Roller bearings of table rolls are under loads with strong shocks, overload up to 300%; radial reactions affect on the supports, constant in the direction, R = 10000 H, axial reaction A = 2000 H; inner rotary ring loading is circulation; inner fixed is local; 7510 tapered roller bearings with dimensions: d = 50 mm, D = 90 mm, B = 23 mm, r = 2 mm and r_1 = 0.8 mm, β = 12°; outside diameter of the bearing housing D_{house} = 125 mm; accuracy class of bearing is 0. Determine the fit for bearings.

Solution. Select the tolerance limit for the connection of the housing with the outer bearing ring undergoing a circulation loading. Find the values of the quantities entering into: b = 23 - (2 + 0,8) = 20,2 mm; k_1 = 1,8 (Annex R); to calculate k_2, find the ratio D/D_{house} = 90/125 ≈ 0,72, hence k_2 = 1,4 (Annex R); k_3 = 1 (Annex R). With these data, P_R = (10000 / 20,2) · 1.8 · 1,4 · 1 = 1250 H.

According to the table in Annex D, accept the tolerance limit *N7* having ES = -10 and EI = -45 μm.

To join the axes of the rollers of the table rolls with the inner bearing rings, accept the tolerance limits *h6* (Annex Q) for which es = 0, ei = -16 μm.

Example 1. The dimensions and tolerances of the stud bolt unit of the transfer mechanism are given (Figure 1, a). Determine the nominal value, tolerance and limit deviations, as well as the limiting dimensions of the closing link. Lead the calculation to complete interchangeability.



a)

b)

Figure 1 – is the stud bolt unit of the transfer mechanism

Solution. The links A_1 = 55h8, A_2 = A_5 = 2,2h8 are decreasing; the links A_3 = 20H9 and A_4 = 40H9 are increasing (on the links A_3 and A_4 tolerances are assigned from a coarser grade, taking into account the increased complexity of processing of body parts).

Tolerances of links are written out from the table of Annex E: TA_1 = 46 μm; TA_2 = TA_5 = 14 μm; TA_3 = 52 μm; TA_4 = 62 μm.

Limit deviations of links: ESA_1 = 0; EIA_1 = 46 μm; ESA_2 = ESA_5 = 0, EIA_2 = EIA_5 = - 14 μm; EIA_3 = 0; ESA_3 = 52 μm; EIA_4 = 62 μm. Construct the dimension chain (Figure 1, *b*).

Calculate by the nominal value of the closing link

$$A_\Delta = A_3 + A_4 - (A_1 + A_2 + A_5) = 20 + 40 - (2{,}2 + 55 + 2{,}2) = 0{,}6 \text{ mm.}$$

Tolerance of the closing link by

$$TA_\Delta = TA_1 + TA_2 + TA_3 + TA_4 + TA_5 = 46 + 14 + 52 + 62 + 14 = 188 \text{ μm.}$$

The limiting deviations of the closing link are determined by

$$ESA_\Delta = ESA_3 + ESA_4 - (EIA_1 + EIA_2 + EIA_5) = 52 + 62 - (-46 - 14 - 14) = 188 \ \mu\text{m};$$

$$EIA_\Delta = EIA_3 + EIA_4 - (ESA_1 + ESA_2 + ESA_5) = 0 \ .$$

Limit sizes of the closing link, if necessary, is convenient to determine by

$$A_{\Delta\max} = A_\Delta + ESA_\Delta = 0{,}6 + 0{,}188 = 0{,}788 \text{ mm};$$

$$A_{\Delta\min} = A_\Delta + EIA_\Delta = 0{,}6 \text{ mm}.$$

Example 2. The nominal dimensions and the initial size A_Δ = 0,6 ± 0,5 mm are specified (deviations are projected based on the operating conditions of the stud bolt of the transmission mechanism (Figure 2, *a*). Determine tolerances and limit deviations in dimensions.

Solution. Since the dimension chain consists of a relatively small number of links having different values, the calculation will be made to the maximum-minimum by the method of one grade tolerances. Draw up a diagram of the dimension chain (Figure 2, b), from which it can be seen that the links A_3 and A_4 are increasing, and the links A_1, A_2 and A_5 *are* decreasing.

Find the tolerances of the constituent links. Tolerance of closing link

$$TA_\Delta = ESA_\Delta - EIA_\Delta = 0{,}05 - (-0{,}05) = 0{,}1 \text{ mm, which corresponds to } IT12.$$

From the table of Annex D, write out the numerical values of the tolerance units of the constituent elements A_1 = 55, i_1 = 1.86; $A_2 = A_5$ = 2,2, $i_2 = i_5$ = 0,55; A_3 = 20, A_3 = 1,31; A_4 = 40, i_4 = 1,56.

The number of tolerance units a is found by

$$a = \frac{100}{1{,}86 + 0{,}55 + 1{,}31 + 1{,}56 + 0{,}55} = 17{,}2 \approx IT = 16 \ .$$

On the constituent links, assign the tolerances for *IT*7: TA_1 = 30 µm; $TA_2 = TA_5$ = 10 µm; TA_3 = 21 µm; TA_4 = 25 µm.

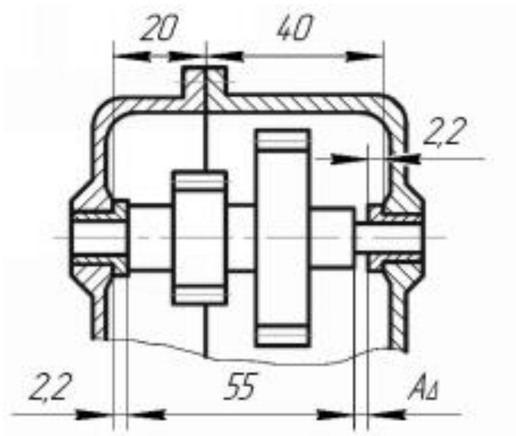
With these tolerances, the tolerance sum of the constituent links is not equal to the initial link tolerance

$$\sum_{j=1}^{m-1} TA_j = 96 < TA_\Delta = 100 \ .$$

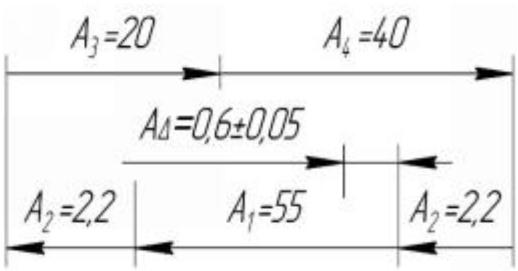
Therefore, take one link, for example A_2, tie down, for it calculate the tolerance by

$$TA_2 = 100 - (30 + 21 + 4 - 25 + 10) = 14 \ \mu\text{m},$$

which corresponds to *IT8*.



