Tomsk Polytechnic University

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TECHNOLOGY of MECHANICAL ENGINEERING

Part 1

Textbook

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This textbook is devoted to fixing of accuracy in mechanical engineering and bases of the theory of cutting tools.

The textbook is prepared at the Department of Mechanical Engineering of Tomsk Polytechnic University. It is recommended for foreign students following the Bachelor Degree Program in Mechanical Engineering at Tomsk Polytechnic University.

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PREFACE

The discipline "Technology of Mechanical Engineering" is a finishing rate in preparation of the experts under the program "Mechanical Engineering". For its study the knowledge of disciplines "Processing of materials", "Strength of materials" is required.

Fundamentals of the discipline "Technology of mechanical engineering" is studied on the senior rate for the bachelor level in 6, 7 and 8 semesters and is divided into 3 parts. The part "**Metrology, Standardization and Certification”** is studied in the sixth semester (3 credits) and contains **course project**. At the end of 6-th semester the **examination** and defense of **course project** is stipulated.

The part second "**Cutting and Cutting Tools**" is studied in the **sixth**, **seventh** and **eighth** semesters (4 + 4 + 4 = 12 credits) and contains **course project** in the **8** semester. **Credit test**, **examination,** and defense of **course project** are stipulated correspondently.

The part third "**Fundamentals of Mechanical Engineering**" is studied in the **sixth** semester (6 credits). At the end of semester the **examination** is stipulated.

The discipline "**Technology of Mechanical Engineering**" is continued in the **seventh** and **eighth** semesters (6 + 2= 8 credits). **Examination, examination** and defense of **course project** are stipulated correspondently.

This textbook provides the most comprehensive introduction to technology of mechanical engineering. Measurements throughout the textbook are given according to the SI Metric system of measurement. The content is generously illustrated, and the language used is simple and direct.

This text book is written on the basis of the book of Victor E. Repp and Willard J. McCarthy “Machine Tool Technology”. The author of this textbook expresses gratitude to the authors of the book for the given opportunity to use its material.

Suggestions on improvements of future editions of the textbook are welcome.

Part 1. Metrology, Standardization and Certification

Chapter1. Introduction

The machining of metal as we know it today has been forming for centuries. It received its great impurities with the development of the steam engine. The transition from the iron age to the steel age in the second half of the nineteenth century facilitated the rapid development of machines. Year by year they have become more ingenious and complicated and production operations would not be possible without measurements. Precision measurement is the key to producing interchangeable parts and mass producing consumer goods. Every part must be made accurately, within specified limits, to the size and shape specified by the designer. Inaccurately made parts will not assemble and fit properly with mating parts. Hence, the finished product may not operate properly, or may wear out sooner than it should.

All workers in the machine shop must be responsible for accurate work. Accurate workmanship depends primarily on accurate measurement and layout work. To insure accuracy, machinists must know the principles of measurements. They also must know how to use the common hand tools, measuring instruments, and gages used in the trade.

1.1. Linear Measurement

The International System of Units (abbreviated SI) is widely used. The United States of America is the last industrialized nation to continue use of the English system of linear (straight line) measurement. But with legislation passed in 1975, conversion to the modern form of the metric system is expected to accelerate. The standard SI metric unit of linear measurement is the **meter.** The meter is subdivided into the following parts:

1 meter =10 decimeters (dm), 1 decimeter =10 centimeters (cm), 1 centimeter = 10 millimeters (mm), 1 micrometer or micron (1 μm) = 0.001 mm. Hence, one **decimeter** is one-tenth meter, one **centimeter** is one-hundredth meter, and one **millimeter** is one-thousandth meter. Other subdivisions of the meter are also included in the metric system. One meter is equal to 39.37 inches. One **inch** is equal to 2.54 centimeters or 25.4 millimeters. Both dimensions are exact and are not rounded (further decimal positions would be zeros.) For measurements finer than one millimeter, steel rules are available with 1/2 mm graduations. For precision measurement, micrometers are available which measure as finely as 0.01 mm.

1.2. Standards for Measurement

Standards for linear measurement were established by the International Bureau of Weights and Measures. This bureau was created in 1875 and is located near Paris, France. It has representatives from most nations of the world. It keeps models or standards for units of metric measurement, including a standard for the meter.

The international standard for the meter is the length between two finely inscribed lines on a platinum-iridium-alloy metal bar at the temperature 20°C. This metal meter bar was declared the International Prototype Meter, and all member nations received exact duplicate copies of it.

17-th General Conference of International Bureau of Weights and Measures defined the length of the meter: 1 meter - length of a path, which light passes in vacuum for 1/299792458 share of second. For a time unit the second, equal to 9192631770 periods of radiation, appropriate to transition between two hyperfine levels of a ground state of atom Cs –133, is accepted. Since light waves do not vary significantly with temperature and atmospheric conditions, this method of precise measurement may be duplicated in all parts of the world.

In the beginning of 20 century the way of slab exact manufacture of different length was invented by Johansson. Since then these gage blocks or Jo blocks (the parallel-plane end measures of length) widely are used to check the accuracy of various measuring tools and instruments. They are also used for making extremely accurate measurements, as required for special work. Precision gage blocks are available with tolerances of plus or minus 0.05 μm. In their manufacture, size is measured to light-wave accuracy with optical instruments.

Without a standard for units of linear measurement, precision measurement, mass production methods, and interchangeability of parts would not be possible. Precision **gage blocks,** as shown in Fig. 5-1, are the practical standard of measurement used in machine shops the world over.

1.3. Interchangeability

Generally, it is considered impossible to produce parts to absolute size, since some error or inaccuracy would always exist, even if a part were made to within 13 μm (0.013 mm). Therefore, in most cases, it is impractical, costly, and wasteful to machine parts to a greater degree of accuracy than that required and specified on the drawing.

Modern mass production methods require that parts be machined to size limitations which provide for interchangeability in use. The size limitations are indicated in notes or dimensions on working drawings.

The automobile is an example of the importance of parts being interchangeable. The parts are manufactured in different sections of the country and are shipped to an assembly plant where they are assembled. Other parts are sent to parts distributors where they are used later to replace worn-out parts. All of the parts, therefore, must be made within specific size limitations. These limitations allow the pares to be interchangeable in all automobiles of the make and model for which they were designed.

Chapter 2. Measuring and Layout Tools

In machine shop work, the term **laying out** means the marking of lines, centers, or circles on metal workpieces. Layout work shows size, shape, hole locations, or areas to be machined. The machinist uses many common tools for measuring and laying out parts to be machined. This unit is concerned with the study of measurement and layout tools such as rules, squares, calipers, and other common measuring tools. Some of these tools also are used for inspection work.

The machinist uses a steel **rule** to make rough measurements. Steel rules are available in lengths from 25 mm to 1 800 mm. Commonly used lengths are 150 mm, 225 mm, and 300 mm. The better ones are made of spring steel, hardened and tempered, and may have graduated measurements on one or both sides.

Metric steel rules have millimeter and half-millimeter graduations. On inch steel rules, the graduations on one side may be 8ths and 16ths and on the other 32nds and 64ths. Some rules also have graduations in l0ths, 50ths, and l00ths.

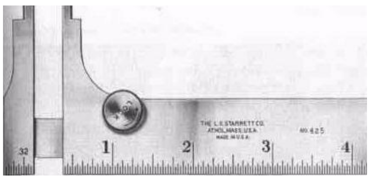
Fig. 2-1. Caliper rule.