a)



b)

Figure 2 – Is the stud bolt unit of the transfer mechanism

Note. In the example considered, the tolerance of the initial link, calculated from the sum of the tolerances of the component links, was slightly less than the specified tolerance. In such cases, it is possible to confine to approximate observance of the equality expressed by find the limiting deviations of the constituent links. Outline the tolerance limits *h7* for dimensions A_1 and A_5, *h8* for size A_2, J_s7 for the dimensions of the housing A_3 and A_4. Then the component dimensions with the limiting deviations will be as follows: $A_1 = 55_{-0,03}$ mm; $A_2 = 2{,}2_{-0,014}$ mm; $A_3 = 20 \pm 0{,}0105$ mm; $A_4 = 40 \pm 0{,}0125$ mm ; $A_5 = 2{,}2_{-0,01}$ mm .

The limiting deviations of the closing size are calculated by

$$ESA_\Delta = 0{,}0105 + 0{,}0125 - (-0{,}03 - 0{,}01 - 0{,}014) = 0{,}077 \ \text{mm};$$

$$EIA_\Delta = -0{,}0105 - 0{,}0125 - 0 = -0{,}023 \ \text{mm}.$$

As $ESA_\Delta - EIA_\Delta = TA_\Delta$, the calculation is correct. However, the tolerance limit of the initial link is located asymmetrically, as was specified in the condition of the problem. If it is necessary to comply with the stipulated condition, it will be necessary to assign all the dimensions of the tolerance limit to all components J_5**.**

The disadvantage of the performed calculation may be a rather high accuracy of the component links. Tolerances of these links can be increased by increasing the tolerance of the initial link (this is not always possible or desirable) or by calculating the tolerances in the probability method.

Example 3. Determine the tolerance of the closing link for the dimension chain (Figure 1, a) probability method.

Solution. Determination of tolerances and limiting deviations of the component links, as well as the nominal value of the closing link A_Δ, is described in the solution of Example 1. The scheme of the dimension chain is shown in Figure 1, *b*.

The tolerance of the closing link is found by

$$TA_\Delta = \sqrt{\mathrm{TA}_1^2 + \mathrm{TA}_2^2 + \mathrm{TA}_3^2 + \mathrm{TA}_4^2 + \mathrm{TA}_5^2} = \sqrt{46^2 + 14^2 + 52^2 + 62^2 + 14^2} = 97 \ \mu\text{m},$$

which corresponds to *IT*12.

To determine the mean deviations of the constituent links, make the formula using the figure 1.

$$E_c A_j = 0{,}5(ESA_j + EIA_j).$$

From here $E_c A_1 = -23$ μm; $E_c A_2 = E_c A_5 = -7$ μm; $E_c A_3 = 26$ μm ; $E_c A_4 = 31$ m.

The mean deviation of the closing link is calculated by

$$E_c A_\Delta = E_c A_3 + E_c A_4 - (E_c A_1 + E_c A_2 + E_c A_5) = 26 + 31 - (-23 - 7 - 7) = 94 \ \mu\text{m}.$$

Find the limiting deviations of the closing link according to the formulas

$$ESA_\Delta = 94 + 0{,}5 \times 100 = 144 \ \mu\text{m};$$

$$EIA_\Delta = 94 - 0{,}5 \times 100 = 44 \ \mu\text{m}.$$

Then $A_\Delta = 0{,}6^{+0{,}144}_{+0{,}044}$.

The application of the probability method for calculating the dimension chain made it possible to increase the accuracy of the closing link by reducing the tolerance by almost 2 times.

Example 4. Determine the tolerances and limiting deviations of the dimensions of the unit (Figure 2, a) by the probability method. The initial size $A_Δ$ = 0,6; $TA_Δ$ = 100 μm.

Solution. Compose the scheme of the dimension chain (Figure 1, *b*). From it establish that the links A_3 and A_4 are increasing, and the links A_1, A_2 and A_5 are decreasing. Calculation is made by the method of one grade tolerances.

Find the tolerances of the constituent links. From the table of Annex D, write out the numerical values of the tolerance units of the constituent links; calculate the number of tolerance units *a* for a given dimension chain by formula

$$a = \frac{100}{\sqrt{1{,}86^2 + 0{,}55^2 + 1{,}31^2 + 1{,}56^2 + 0{,}55^2}} = 35{,}2$$

The value of *a* is between the eighth and ninth grades: *a* = 25 for *IT8; a = 40* for *IT9.* Tolerances for all links will be assigned preliminary to the nearest larger value of *a*: TA_1 = 74; $TA_2 = TA_5$ = 25; TA_3 = 52; TA_5 = 62 μm.

With a probability calculation method, the relationship between the initial link tolerance and the constituent link tolerance is expressed by the formula. According to this formula, the tolerances of the constituent links are corrected, if they are assigned according to the grade, in which *a* is not equal to the calculated one. For this, represent the formula in the following form: $TA_Δ^2 = TA_1^2 + TA_2^2 + TA_3^2 + TA_4^2 + TA_5^2$.

Link A_1 will be tied down, then its tolerance

$$TA_1 = \sqrt{TA_Δ^2 - [TA_2^2 + TA_3^2 + TA_4^2 + TA_5^2]} = \sqrt{100^2 - [25^2 + 52^2 + 62^2 + 25^2]} \approx 48 \text{ μm},$$

which is approximately equal to IT8 = 46 μm.

Find the mean and limiting deviations of the initial link. For the enlarging links, designate the tolerance limits H9 (Es = + TD), EI = 0); for decreasing A_2 and A_5 h9, for A_1 h8 (es = 0, ei = -Td).

The mean deviation of the constituent links is determined by the formula

$$E_cA_1 = -23 \text{ μm};\ E_cA_2 = -12{,}5 \text{ μm};\ E_cA_3 = 26 \text{ μm};\ E_cA_4 = 31 \text{ μm}.$$

The mean deviation of the initial link is calculated from formula

$$E_cA_Δ = 26 + 31 - (-23 - 12{,}5 - 12{,}5) = 105 \text{ μm}.$$

The limiting deviations of the initial link can be found by formulas

$$ESA_Δ = 105 + 0{,}5 \times 100 = 155 \text{ μm};$$

$$EIA_{\Delta} = 105 - 0{,}5 \times 100 = 55 \text{ µm.}$$

Obtain $A_{\Delta} = 0{,}6^{+0{,}115}_{+0{,}055}$.

The application of a probability calculation method with the same allowance for the initial link allowed the application of the ninth grade for processing the constituent links instead of the seventh.

For the initial link, by the condition of the problem, it was required to assign a symmetric tolerance limit, for example $A_{\Delta} = 0{,}6 \pm 0{,}05$ mm. In order to withstand this condition, it is necessary to assign the tolerance limits J_s9 for the increasing ones, j_s9 for the decreasing ones, and j_s8 for the dependent link A_1.

Example 5. For turned part machining the shaft (Figure 2, *a*), a dimension of $\varnothing$ $80.5_{-0.12}$ mm is set .For the final processing of the shaft, the size is set by grinding $\varnothing 80_{-0.74}$ mm. Determine the depth of milling of the transverse slot, if the slot is milled before grinding, and its final depth is set equal to $8^{+0.15}$ mm.

Solution. Compose a dimension chain, guided by the following considerations: the shaft is completely processed in the centers; and the form deviation can be ignored; since the slot is milled after grinding, then 'the given size of the slot depth is obtained last, in fact, this dimension is the closing link of the composed dimension chain; in the problem the measuring base is replaced, at the final machined shaft the slot depth is measured from point *b* (Figure 3, a) in the problem such a base is the point *a*. In the dimension chain it is necessary to introduce an additional link A_3, which is the desired quantity.

Begin the construction of the dimension chain (Figure 3, *b*) from the point *b*. Pit – the constituent links in the direction of the arrows. The increasing links are the links $A_1 = 80_{-0.74}/2 = 40_{-0.37}$ *and* A_3; reducing the link $A_2 = 80{,}5_{-0.12}/2 = 40.25_{-0.06}$; the closing link $A_{\Delta} = 8^{+0.15}$.

Nominal size of the link A_3 find by the formula

$$A_3 = A_{\Delta} - A_1 + A_2 = 8 - 40 + 40{,}25 = 8{,}25 \text{ mm.}$$

Limit sizes $A_{3\max}$ and $A_{3\min}$ is determined by transforming the formulas

$$A_{3\max} = A_{\Delta\max} - A_{1\max} + A_{2\min} = 8{,}15 - 40 + 40{,}19 = 8{,}34 \text{ mm;}$$

$$A_{3\min} = A_{\Delta\min} - A_{1\min} + A_{2\max\,in} = 8 - 39{,}963 + 40{,}25 = 8{,}287 \text{ mm.}$$

Limit deviation of size A_3 is convenient to calculate by formulas, mm

$$ESA_3 = A_{3\max} - A_3 = 8{,}34 - 8{,}25 = 0{,}09 \text{ µm;}$$

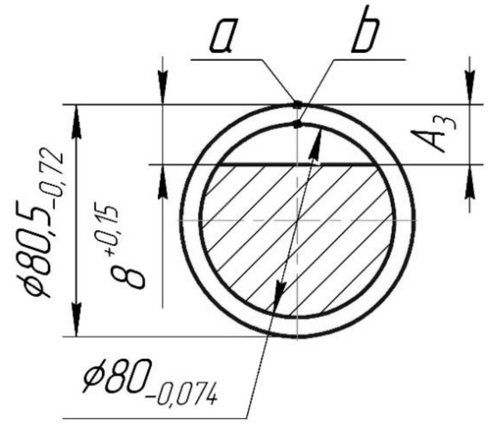
$$EIA_3 = A_{3\min} - A_3 = 8{,}287 - 8{,}25 = 0{,}037 \text{ µm.}$$

Tolerance of the link A_3:

$$TA_3 = ESA_3 - EIA_3 = A_{3\max} - A_{3\min} = 0{,}053 \text{ mm.}$$

Obtain $A_3 = 8{,}25^{+0{,}090}_{+0{,}037}$.

Calculate ESA_3, EIA_3, TA_3 by the formulas

a) 

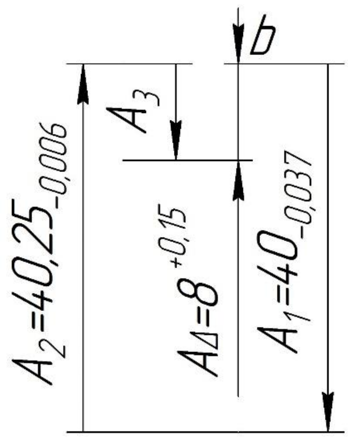
b) 

Figure 3 – Is a shaft

Annex A
(informative)

Basic series of preferred numbers

R5	R10	R20	R40	R5	R10	R20	R40
1,0	1,0	1,0	1,00				3,35
			1,06			3,55	3,55
		1,12	1,12				3,75
			1,18	4,00	4,00	4,00	4,00
	1,25	1,25	1,25				4,25
			1,32			4,50	4,50
		1,40	1,40				4,75
			1,50		5,00	5,00	5,00
1,60	1,60	1,60	1,60				5,30
			1,70			5,60	5,60
		1,80	1,80				6,0
			1,90	6,30	6,30	6,30	6,30
	2,00	2,00	2,00				6,70
			2,12			7,10	7,10
		2,24	2,24				7,50
			2,36		8,00	8,00	8,00
2,50	2,50	2,50	2,50				8,50
			2,65			9,00	9,00
		2,80	2,80				9,50
			3,00	10,00	10,00	10,00	10,00
	3,15	3,15	3,15				

Note

1. Additional series R80 and R160 with a denominator of progression, respectively, $q \approx 1{,}03$ и $q \approx 1{,}02$ are also established

2. Numbers over 10 are obtained by multiplying the numbers given in the table by 10; 100, etc., and less than 1 – multiplying by 0,1; 0,01, etc.