A **caliper rule**, Fig. 2-1, is used to make rapid measurements to scale dimensions. Outside measurements of various thicknesses are read at the **out** graduation line; inside measurements are read at the graduation line.

A **rule depth gage**, Fig. 2-2, is used to measure the depth of recesses accurately and quickly. The rule is adjusted to the depth of the recess being measured and is locked with a knurled nut and friction spring. Graduations are 1/2 mm.

A **combination depth and angle gage** is shown in Fig. 2-3. In addition to measuring the depth of recesses, this tool may be used to measure or lay out angles of 30°, 45°, and 60°. Angles are set by swinging the rule so that the line on the center turret is aligned with the desired angle line. The rule graduations are in 1/2 mm.

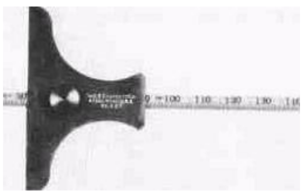
Fig. 2-2. Rule depth gage.
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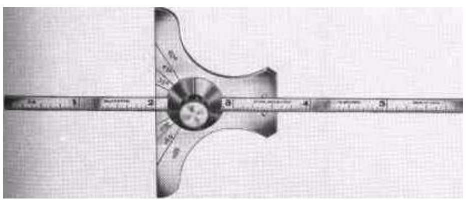


Fig. 2-3. Combination depth and angle gage.

Squares are important tools which serve several purposes, including laying out lines on parts and checking two surfaces for right-angle squareness. Several types of squares are available.

A **combination set** consists of the steel rule or blade, a square head, a center head, and a protractor head, Fig. 2-4. The square head and the protractor head are furnished with a spirit level. Although the level is not a precision level, it is an aid in measuring angles in relation to the vertical or horizontal plane. The blade and the square head together make up a **combination square.** It may be used to lay out or test 90° or 45° angles and to lay out lines parallel to an edge. It also may be used to measure the height of parts or the depth of slots or grooves.

A **protractor head** is used to test, measure, or lay out angles to within 1°

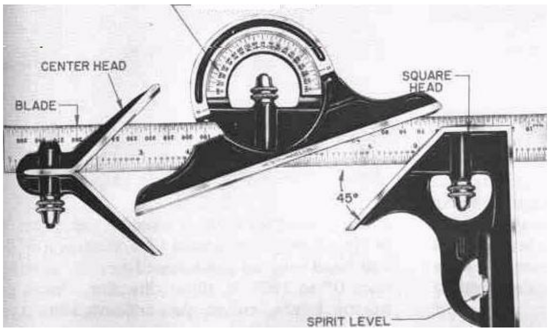


Fig. 2-4. Combination set.

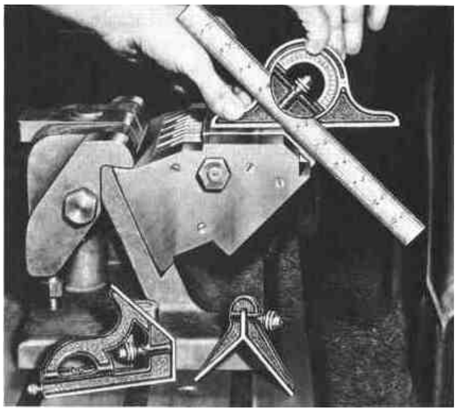


Fig. 2-5. Measuring an angle with a protractor head.

accuracy, Fig. 2-5. The head may be graduated from 0° to 90° or from 0° to 180° in either direction. Some protractor heads, called the **nonreversible type,** have a shoulder extending from one side of the blade only. A second type, the **reversible type,** has a shoulder on both sides for measuring from either side.

A **center head** is used in locating and laying out the center of round bars or other round objects. It also is used to locate the center of square bars or square objects.

Thus, a **combination set** may serve as a steel rule, height gage, bevel protractor, level, depth gage, marking gage, or plumb. The combination set is used for work which generally does not require extreme accuracy.

A **steel square** is a precision tool, which is used when extreme accuracy is required. It is used for laying out lines or for testing the squareness of two surfaces with each other. Since it has no movable parts, it is extremely accurate. Compared to other types of squares, it is expensive. It is available in several sizes. The steel square is widely used by toolmakers and machinists for checking work on both surface plates and machine tools. Although it is hardened and tempered, the steel square should be handled carefully. Dropping or severe abuse may spoil its accuracy.

A **die maker's square** is used for measuring clearance on dies. It also is used to check draft angles on foundry patterns. The blade has 1/2 mm graduations and may be set at any angle up to 10°. The angle is indicated on the beam of the square by the line on the pointer. The offset blade may be used for measuring angles in places where the straight blade cannot be used.

A **cylindrical square** is used to tell the **out-of-squareness** of work in units of 0.005 mm without the use of transfer tools. It is a true cylinder, with one end lapped perfectly square and the other end lapped at a fixed angle in relation to the sides.

To check a workpiece for squareness, the cylindrical square is placed on a surface plate with the angular end down. The base of the cylindrical square is placed against the work and rotated until light between the cylindrical square and the workpiece is shut out. The topmost dotted curve in contact with the part is read, and the number at the top of the cylindrical square indicates the out-of-squareness of the part in 2, 4, 6, 8, 10, or 12 ten-thousandths of an inch. The same reading may be obtained at two places on the circumference of the cylindrical square. Thus, the tool is self-checking.

Calipers are used chiefly for determining diameters. **A** and **B** in Fig. 2-6 are outside calipers used for measuring outside diameters. **C** and **D** are inside calipers for measuring inside diameters. **B** and **D** are called firm joint calipers, while **A** and **C** are spring type, commonly called spring **calipers. Hermaphrodite calipers** is used by machinists for scribing lines at a desired distance parallel to a flat or curved surface. It also is used for locating the center of circular objects.

Center gages, Fig. 2-7, is used principally when grinding and setting thread-cutting tools.

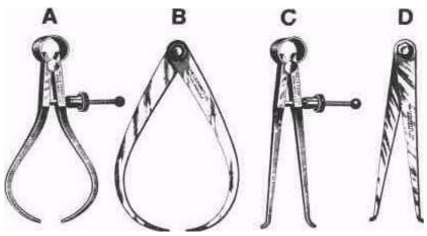


Fig. 2-6. Calipers.

Fig. 2-7. Center gage.

A **radius gage** is used for determining the radius of fillets and rounds on machine parts. It also may be used for laying out fillets and rounds. Each blade is marked with its radius. Other styles of radius gages also are available in larger sizes. A **screw pitch gage**, Fig. 2-8, is used to determine the pitch of a thread or to compare the threads of different objects, as, for example, the thread on a bolt with that on a nut. The notches on each blade are cut to match the pitch of a standard thread. Each blade is stamped with the pitch of threads it represents.

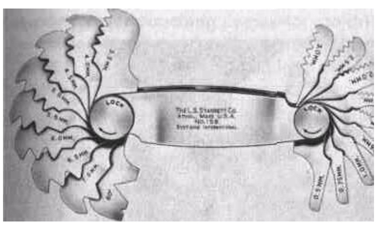


Fig. 2-8. Screw pitch gage.