Annex B
(informative)

Basic series of normal linear dimensions in the interval 1 ... 1600 mm

Ra5	Ra10	Ra20	Ra40	Ra5	Ra10	Ra20	Ra40	Ra5	Ra10	Ra20	Ra40
1,0	1,0	1,0	1,0				11,5			140	140
			1,05		12	12	12				150
		1,1	1,1				13	160	160	160	160
			1,15			14	14				170
	1,2	1,2	1,2				15			180	180
			1,3	16	16	16	16				190
		1,4	1,4				17		200	200	200
			1,5			18	18				210
1,6	1,6	1,6	1,6				19			220	220
			1,7		20	20	20				240
		1,8	1,8				21	250	250	250	250
			1,9			22	22				260
	2,0	2,0	2,0				24			280	280
			2,1	25	25	25	25				300
		2,2	2,2				26		320	320	320
			2,4			28	28				340
2,5	2,5	2,5	2,5				30			360	360
			2,6		32	32	32				380
		2,8	2,8				34	400	400	400	400
			3,0			36	36				420
	3,2	3,2	3,2				38			450	450
			3,4	40	40	40	40				480
		3,6	3,6				42		500	500	500
			3,8			45	45				530
4,0	4,0	4,0	4,0				48			560	560
			4,2		50	50	50				600
		4,5	4,5				53	630	630	630	630
			4,8			56	56				670
	5,0	5,0	5,0				60			710	710
			5,3	63	63	63	63				750
		5,6	5,6				67		800	800	800
			6,0			71	71				850
6,3	6,3	6,3	6,3				75			900	900
			6,7		80	80	80				950
		7,1	7,1				85	1000	1000	1000	1000
			7,5			90	90				1060
	8,0	8,0	8,0				95			1120	1120
			8,5	100	100	100	100				1180
		9,0	9,0				105		1250	1250	1250
			9,5			110	110				1320

10	10	10	10				120			1400	1400
			10,5		125	125	125				1500
		11	11				130	1600	1600	1600	1600

Annex C
(informative)

Number of tolerance units *(a)* for the USTF accuracy grades

Grade	a	Function of the grade	Grade	a	Function of the grade
01	1,0	End measuring rod	9	40	Coupling size
0	1,41		10	64	
1	2,0		11	100	
2	2,74	Gauges, high precise measurements	12	160	
3	3,74		13	250	
4	5,12		14	400	Dimensions with unspecified tolerances
5	7,0	Coupling size	15	640	
6	10,0		16	1000	
7	16,0		17	1600	
8	25,0		18	2500	
Note 1. For sizes up to 500 mm, tolerances from 01 to 4 grades are determined by the following formulas: $IT01 = 0,3 + 0,008D_m$; $IT0 = 0,5 + 0,12D_m$; $IT1 = 0,8 + 0,20D_m$; $IT2 = \sqrt{IT1 \cdot IT3}$; $IT3 = \sqrt{IT1 \cdot IT5}$; $IT4 = \sqrt{IT3 \cdot IT5}$, where *IT* is in micrometers, *D* is in millimeters					

Annex D
(informative)

Tolerance units *i* (*I*) and values of geometric mean diameters D_m (mm) for the main intervals of nominal dimensions ESKD

Interval	D_m	$i(I)$	Interval	D_m	$i(I)$
up to 3	1,73	0,55	from 180 to 250	219	2,89
from 3 to 6	4,24	0,73	from 250 to 315	281	3,22
from 6 " 10	7,75	0,73	from 315 to 400	355	3,54
" 10 " 18	13,4	1,08	from 400 to 500	448	3,89
" 18 " 30	23,2	1,31	from 500 to 630	564	4,35
" 30 " 50	38,7	1,56	from 630 to 800	713	5,00
" 50 " 80	63,2	1,86	from 800 to 1000	896	5,70
" 80 " 120	97,8	2,17	from 1000 to 1250	1127	6,60
" 120 " 180	147	2,52	from 1250 to 1600	1400	7,70

Annex E
(informative)

Table E.1 is tolerances for sizes up to 1600 mm in grades from 01 to 04

Interval of size	Tolerances for grades, microns					
	01	0	1	2	3	4
up to 3	0,3	0,5	0,8	1,2	2,0	3,0
from 3 to 6	0,4	0,6	1,0	1,5	2,5	4,0
from 6 " 10	0,4	0,6	1,0	1,5	2,5	4,0
" 10 " 18	0,5	0,8	1,2	2,0	3,0	5,0
" 18 " 30	0,6	1,0	1,5	2,5	4,0	6,0
" 30 " 50	0,6	1,0	1,5	2,5	4,0	7,0
" 50 " 80	0,8	1,2	2,0	3,0	5,0	8,0
" 80 " 120	1,0	1,5	2,5	4,0	6,0	10
" 120 " 180	1,2	2,0	3,5	5,0	8,0	12
" 180 " 250	2,0	3,0	4,5	7,0	10	14
" 250 " 315	2,5	4,0	6,0	8,0	12	16
" 315 " 400	3,0	5,0	7,0	9,0	13	18
" 400 " 500	4,0	6,0	8,0	10	15	20
" 500 " 630	4,5	6,0	9,0	11	16	22
" 630 " 800	5,0	7,0	10	13	18	25
" 800 " 1000	5,5	8,0	11	15	21	29
" 1000 " 1250	6,5	9,0	13	18	24	34
" 1250 " 1600	8	11	15	21	29	40

Table E.2 is tolerances for sizes up to 1600 mm in grades from 05 to 18*

Interval of size	Tolerances for grades, microns					
	5	6	7	8	9	10
1	2	3	4	5	6	7
up to 3	4,0	6,0	10	14	25	40
from 3 to 6	5,0	8,0	12	18	30	48
from 6 " 10	6,0	9,0	15	22	36	58
" 10 " 18	8,0	11	18	27	43	70
" 18 " 30	9,0	13	21	33	52	84
" 30 " 50	11	16	25	39	62	100
" 50 " 80	13	19	30	46	74	120
" 80 " 120	15	22	35	54	87	140
" 120 " 180	18	25	40	63	100	160

1	2	3	4	5	6	7
" 180 " 250	20	29	46	72	115	185
" 250 " 315	23	32	52	81	130	210
" 315 " 400	25	36	57	89	140	230
" 400 " 500	27	40	63	97	155	250
" 500 " 630	30	44	70	110	175	280
" 630 " 800	35	50	80	125	200	320
" 800 " 1000	40	56	90	140	230	360
" 1000 " 1250	46	66	105	165	260	420
" 1250 " 1600	54	78	125	195	310	500

* Tolerances of grades 11 ... 15 are calculated by the formula $IT(N+5)=10\,ITN$. Tolerances of grades 16 and higher – by the formula $IT(N+10)=100\,ITN$.

Annex F
(informative)

Coefficients of linear expansion in the temperature range from 20 to 100 °C

Material	$\alpha \cdot 10^{-6}$	Material	$\alpha \cdot 10^{-6}$
Steel 10	12,4-13,2	Steel 50	12,8-13,0
Steel 15	12,4-13,2	Steel 55	11,6-12,7
Steel 20	12,3-13,1	Steel 60	11,6-11,9
Steel 25	12,2-13,0	Cast iron	10,0-12,0
Steel 30	12,1-12,9	Bronze BrSnZnPb6-6-3	15,1-19-1
Steel 35	12,0-12,9	Bronze BrAlFe9-4	15,8-19-8
Steel 40	11,9-12,8	Brass CuZnAlFeMn 66-6-3-2	16,7-20,7
Steel 45	11,9-12,7	Brass CuZnMnSnPb 58-2-2-2	16,0-18,0

Annex G
(informative)

Is a dimensional accuracy and surface roughness of parts

Type of treatment	Grades	Ra, µm
1	2	3
Finishing	5-8	0,1…0,4
Fine finishing	5	0,05
Chiselling	14-15	25…50
Chipping finishing	12-13	3,2…12,5
Sandmold casting	14-17	100
Metal mold casting	14-16	25
Rough turning	15-17	25…100
Semifinish-turning	12-17	6,3…12,5
Finish-turning	7-13	1,6…3,2
Fine turning, diamond	6	0,4…0,8
High-speed turning	11	1,6
Gear machining by shaping	7-10	3,2…6,3
Gear machining by milling	7-10	3,2
Gear machining by grinding	5-6	0,4…0,8
Gear machining by shaving	5-6	0,8…1,6
Lapping	6-8	0,4…3,2
Feather lapping	5	0,1…1,6
Stretching	7-9	0,8…6,3
Finishing Stretching	7	0,1…0,2
Deployment	7-10	1,6…12,5
Fine deployment	6-7	0,8
Rough boring	15-17	50…100
Semifinish-boring	12-14	12,5…25
Finish-boring	6-9	1,6…3,2
Fine boring, diamond	7	0,4…0,8
Screwcutting by die, tap	6-8	3,2…12,5
Screwcutting by graver, rack	6-8	3,2…6,3
Thread cutting with a milling cutter	8	3,2…6,3
Knurling	6-8	0,4…0,8
Grinding	4-6	1,6…3,2
Drilling	12-14	6,3…25
Jig drilling	10-11	6,3…25
Shaping	10-14	3,2…25

1	2	3
Fine shaping	7-10	1,6
Superfinishing	4-5	0,1…0,4
Rough milling	11-14	6,3…50
Final milling	10-11	3,2…6,3
Fine milling	7-9	1,6
High-speed milling	11-14	0,8…3,2
Honing	6-8	0,2…0,4
Rough shaving	11	1,6…6,3
Fine shaving	8-9	0,1…0,8
Pregrinding	8-11	3,2…6,3
Grind finishing	6-8	0,8…1,6
Polish-grinding	4-7	0,2…0,4

Annex H
(informative)

Is examples of application of the accuracy degrees, the form of the surfaces position

Accuracy degree	Examples of tolerance applications				
	Flatness and straightness	Cylindricity	Parallelity	Perpendicularity and axial runout	Coaxiality and radial runout
1…4	-	Balls and rollers, bearing races and bearing seat area of special and high precision rolling bearings and associated shafts and housings; high-pressure bush bearing	Super precision and particular accuracy guides	Super precision and particular accuracy guides; rings of super precise rolling bearings and shaft collars for them	Working surfaces of bearing rings of super precise accuracy; centering plug of shafts for super-precision gear wheels: the pilot fillets of the shafts of large heavily loaded machines
5, 6	Thrust bearing of heavily loaded machines; Precise guiding machines and devices	Bearing seat area of normal rolling bearings and associated shafts and housings; sliding bearings operating under heavy loads	The ends of the rolling bearings and the axis of the holes in the high precision gears	Bearing collars of shafts and casings for high precision rolling bearings; flanges of shafts and couplings; ends of bearing shells	Bearing rings of normal accuracy; mounting surfaces of shafts under gear wheels of normal and increased accuracy; high-speed shafts of high accuracy
7, 8	Thrust bearings of low power machines; supporting and detachable surfaces of bearing housings, reducers,	Bush bearing in knots of large and slow-moving machines and reducers	Support surfaces of caps and rings of rolling bearings, as well as the axis of holes in the gears bodies of normal	Collars of shafts and casings for rolling bearings of normal accuracy; the ends of the hubs and spacer sleeves; holes in the housings of bevel reducers	Holes for end caps and liners in bearing housings; high-speed shafts of normal accuracy; gear rims and matching bore of gear wheels of low accuracy

	foundation frames		accuracy		
9, 10	Mounting and supporting surfaces of various mechanisms and flange connections	Sliding bearings operating at low speeds and pressures	End faces of bearing caps in heavy engineering; axes of slow speeds	End faces of low-loaded bearings; axes of threaded pins relative to support surfaces	Seated surfaces of shafts for gear wheels of reduced accuracy
11, 12	Nonchargeable working surfaces, supporting and detachable surfaces in mechanisms of reduced accuracy				

Annex J
(informative)

Recommendations for the parameters choice of roughness and direction of surface lays

Design feature, operating conditions of surfaces	Roughness parameter of direction of lay
Friction, sensitive to wear, contact stresses	*Ra (Rz)* and *tp*; multidirectional
Pressure-tight joint surface; cylindrical flat, face	*Ra (Rz)* and *tp*; perpendicular to the axis, multidirectional, circular
Fixed, loaded, for example, press joint surfaces	*Ra (Rz)*; parallel to press axis; *Ra (Rz)* and *tp*; multidirectional
Experiencing cyclic loads; work in bending or torsion under variable loads	*Rmax*, *Sm (S)*; multidirectional или parallel to neutral axis or axis of swing
Note – From the height parameters of the lays, the *Ra* parameter is preferable. The parameters *Rz* and *Rmax* are normalized, if it is necessary to limit the height of the lays, it is impossible to control the parameter with a profilometer or reference samples, etc.	