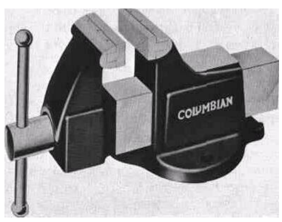


Fig. 2-9. Machinist's vise.

A **machinist's vise**, Fig. 2-9, is essential in the machine shop, toolroom, and maintenance shop. It is mounted on a work bench and holds work for various operations performed there.

In **layout work**, the machinist scribes lines to locate centers for holes and circles and to define the outline of parts. Layout work is similar to drafting. However, it generally is performed to greater accuracy, usually to 1/2 mm or closer tolerances. A **scriber** is used for marking layout lines on workpieces. Several types of scribers are available. They usually are designed so that the hardened point may be replaced when it becomes badly worn. When the point becomes dull, it may be resharpened on an abrasive stone. The point of some scribers is bent at a 90° angle for use in scribing the inside of cylindrical objects.

Dividers are used chiefly for spacing, scribing circles, and laying out work. Distances may be transferred directly from a rule to the work. Dividers are available in sizes from 51 to 305 mm and have a solid nut or a quick-adjusting, automatic-closing nut.

A **prick punch** is used to mark sharp, small points along layout lines, or to prepare for center punching before drilling. The prick punch is similar to a center punch, but has a sharper point. The included angle of the point on a prick punch usually is about 30°.

A **center punch** is a hardened tool whose point usually is ground to an included angle of about 60° to 90°. This tool is used for **center punching** preparatory to drilling. The center punch mark aids in guiding the drill so that it will drill in the desired location without drifting to the side.

A **surface plate** is a heavy plate of steel, cast iron, or granite, with a precision flat surface. The surface is machined, ground, and scraped for extreme flatness. It provides the necessary flat surface for making the accurate measurements required in precision layout and inspection work. Surface plates are available in sizes ranging from 305 mm square to 1 219 x 3 048 mm or larger. They are expensive and should be used carefully. A surface plate never should be hammered or struck, since the smallest nick or dent will affect its accuracy. A thin film of oil should be applied to the surface of iron or steel plates to prevent rust when not used regularly.

A **straight-edge** is made of cast iron. This tool is portable and is placed on machined surfaces to check them for flatness or straightness. The straight-edge has a scraped surface which should be protected with a wooden cover when not in use. Straight-edges are available in various lengths ranging from 457 mm to 4 572 mm.

Fig. 2-10. Using a telescoping gage.

Several types of **V-blocks** are used in doing layout work and in making machining set-ups. V-blocks may be used singly or in pairs. The work is held in the V-block with adjustable screws.

Telescoping gages are used to gage inside diameters or distances, as shown in Fig. 2-10. The gage is equipped with a plunger which is under spring tension when retracted. When a part is gaged, the tool first is inserted with the plunger retracted. The plunger is released and the tool positioned for accurate gaging. The knurled nut on the handle is then tightened, and the gage is extracted from the part. The distance across the ends of the gage then is measured with a micrometer caliper. Telescoping gages are available in sizes which measure distances from 8 mm to 150 mm.

Small-hole gages are used for gaging the size of small holes and narrow slots. They are available in sizes which measure the diameters of holes or recesses from 3 mm to 13 mm. In using the gage, select a gage of the proper size, insert the gage in the hole or recess to be measured, and turn the knurled screw on the handle to expand the ball-shaped end to size. (A very slight pressure or drag will be felt when the screw tension is right.) Then use a micrometer caliper to measure the diameter of the ball end.

The edges on **adjustable parallels** are precision ground and parallel with each other. The parallels are made in various sizes and are adjustable in width within a range for each size. A screw is provided to lock the tool firmly at the adjusted width. Adjustable parallels may be used to gage the width of grooves or slots. The size of the parallel then may be measured with a micrometer. Adjustable parallels may be used as spacers for part location in accurate assembly work, or they may be set at a specific size to serve as gage blocks. They also may be used in machine vises for setting work at the proper height for drilling, shaping, grinding, or milling, although solid parallels are usually used for this purpose.

Chapter 3. Micrometers and Vernier Measuring Tools

3.1. Micrometers

Micrometers are the precision measuring tools most commonly used by machinists. They are available in a variety of types and sizes, but the most common is the outside micrometer caliper. Figure 3-1 shows a 0-25 mm outside micrometer of traditional design. Several micrometers of more recent design include the following:

1. Indicating micrometer, which can also be used as a comparator for quickly checking parts in quantity;
2. Direct reading micrometer, which provides a numerical display of the micrometer reading;
3. Dual reading micrometer, which reads both in metric and in English measurement;
4. All electronic micrometer, which has a motorized spindle and electronic digital readout.

A micrometer often is called a «mike». Plain metric micrometers measure accurately to one-hundredth of a millimeter (0.01 mm). Some metric micrometers are equipped with a vernier which makes it possible to measure accurately to one-thousandth of a millimeter (0.001 mm).

The parts of a traditional micrometer are shown in Fig. 3-1. The principal parts are the **frame,** the anvil, the **spindle** with a precision screw thread, the **sleeve** which is also called a **barrel or hub,** and the **thimble.**

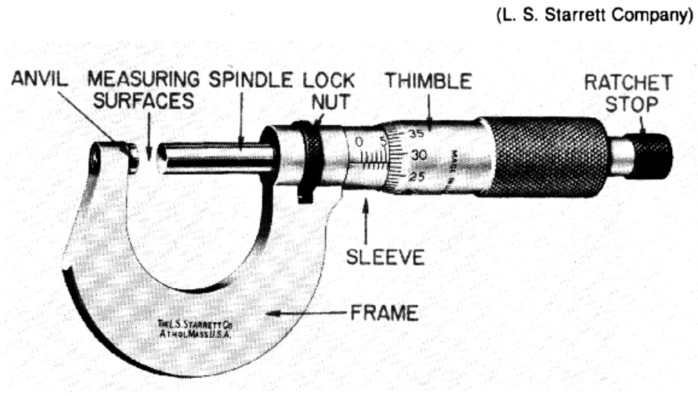


Fig. 3-1. Principle micrometer parts

The **ratchet** and the **lock nut** are convenient accessories which are available on some micrometers. With the use of the ratchet, a consistent pressure can be applied on the spindle when measurement is made, regardless of who uses the tool. A consistent pressure is important in making accurate measurements. Without the ratchet, one must develop the right feel for accurate measurement. The lock nut locks the spindle in position after measurements are made.

Many kinds and sizes of micrometers have been developed for various measuring applications. The following are the principal kinds:

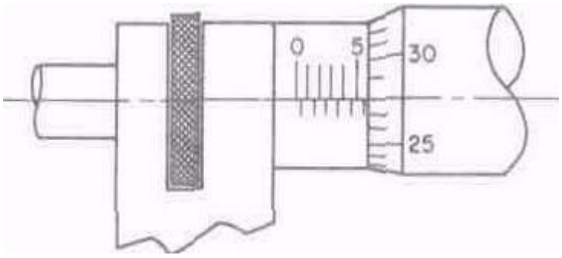


Fig. 3-2. Micrometer reading of 5.78 mm.

Outside micrometers, also called **micrometer calipers,** Fig. 3-1, are used for measuring outside diameters or thickness. Outside micrometers are available in various sizes which are limited to 25 mm measuring ranges, such as 0-25 mm, 25-50 mm, and so on. Large micrometers are available for measuring within various 25 mm ranges. Some of them may be used for measurements simply by changing and installing the appropriate anvil.