Annex K
(informative)

Is examples of the normalization of the surface roughness *Ra* of typical machine parts and their joints

Mounting surfaces of parts	Grades	Nominal dimensions (mm) and parameter Ra		
		up to 50	50…120	120…500
Interchangeable	6, 7	0,4…0,8 (0,4…1,6)	0,8…1,6 (0,8…3,2)	
Assembled, under press	8	0,4…0,8 (0,4…1,6)	0,8…1,6 (1,6…3,2)	1,6…3,2*

Workpiece surface for the fits with accurate centering	Radial runout tolerance, µm				
	2,5	4 и 6	10	16	25
Shaft	0,05	0,1	0,2	0,4	0,8
Hole	0,1	0,2	0,4	0,8	1,6

Plain bearing seat area	Grades		Liquid lubrication
	6…9	10…12	
Shaft	0,4…0,8	0,8…3,2	0,1…0,4*
Hole	0,8…1,6	1,6…3,2	0,2…0,8

Surface of axles, shafts for seals		Speed, m/s		
		up to 3	from 3 to 5	from 5
Rubber	polishing	0,8…1,6	0,4…0,8	0,2…0,4
Felt		0,8…1,6		-
hydrodynamic and fat groove		3,2…6,3		

Surfaces of guides	Speed, m/s	Flatness tolerance of 100 mm, µm			
		up to 6	up to 10	up to 25	up to 60
Slade	Not more than 0,5 (more than 0,5)	0,2 (0,1)	0,4 (0,2)	0,8 (0,4)	1,6 (0,8)
Linear		0,1 (0,05)	0,2 (0,1)	0,4 (0,2)	0,8 (0,4)

Flange joints and other similar fixed joints	Perpendicularity tolerance of 100 mm, µm		
	до 25	до 60	св. 60
	1,6	3,2	6,3

The connector surface of the gearbox housings, bearings, etc.	Tight joint		Leaking joint
	with sealing	without sealing	
	3,2…6,3	0,8…1,6	6,3…12,5

Working surfaces of belt pulleys	Pads	Clutchs	Wheels	Tires
	1,6…3,2	0,8…1,6	0,1…0,8	0,2…0,8

Connections with prismatic and semicircular keys	Working areas	Nonworking area
	1,6…3,2	6,3…12,5

* Refers to shafts and holes
Note – the parameter values without brackets refer to shafts, and in brackets to the holes

Annex L
(informative)

Is tolerances and deviations of gauges, μm

Grades of item tolerances	Designation of dimensions and tolerances	Size range, mm													Tolerance on the gauge form
		Up to 3	From 3 to 6	From 6 to 10	From 10 to 18	From 18 to 30	From. 30 to 50	From 50 to 80	From 80 to 120	From 120 to 180	From 180 to 250	From 250 to 315	From 315 to 400	From 400 to 500	
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16
6	Z	1	1,5	1,5	2	2	2,5	2,5	3	4	5	6	7	8	
	Y	1	1	1	1,5	1,5	2	2	3	3	4	5	6	7	
	α, α_1	0	0	0	0	0	0	0	0	0	2	3	4	5	
	Z_1	1,5	2	2	2,5	3	3,5	4	5	6	7	8	10	11	
	Y_1	1,5	1,5	1,5	2	3	3	3	4	4	5	6	6	7	
	H, Hs	1,2	1,5	1,5	2	2,5	2,5	3	4	5	7	8	9	10	IT1
	H_1	2	2,5	2,5	3	4	4	5	6	8	10	12	13	15	IT2
	Hp	0,8	1	1	1,2	1,5	1,5	2	2,5	3,5	4,5	6	7	8	IT1
7	Z, Z_1	1,5	2	2	2,5	3	3,5	4	5	6	7	8	10	11	
	Y, Y_1	1,5	1,5	1,5	2	3	3	3	4	4	6	7	8	9	
	α, α_1	0	0	0	0	0	0	0	0	0	3	4	6	7	
	H, H_1	2	2,5	2,5	3	4	4	5	6	8	10	12	13	15	IT2
	Hs	-	-	1,5	2	2,5	2,5	3	4	5	7	8	9	10	IT1
	Hp	0,8	1	1	1,2	1,5	1,5	2	2,5	3,5	4,5	6	7	8	IT1
8	Z, Z_1	2	3	3	4	5	6	7	8	9	12	14	16	18	
	Y, Y_1	3	3	3	4	4	5	5	6	6	7	9	9	11	
	α, α_1	0	0	0	0	0	0	0	0	0	4	6	7	9	
	H	2	2,5	2,5	3	4	4	5	6	8	10	12	13	15	IT2
	H_1	3	4	4	5	6	7	8	10	12	14	16	18	20	IT3
	Hs*, Hp	1,2	1,5	1,5	2	2,5	2,5	3	4	5	7	8	9	10	IT1
9	Z, Z_1	5	6	7	8	9	11	13	15	18	21	24	28	32	
	Y, Y_1	0	0	0	0	0	0	0	0	0	0	0	0	0	
	α, α_1	0	0	0	0	0	0	0	0	0	4	6	7	9	
	H	2	2,5	2,5	3	4	4	5	6	8	10	12	13	15	IT2
	H_1	3	4	4	5	6	7	8	10	12	14	16	18	20	IT3
	Hs, Hp	1,2	1,5	1,5	2	2,5	2,5	3	4	5	7	8	9	10	IT1
10	Z, Z_1	5	6	7	8	9	11	13	15	18	24	27	32	37	
	Y, Y_1	0	0	0	0	0	0	0	0	0	0	0	0	0	
	α, α_1	0	0	0	0	0	0	0	0	0	7	9	11	14	
	H	2	2,5	2,5	3	4	4	5	6	8	10	12	13	15	IT2
	H_1	3	4	4	5	6	7	8	10	12	14	16	18	20	IT3
	Hs, Hp	1,2	1,5	1,5	2	2,5	2,5	3	4	5	7	8	9	10	IT1
11	Z, Z_1	10	12	14	16	19	22	25	28	32	40	45	50	55	
	Y, Y_1	0	0	0	0	0	0	0	0	0	0	0	0	0	
	α, α_1	0	0	0	0	0	0	0	0	0	10	15	15	20	
	H, H_1	4	5	6	8	9	11	13	15	18	20	23	25	27	IT4
	Hs	-	-	4	5	6	7	8	10	12	14	16	18	20	IT3
	Hp	1,2	1,5	1,5	2	2,5	2,5	3	4	5	7	8	9	10	IT1
12	Z, Z_1	10	12	14	16	19	22	25	28	32	45	50	65	70	
	Y, Y_1	0	0	0	0	0	0	0	0	0	0	0	0	0	
	α, α_1	0	0	0	0	0	0	0	0	0	15	20	30	35	
	H, H_1	4	5	6	8	9	11	13	15	18	20	23	25	27	IT4
	Hs	-	-	4	5	6	7	8	10	12	14	16	18	20	IT3
	Hp	1,2	1,5	1,5	2	2,5	2,5	3	4	5	7	8	9	10	IT1
13	Z, Z_1	20	24	28	32	36	42	48	54	60	80	90	100	110	
	Y, Y_1	0	0	0	0	0	0	0	0	0	0	0	0	0	
	α, α_1	0	0	0	0	0	0	0	0	0	25	35	45	55	
	H, H_1	10	12	15	18	21	25	30	35	40	46	52	57	63	IT5
	Hs	-	-	9	11	13	16	19	22	25	29	32	36	40	IT5
	Hp	2	2,5	2,5	3	4	4	5	6	8	10	12	13	15	IT2
14	Z, Z_1	20	24	28	32	36	42	48	54	60	100	110	125	145	
	Y, Y_1	0	0	0	0	0	0	0	0	0	0	0	0	0	
	α, α_1	0	0	0	0	0	0	0	0	0	45	55	70	90	
	H, H_1	10	12	15	18	21	25	30	35	40	46	52	57	63	IT5
	Hs	-	-	9	11	13	16	19	22	25	29	32	36	40	IT5
	Hp	2	2,5	2,5	3	4	4	5	6	8	10	12	13	15	IT2
15	Z, Z_1	40	48	56	64	72	80	90	100	110	170	190	210	240	