The metric micrometer reading of 5.78 mm illustrated in Fig. 3-2, is obtained as follows:

1. Upper sleeve reading (whole millimeters)	5.00 mm
2. Lower sleeve reading (half millimeters)	0.50 mm
3.Thimble reading (hundredths of a millimeter)	0.28 mm
Total Reading	5.78 mm

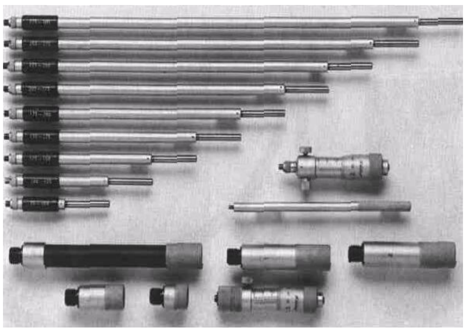


Fig. 3-3. Tubular inside micrometer.

Inside micrometers are used for measuring inside diameters, parallel surfaces, or other inside dimensions. There are several types and sizes of inside micrometers available. A small inside micrometer caliper may be used to measure within the range from 5 mm to 25 mm. Figure 3-3 shows a **tubular inside**

micrometer. It may be used for measuring inside diameters from 40 mm to 300 mm, in range increments of 13 mm. Measuring rods are added to either or both ends of the micrometer head to increase its range. A tubular inside micrometer can be used for measuring large diameters. Mike hole gages are used for accurately measuring the diameter of relatively small holes. They are available in various size ranges from 6 mm to 200 mm. Setting rings are available in various sizes for testing and setting the accuracy of inside micrometers.

Fig. 3-4. Measuring depth of a shoulder with a depth micrometer.

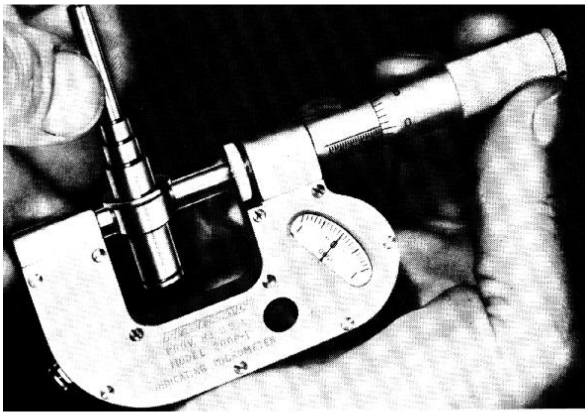


Fig. 3-5. Measuring diameter of a shaft with a dial indicating micrometer.

Depth micrometers are used for measuring the depth of holes, grooves, shoulders, and projections, as shown in Fig. 3-4. The measuring range for depth micrometers can be increased in multiples of 25 mm, as desired, by installing interchangeable measuring rods.

Thread micrometers are used to measure the pitch diameter of screw threads (Fig. 10-5).

Vernier metric micrometers and **dial indicating micrometer** (Fig. 3-5) are designed for measuring to 0.001 mm or 0.002 mm. The vernier scale is placed on the sleeve above the usual graduations, but note that the millimeter and half millimeter graduations are both marked on the lower part of the sleeve. This allows the vernier to be read without having to twist the micrometer.

3.2. Vernier Measuring Tools

A vernier is a short rule or scale that is mounted on a measuring instrument so that its graduations subdivide the divisions on the main scale. Verniers increase the degree of precision which can be obtained from both linear and angular measuring tools. The French mathematican, Pierre Vernier, invented the vernier scale about 1630 A.D.

A **vernier caliper** is made up of a graduated beam with a fixed measuring jaw, a movable jaw wich carries a vernier scale and a mechanism for making fine adjustments, Fig. 3-6.

Vernier calipers are capable of making both outside and inside measurements. In

(L. S. Starrett Company)

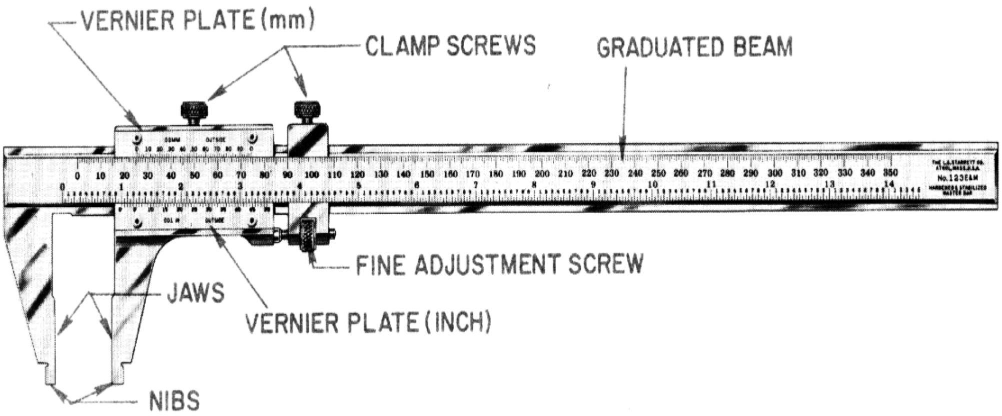


Fig. 3-6. A vernier caliper

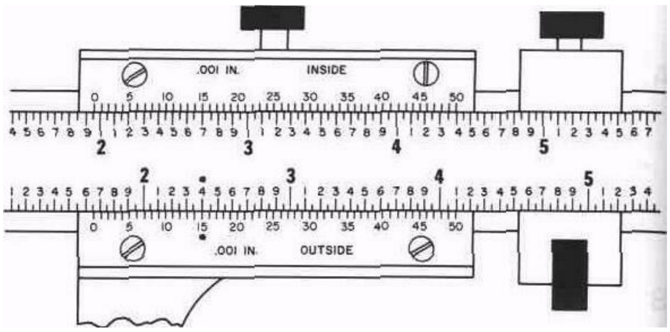


Fig. 3-7. A 50-division inch vernier reading of 1.665".

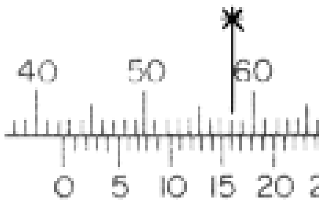


Fig. 3-8. Reading of 42.16 mm on metric vernier scale.

addition, some are also provided with a depth measuring rod. When inside measurements are made the size is read on the single scale for both outside and inside measurements but it is necessary to add a number which is marked on the jaw. The reading in Fig. 3-7 is explained as follows:

1. The zero on the vernier plate is between the 1" and 2" marks on the beam, making 1.000".
2. The zero on the vernier plate has passed the 6 on the beam, indicating 6/10ths or 0.600".
3. The zero on the vernier plate has passed the midpoint division between the 6th and 7th marks, adding 1/20th"or 0.050".
4. The 15th line on the vernier matches a line on the beam, adding <u>0.015</u>"
5. Total reading is 1.665".

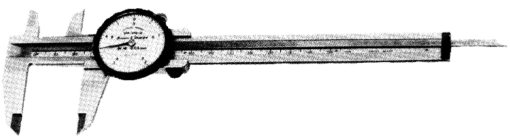


Fig. 3-9. A vernier caliper with a dial indicater and a depth measuring rod.