	Y, Y_1	0	0	0	0	0	0	0	0	0	0	0	0	0	
	α, α_1	0	0	0	0	0	0	0	0	0	70	90	110	140	
	H, H_1	10	12	15	18	21	25	30	35	40	46	52	57	63	IT5
	Hs	-	-	9	11	13	16	19	22	25	29	32	36	40	IT5
	Hp	2	2,5	2,5	3	4	4	5	6	8	10	12	13	15	IT2
16, 17	Z, Z_1	40	48	56	64	72	80	90	100	110	210	240	280	320	
	Y, Y_1	0	0	0	0	0	0	0	0	0	0	0	0	0	
	α, α_1	0	0	0	0	0	0	0	0	0	110	140	180	220	
	H, H_1	10	12	15	18	21	25	30	35	40	46	52	57	63	IT5
	Hs	-	-	9	11	13	16	19	22	25	29	32	36	40	IT5
	Hp	2	2,5	2,5	3	4	4	5	6	8	10	12	13	15	IT2

Annex M
(informative)

Limit deviations of the sizes of radial and radial thrust bearings, µm

Nominal bore diameter d, outer D, mm	Inner ring					Inner and outer rings	Outer ring		
	Plain bore			U_p	R_i				
	d_m	d				B or C	D_m	D	
	-EI	-EI	+ES	not more		-ei	-ei	-ei	+es
	Accuracy class 0								
from 2,5 to 10	8	10	2	15	10	120	–	–	–
from 10 " 18	8	11	3	20	10	120	8	10	2
" 18 " 30	10	13	3	20	13	120	9	11	2
" 30 " 50	12	15	3	20	15	120	11	14	3
" 50 " 80	15	19	4	25	20	150	13	17	4
" 80 " 120	20	25	5	25	25	200	15	20	5
" 120 " 150	–	–	–	–	–	250	18	24	6
" 150 " 180	–	–	–	–	–	250	25	32	7
" 180 " 250	–	–	–	–	–	300	30	38	8
" 250 " 315	–	–	–	–	–	350	44	39	10
	Accuracy class 6								
from 2,5 to 10	7	8	1	10	6	120	–	–	–
from 10 " 18	7	8	1	10	7	120	7	8	1
" 18 " 30	8	9	1	10	8	120	8	9	1
" 30 " 50	10	11	1	10	10	120	9	11	2
" 50 " 80	12	14	2	12	10	150	11	13	2
" 80 " 120	15	18	3	12	13	200	13	15	2
" 120 " 150	–	–	–	–	–	250	15	18	3
" 150 " 180	–	–	–	–	–	250	18	21	3
" 180 " 250	–	–	–	–	–	300	20	24	4
" 250 " 315	–	–	–	–	–	350	29	24	5

Annex O
(informative)

Limit deviation of the dimensions of the tapered roller bearings, µm

Nominal bore diameter d, outer D, mm	Inner ring				Outer ring			
	d_m	d		R_i	D_m	D		R_a
	-EI	-EI	+ES		-ei	-ei	+es	
	Accuracy class 0							
from 10 to 18	8	11	3	15	–	–	–	–
from 18 " 30	10	13	3	18	9	11	2	18
" 30 " 50	12	15	3	20	11	14	3	20
" 50 " 80	15	19	4	25	13	17	4	25
" 80 " 120	20	25	5	30	15	20	5	35
" 120 " 150	–	–	–	–	18	24	6	40
" 150 " 180	–	–	–	–	25	32	7	45
" 180 " 250	–	–	–	–	30	38	8	50
	Accuracy class 6							
from 10 to 18	7	8	1	7	–	–	–	–
from 18 " 30	8	9	1	8	8	9	1	9
" 30 " 50	10	11	1	10	9	11	2	10
" 50 " 80	12	14	2	10	11	13	2	13
" 80 " 120	15	18	3	13	13	15	2	18
" 120 " 150	–	–	–	–	15	18	3	20
" 150 " 180	–	–	–	–	18	21	3	23
" 180 " 250	–	–	–	–	20	24	4	25