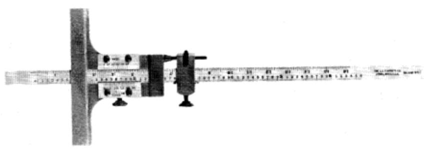


Fig. 3-10.A vernier depth gage.

Many kinds and sizes of vernier calipers have been developed for various measuring applications. The following are the principal kinds: vernier depth gage (Fig. 3-10), vernier height gage (Fig. 3-11), vernier gear tooth caliper (Fig. 11-8).

A **universal bevel protractor** equipped with a vernier, Fig. 3-12, measures angles accurately to 5 minutes or one-twelfth of a degree. This tool also is called a **vernier protractor.** It may be used to lay out, measure, or check angles. A 150 or 300 mm blade may be inserted in the graduated dial and locked in position with the blade clamp nut. The blade and dial are swiveled to the angle desired, and the dial is locked with the dial clamp nut. An acute-angle attachment is provided for use in measuring angles of less than 90°.

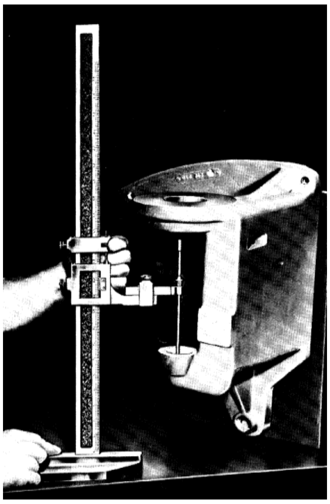


Fig. 3-11. A vernier height gage.

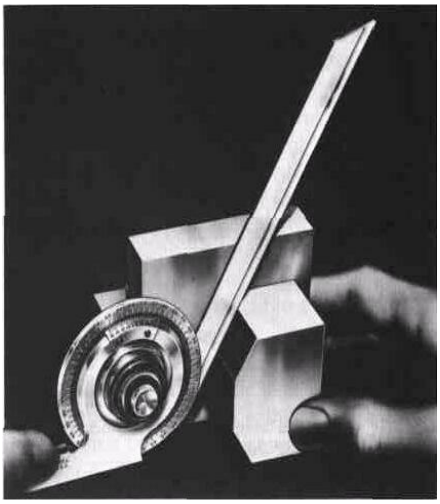


Fig. 3-12. A universal bevel protractor.

The protractor dial is graduated 360°, reading in whole degrees from 0 to 90, 90 to 0, 0 to 90, and 90 to 0. Each 10 degrees is numbered, and a long graduation divides each 5 degrees. The vernier plate is graduated with 12 spaces. Thus, each line here represents 5 minutes or one-twelfth of a degree. Every third line on the vernier plate is numbered to represent 15, 30, 45, and 60 minutes. Both the protractor dial and the vernier plate have numbers in both directions from zero.

When angles are read in whole degrees, the zero line on the vernier plate coincides with a graduation line on the protractor dial. Also, the graduation for 60 minutes will coincide exactly with a graduation on the dial. When angles are read which are not in whole degrees, the following procedure is used. Note how many degrees can be read from the zero line on the dial up to the zero line on the plate. Then, reading in the **same direction** (and this is important) note the number of minutes indicated by the line on the vernier which coincides exactly with a line on the dial. Add this amount to the number of whole degrees. Remember, each graduation line on the vernier represents 5 minutes.

Chapter 4. Dial Indicating Instruments

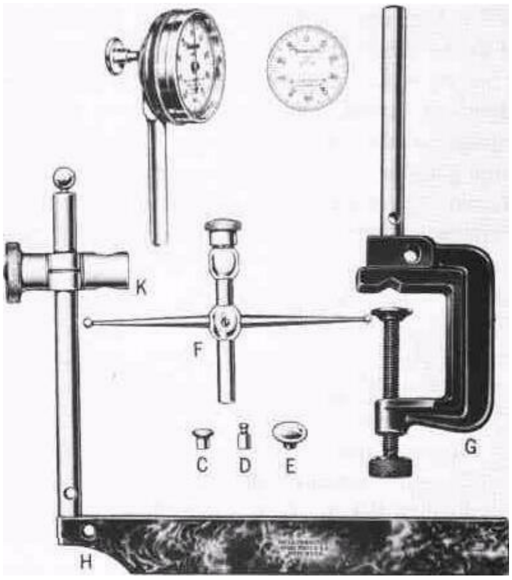


Fig.4-1. Universal dial indicator set. C, D and E are contact points; F is hole attachment; G- clamp; H- tool post holder; K-sleeve.

Universal dial indicator sets are used widely. With the accessories, Fig. 4-1, provided in this set, the dial indicator may be mounted on a surface gage for use on a surface plate or on a machine table. It may be mounted in the tool post of a lathe. It is particularly useful for aligning work in a four-jaw chuck, as in Fig. 4-2, where it is being used with a hole attachment. It is also provided with a clamp so that it may be attached wherever needed for special applications.

A full range of accessories permits dial indicators to be held in surface gages, vernier height gages, and in lathe tool posts. With the aid of a magnetic base, the universal dial test indicator can be quickly attached to any machine. It can also be held in a drill chuck or collet in a jig borer or vertical milling machine for use in checking alignment of vises, fixtures, or workpieces.

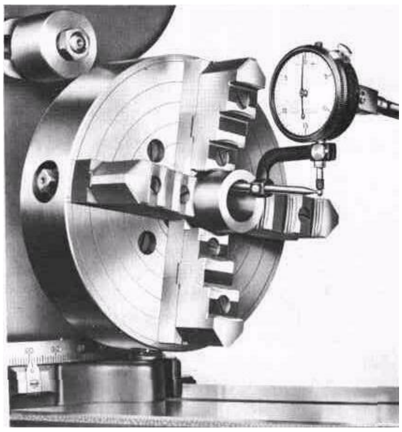


Fig. 4-2. Hole attachment permits accurate internal tests with dial indicator.

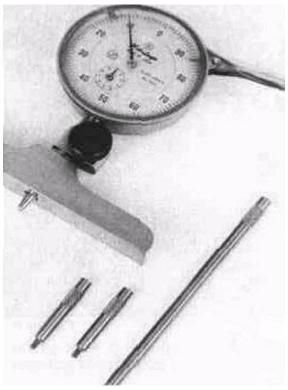


Fig. 4-3. Dial indicating depth gage with extension points.

Numerous special gages are equipped with dial indicators. These gages are used widely for determining whether parts are within required size limits.

Dial indicating depth gages, Fig. 4-3, are used for gauging or testing the depth of holes, slots, shoulders, recesses, and keyways. Extension points increase the measurement size, measuring depths at which the gage may be used.

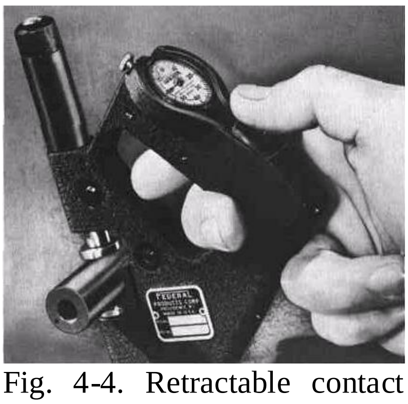


Fig. 4-4. Retractable contact snap gage.