Annex P
(informative)

Recommended tolerance limits for rolling bearings for local and oscillation loading of their rings

<table>
<tr><td rowspan="3">Kind of loading</td><td rowspan="3">Size of bore diameter, mm</td><td colspan="3">Fits</td><td rowspan="2">Type of bearing</td></tr>
<tr><td rowspan="2">to shaft</td><td colspan="2">in steel or cast iron housing</td></tr>
<tr><td>solid</td><td>split</td><td></td></tr>
<tr><td colspan="6">quiescent load or load with moderate shocks and vibration</td></tr>
<tr><td rowspan="3">Local</td><td>up to 80</td><td rowspan="2">h, g, f</td><td>H</td><td rowspan="3">H</td><td rowspan="3">all types, except for stamped needle</td></tr>
<tr><td>from 80 to 260</td><td rowspan="2">G</td></tr>
<tr><td>from 260 to 500</td><td>f</td></tr>
<tr><td colspan="6">load with shocks and vibration</td></tr>
<tr><td rowspan="5">Local</td><td>up to 80</td><td rowspan="2">h</td><td>J_s, J</td><td rowspan="3">J_s, J</td><td rowspan="3">all types, except for stamped needle and double-row tapered</td></tr>
<tr><td>from 80 to 260</td><td rowspan="2">H</td></tr>
<tr><td>from 260 to 500</td><td>g</td></tr>
<tr><td>up to 120</td><td>h</td><td rowspan="2">H</td><td rowspan="4">J_s, J</td><td rowspan="2">double-row tapered-roller</td></tr>
<tr><td>from 120 to 1600</td><td>g</td></tr>
<tr><td rowspan="2">Oscillation</td><td>up to 80</td><td>k</td><td rowspan="2">K</td><td rowspan="2">all types</td></tr>
<tr><td>from 80 to 260</td><td>j_s, j</td></tr>
<tr><td colspan="6">Note – The grade of the fundamental deviation of the tolerance limit is determined by the accuracy class of the bearing. For example, for bearings of accuracy classes 0 and 6, tolerances are IT6 for shafts and IT7 for housings</td></tr>
</table>

Annex Q
(informative)

Limits of radial load intensity, N/mm

Diameter of inner bearing ring, *d*, mm	The fundamental deviation of the shaft tolerance limit			
	j_s	*k*	*m*	*n*
from 18 to 80	up to 300	300-1400	1400-1600	1600-3000
from 80 " 180	" 600	600-2000	2000-2500	2500-4000
" 180 " 360	" 700	700-3000	3000-3500	3500-6000
Diameter of outer bearing ring, *D*, mm	The fundamental deviation of the housing tolerance limit			
	K	*M*	*N*	*P*
from 50 to 180	up to 800	800-1000	1000-1300	1300-2500
from 180 " 360	" 1000	1000-1500	1500-2000	2000-3300
" 360 " 630	" 1200	1200-2000	2000-2600	2600-4000
Note – See Annex P				

Annex R
(informative)

Table R.1 is a dynamic fit coefficient k_1

Type of load	k_1
Loads with moderate shocks and vibration; overload up to 150%	1,0
Loads with heavy shocks and vibration; overload up to 300%	1,8

Table R.2 is the coefficient, taking into account the loosing of the interference fit k_2

$\frac{d_{hole}}{d}$ or $\frac{D}{D_{house}}$	Shaft			Housing
	$D/d \leq 1{,}5$	$D/d > 1{,}5...2$	$D/d < 1{,}5...2$	
Up to 0,4	1,0	1,0	1,0	1,0
From 0,4 to 0,7	1,2	1,4	1,6	1,0
" 0,7 " 0,8	1,5	1,7	2,0	1,4
" 0,8	2,0	2,3	3,0	1,8

Notes

d, D are respectively, the diameters of the seating surfaces of the inner and outer bearing rings;

d_{hole} is a hole diameter of hollow shaft;

D_{house} is an outer surface diameter

Table T.3 is values k_3

$\frac{A}{R} ctg\beta$	k_3
Up to 0,2	1,0
From 0,2 to 0,4	1,2
" 0,4 " 0,6	1,4
" 0,6 " 1,0	1,6
" 1,0	2,0

Notes

1 R and A are respectively, axial and radial reaction, β is pressure angle;

2 For radial and radial thrust bearings with one outer or inner ring

Annex S
(informative)

Normal fit-up G_{re} for some types of rolling bearings with plain bores

Nominal diameter, *d*, mm	Type of bearing									
	1		2		3		4		5	
	Radial initial gap, μm									
	min	max	min	max	min	max	min	max	min	max
From 2,5 to 6			5	10						
	2	13								
" 6 " 10			6	12						
" 10 " 14	3	18	6	13						
					10	40	9	17	20	35
" 14 " 18			8	15						
" 18 " 24			10	17						
	5	20								
" 24 " 30			11	19	10	45			25	40
" 30 " 40	6	20	13	23	15	60	10	20	30	45
" 40 " 50	6	23	14	25	20	55	13	23	35	55
" 50 " 65	8	28	16	30	20	65	15	27	40	65
" 65 " 80	10	30	18	35	25	75	20	35	50	80
" 80 " 100	12	36	22	42	30	80	25	45	60	100
" 100 " 120	15	41	25	50	35	90	30	50	75	120

Note
1 – Radial ball single-row without grooves for inserting balls;
2 – Radial spherical ball;
3 – Radial roller with short cylindrical rollers;
4 – Radial spherical single row roller;
5 – Radial double-row roller.

Annex T
(informative)

Roughness and form accuracy of the mounting surfaces of shafts and casings for connection to ball and roller bearings

Mounting surfaces of parts	Accuracy class of ball and roller bearings	Ra, µm, not more, for nominal diameters, mm	
		up to 80	more than 80
Shaft and casing	0	1,25	2,5
	6	0,63	1,25
Shoulder faces of shafts and casings	0	2,5	2,5
	6	1,25	2,5
Note – The ovality deviation and taper of the shafts and housings in the mounting area of the rolling bearings should not exceed 0.5 Td or 0.5 TD			