Dial indicating hole gages, Fig. 4-5, are used to gauge or test holes for size, taper, out-of-roundness, or other irregular conditions. They are available in a wide range of sizes.

Dial indicating snap gages are used for gauging diameters of parts to determine whether they are within the size limits specified. In use, the gage is

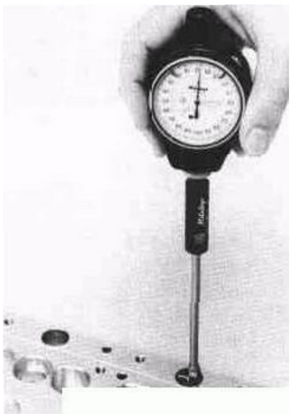


Fig.4-5. Dial indicating hole gage.

snapped over the diameter of the part being gauged. Snap gages are efficient for checking parts which are produced in large numbers, and they are available in a wide range of sizes. Each gage may be used for measuring within a 25 mm range, such as 0-25 mm, 25-50 mm, etc. Size is set by adjusting the frame itself with the knurled wheel near the indicator. The gage may be used at an inspection bench where it can be mounted in a bench stand or right at the machine. **A retractable contact snap gage** is shown in Fig. 4-4. The contactor point is opened with a button located conveniently for thumb operation. This type of gage is available in sizes which gauge, within certan ranges, from 0-112 mm.

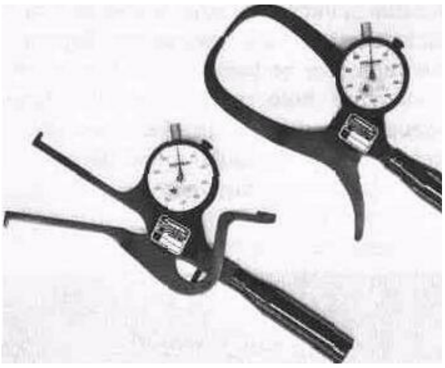


Fig. 4-6. Dial indicating caliper gages.

Dial indicating caliper gages, Fig. 4-6, have revolution counters which make it possible to measure directly through their complete range of 75 mm. Calipers of this type are available with 0.02 mm graduations.

Setting discs and setting rings are used for checking and setting indicating caliper gages, snap gages, hole gages, comparators, and other types of gages. Setting discs and rings are available in many sizes.

Chapter 5. Gages

5.1. Gage Blocks

Gage blocks are recognized throughout the world as a practical **reference standard** for measuring length. They are used in precision measurement laboratories, inspection departments, toolrooms, and machine shops for calibrating and setting many types of inspection gages, measuring tools, and measuring instruments. Gage blocks, therefore, are the connecting link between the national standard of measurement and measurement in the shop.

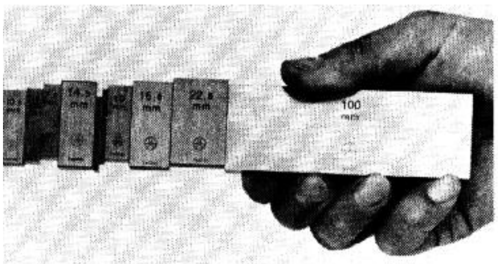


Fig. 5-1. Gage blocks wrung together.

Manufacturers produce gage blocks in relation to master gage blocks which must be accurate to plus or minus 0.05 microns

(0.05 μm). In many cases, their master gage blocks are accurate to within plus or minus 0.025 microns. These master gage blocks are checked periodically for accuracy with an **interferometer.** This instrument measures gage blocks in units of light wave length to within a fraction of the required tolerance. (Light wave length is not significantly affected by changes in temperature and atmospheric conditions.)

Gage blocks may be used together to make up various gage block combinations of greater length (Fig. 5-1). When gage blocks are properly combined, they are said to be **wrung** together. Their surfaces are so flat and smooth that when properly wrung, they stick together as though magnetized. Some have been known to support a dead weight of over 90 kilograms in tension when wrung together.

Gage blocks normally are classified according to three accuracy classifications: Class 1 (formerly AA), Class 2 (formerly A+), and Class 3 (a blend of former A and B).

Some manufacturers produce gage blocks with the special classification, **laboratory master gage blocks,** to tolerances of half that for Class 1 blocks. These gage blocks, as well as regular **master** gage blocks (Class 1), are intended for special purposes in temperature-controlled gauging and measurement laboratories. They are used for experimental work, research work, and as grand master gages for measurement and inspection of other gages.

Class 2 gage blocks often are called **inspection gage blocks.** They are used primarily for inspection of finished parts and for inspecting and setting working gages. They also may be used as masters in inspection departments and toolrooms.

Class 3 gage blocks often are called **working gage blocks.** They are used for many applications requiring accurate measurement throughout the shop. They are used on surface plates for accurate layout, on machines for setting cutting tools accurately, and for ordinary inspection work.

Gage blocks are available in **sets** or as **individual blocks**. Sets are available with from 5 to more than 100 gage blocks. A commonly used standard set of 88 metric gage blocks consists of:

3 blocks: 0.5, 1.00, 1.005 mm;

9 blocks: 1.001 through 1.009 by 0.001 mm;

49 blocks: .1.01 through 1.49 by 0.01 mm;

17 blocks: 1.5 through 9.5 by 0.5 mm;

10 blocks: 10 through 100 by 10 mm.

In building up a specific gage block combination, use as few blocks as possible. You should know the sizes of the gage blocks in the set available. Start by selecting gage blocks which will remove the right-hand figure in the decimal size which you wish to build. Then select blocks which will remove the next right-hand decimal, and so on. Several examples will illustrate the procedure:

Example 1. Build up 48.357 mm: 1.007+1.05+1.3+5+40=48.357 mm.

Angle gage blocks are precision tools used for accurate measurement of angles. A set of 16 may be used for measuring 356,400 angles in steps of 1 second up to 99 degrees. These angle gage blocks may be wrung together in various combinations, just as rectangular gage blocks are. Angle blocks can also be wrung together for inspection of a simple angle on a part.

Angle blocks are manufactured in two accuracy classifications: **laboratory master angle gage blocks,** which are the most expensive, have an accuracy classification of plus or minus 1/4 second; **toolroom angle gage blocks** have an accuracy of plus or minus 1 second.

Angle gage blocks are so designed that they may be combined in plus or minus positions. One end of each angle block is marked **plus,** while the opposite end is marked **minus.** Several examples will illustrate how the blocks may be combined in either position, thus forming different angles. The plus end of a 15° angle block may be wrung together with the plus end of a 5° block to form a 20° angle. Wringing the plus end of the 15° block together with the minus end of the 5° block forms an angle of 10°. The angle blocks may be wrung together to form angles in steps of degrees, minutes, seconds, or in any combination of these units.

5.2. Limits Gages

Go and not-go gages are also called **limits gages.** Because they have two gauging surfaces or points, they sometimes also are called **double gages.** One gauging surface is used for testing the upper size limit; the other, the lower size limit. The common types of limits gages include snap gages, ring gages, and plug gages.

The principles involved in testing parts with go and not-go gages are illustrated in Fig. 5-2. The figure shows how cylindrical parts are checked with a limits **snap gage.** The upper gauging point is the **go** point, while the lower is the **not-go** point. If the part is too small it will pass between both the upper and the lower gauging points. If the part is satisfactory it will pass the upper point, but hangs on the lower gauging point. If the part is too large it will hangs on the upper gauging point. Most such gages are labeled "go" and "not go", meaning that the part will **go** at the first point, but it will **not go** at the second.

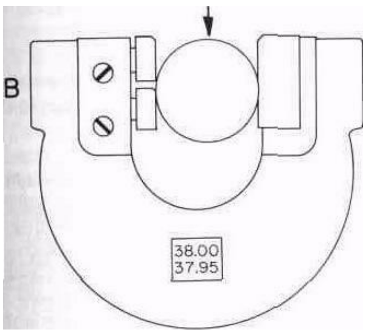


Fig. 5-2. Limits snap gage. Part is satisfactory.


Fig. 5-3. Adjustable limits snap gage.

Limits gages never should be forced under high pressures when checking parts. The pressures applied to the gauging surfaces should be slight. Limits gages have contact surfaces which are hardened, precision-ground, and lapped. Although they are designed for accuracy and wear resistance, they should be handled and used carefully.

One of the most widely used limits gages is the **snap gage**, Fig. 5-2. This caliper-type gage is used to gauge thicknesses, lengths, and outside diameters. Snap gages are available in several styles and sizes. That shown in Fig. 5-3 is an adjustable limits gage. It has one stationary anvil and two button anvils which are adjustable. The outer button is set to the go size, and the inner button to the not-go size. The procedure for using this type of snap gage was described earlier and is shown in Fig. 5-2.

The size limits on a snap gage may be checked with gage blocks. Limits snap gages may be supplied by the manufacturer set and **sealed** at specified size limits, or **unset** and **unsealed.** When preset, the adjustment screws usually are sealed with sealing wax and the size limits are stamped on the gage.

Snap gages of special types are fitted with special anvils, buttons, or rolls for gauging special forms or external threads. A roll thread snap gage is used to check the size limits for the pitch diameter of screw threads.

Three types of **ring gages** commonly are used for checking the external diameters of parts: plain ring gages, taper ring gages, and thread ring gages.

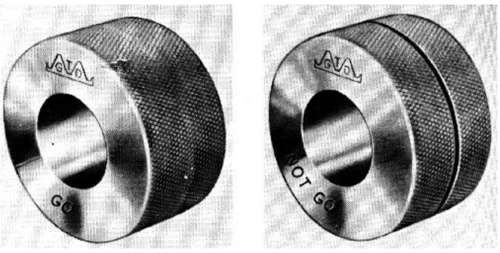


Fig. 5-4. Plain ring gages.

Plain ring gages (Fig. 5-4) are designed in the form of a cylindrical ring. These gages are used for checking the external diameters of straight round parts. The **not-go** ring (identified by the groove around the outside diameter)) is used to check the minimum size limit. The **go** ring is used to check the maximum size limit. The **go** ring will pass over a part which is within specified size limits with little or no interference. The **not-go** ring will not pass over the work. If both rings pass over, the part is undersize. If neither does, the part is oversize.

Taper ring gages (Fig. 5-5) are used for checking the size and the amount of external cylindrical taper on parts. Cylindrical taper ring gages are used for checking the taper shanks on drills, reamers, lathe centers, and other machine accessories.

In using a taper ring gage, first draw three equally spaced chalk lines lengthwise on the external tapered surface which is to be checked. Then slip the ring over the external taper by applying a light pressure for good surface contact.

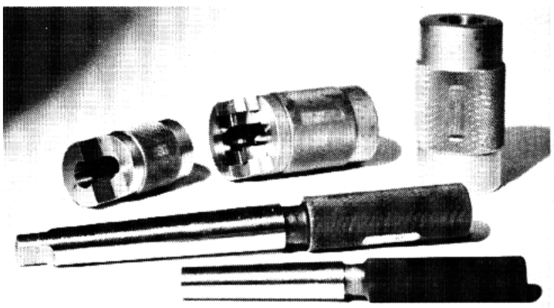


Fig. 5-5. Cylindrical taper ring and plug gages.

Rotate the ring forward and backward a small amount while continuing to apply light pressure. Remove the gage and observe the external tapered surface. If all three chalk lines have been uniformly rubbed and distributed, the taper is correct. If the chalk lines have been rubbed harder at one end than at the other, then a correction should be made on the taper. The correct amount of taper should be established before the part is machined to finished size. Size may be determined by measuring the small diameter with a micrometer or by noting the distance to which the taper enters the ring gage.

Thread ring gages of the go and not-go type are used for checking the pitch diameter of external threads (Fig. 10-6, A).

Three basic types of **plug gages** are used for checking the accuracy of holes: plain cylindrical plug gages, cylindrical taper plug gages, and thread plug gages. Plug gages of special types also are available for use in checking holes of special shape, such as square holes and rectangular holes.

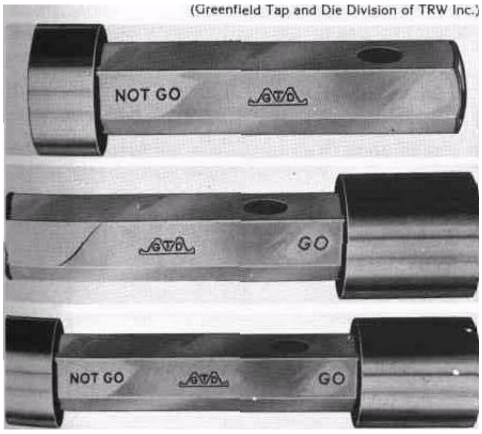


Fig. 5-6. Plain cylindrical plug gages.
Top — single-end gages.
Bottom — double-end gage.

Plain cylindrical plug gages (Fig. 5-6) are accurate cylinders which are used for checking the size limits of straight cylindrical holes. The gage is provided with a handle for convenient use. The gage may be either the single-end type or the double-end type. The go gage is longer than the not-go gage; it should enter the hole with little or no interference. If great pressure is required, the hole is undersize and is not acceptable. The not-go gage should not enter the hole. If it does, the hole is too large.

A **progressive-type plug gage** has both the go and the not-go gages on the same end of the handle. It is efficient for checking through holes, but it cannot be used for shallow, blind holes.

Tapered cylindrical plug gages are used for checking the size and amount of taper in tapered cylindrical holes in drill sleeves, in machine tool spindles, and in adapters for use with taper shank tools. In using the tapered cylindrical plug gage to check a tapered hole, use the same procedure described above for using a tapered cylindrical ring gage. The hole is finished to size when the gage enters to the end of the plug or to a depth indicated by a mark on the gage.

Thread plug gages are used for checking the size limits for the pitch diameter of internal screw threads (Fig. 10-6, B).

5.3. Snape Measurement

Where only close approximation of shape is required, template type gages such as screw pitch and radius gages, and custom-made templates are often adequate. Templates, however, cannot provide the degree of accuracy required for precision measurement of thread and gear tooth profiles, cams, miniature instrument parts, and other small parts calling for precise shape. Highly precise inspection of shape can be obtained with **toolmaker's microscopes** and **optical comparators**.

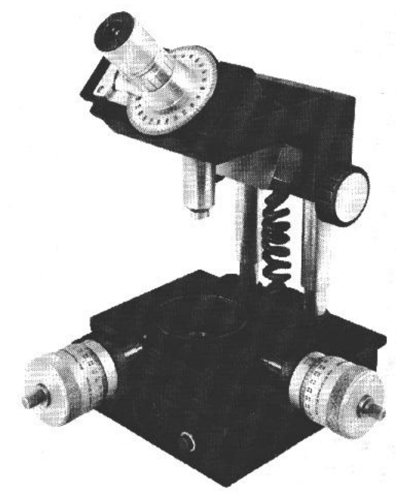


Fig. 5-7. Toolmaker's microscope.

The **toolmaker's microscope** is designed especially for use in toolmaking. Images can be enlarged from 10 to 200 times, depending on equipment and job requirements. Images are not reversed as in ordinary microscopes. The platform, called a **stage,** on which workpieces are mounted for measurement, can be moved both crossways and sideways with vernier micrometer accuracy. A vernier protractor built into the microscope head, reads to five minutes of arc through the magnifying lens. The microscope optics provide 90° cross hairs and concentric circles with diameters from 0.25-5 mm by 0.25 mm. An attachment allows cylindrical workpieces to be supported between centers. Direct angular measurement, thread dimensions, radius, hole roundness, and even squareness can be easily checked with a high degree of accuracy.

An **optical comparator** projects an accurately enlarged shadow-like profile of the part being measured into a screen. Here, both its size and shape are compared to a master drawing or template. Magnification of 10, 20, and 50 diameters are commonly used. A stage provides sideways and crossways movement of the part to vernier micrometer accuracy. Angular measurement can be directly from templates or from a protractor attachment. Optical comparators are especially useful for checking small, irregularly shaped objects which cannot be easily measured with conventional tools. Flexible parts, such as springs and rubber or plastic objects, which would distort under the pressure of conventional measuring tools, can easily be inspected. The accuracy of thread forms, gear tooth shape, and cam profiles are easily checked in this manner.

5.4. Sine Bar

The **sine bar** is a precision tool used to establish or check angles to within one minute of arc. Sine bars must be used in conjunction with some true surface, such as a surface plate, from which accurate measurements may be taken. They are used to establish and check angles for layout work and inspection work. They may be used for making machining setups such as those often required for surface grinding. They also may be used to accurately determine unknown angles.

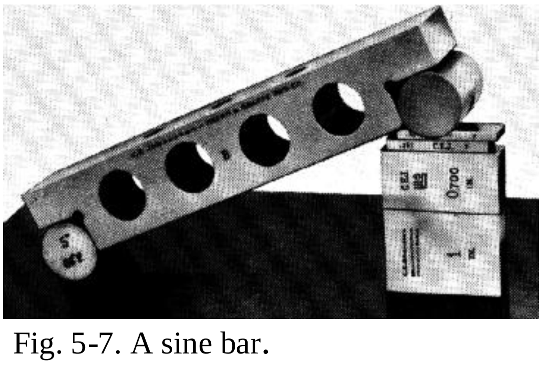


Fig. 5-7. A sine bar.

The sine bar is a hardened and precision-ground steel bar which has two hardened and precision-ground steel rolls of the same diameter attached. The edge of the bar is parallel with the center line of the rolls. For convenient mathematical calculation, it is available in lengths which provide a distance of 127 mm, 254 mm, or 508 mm (5", 10", or 20") between the centerlines of the rolls. The sine bar is named after the sine trigonometry function, which states that the sine of an angle is equal to the length of the side opposite the angle divided by the length of the hypotenuse. Either unknown angle α can be found when the side **a** opposite the angle, and the hypotenuse **c** are known: $\sin\alpha = a/c$.

Chapter 6. Inspection of Surface Form, Disposition of Surfaces and Surface Finish

6.1. Inspection of Surface Form

The analysis of geometrical accuracy of a part is based on the comparison of the actual surface form with the nominal surface form (the surface which does not have errors of surface and has ideal form.)

The tolerance of deviation of the form and disposition of surfaces depends on geometrical accuracy of a part and a value of a part length or length of specified site. The geometrical accuracy of a part depends on a purpose of a part.

16 degrees of accuracy are established for each kind of the tolerance of the form and disposition of surfaces. The numerical value of the tolerances one degree to another is changed with a factor of increase 1.6. Degrees of accuracy 9 - 10 are made by turning, shaping and milling; degrees of accuracy 7 - 8 are made by middle accurate turning, shaping and milling; degrees of accuracy 5 - 6 are made by grinding.

The following levels of relative geometrical accuracy are established depending on a ratio between a size tolerance and tolerances of a form or surfaces dispositions: A - normal relative geometrical accuracy (tolerances of a form or surfaces dispositions make approximately 60 % of a size tolerance); B - increased relative geometrical accuracy (the ratio is 40 %); C - high relative geometrical accuracy (the ratio is 25 %.)

Tolerances of the form or surfaces dispositions specify only for the functional or technological reasons (when they should be less the size tolerance.)

Deviation from a straightness (the term «unstraightness» is not recommended) is the greatest deviation of an actual surface form from an adjoining straight. The adjoining straight is a straight which touch surface but does not cross the actual profile of the part, Fig. 6-1. The adjoining straight have to set so that the deviation from one to the farthest point of actual profile will be least within the limits of measured section. This measurement is made in any direction and on any site of the part if it is not specified. Measured deviation is compared with a tolerance of deviation from a straightness, Table 6-1.

Deviation from a flatness (the term «unflatness» is not recommended) is the greatest deviation of an actual surface form from an adjoining plane. The adjoining plane is a plane which touch surface but does not cross the actual profile of the part. The adjoining plane have to set so that the deviation from one to the farthest point of actual profile will be least into the limits of measured section.

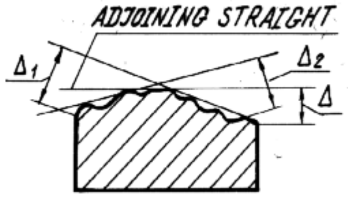


Fig. 6-1. Adjoining straight: $\Delta<\Delta_1<\Delta_2$

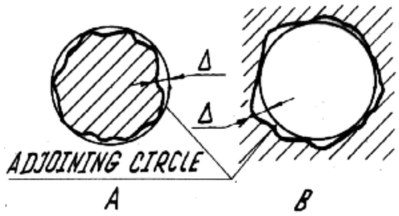


Fig. 6-2. Adjoining circle: A- draw-out; B- draw-in.

Table 6-1. Tolerances of deviations from a straightness and from a plane, μm.

Size, mm	Degree of accuracy							
	5	6	7	8	9	10	11	12
0 - 10	1.6	2.5	4	6	10	16	25	40
10 - 16	2	3	5	8	12	20	30	50
16 - 25	2.5	4	6	10	16	25	40	60
25 - 40	3	5	8	12	20	30	50	80
40 - 63	4	6	10	16	25	40	60	100
63 - 100	5	8	12	20	30	50	80	120
100 - 160	6	10	16	25	40	60	100	160
160 - 250	8	12	20	30	50	80	120	200
250 - 400	10	16	25	40	60	100	160	250
400 - 630	12	20	30	50	80	120	200	300
630 - 1000	16	25	40	60	100	160	250	400
1000 - 1600	20	30	50	80	120	200	300	500
1600 - 2500	25	40	60	100	160	250	400	600
2500 - 4000	30	50	80	120	200	300	500	800
4000 - 6300	40	60	100	160	250	400	600	1000
6300 - 10000	50	80	120	200	300	500	800	1200

Deviation from a roundness (the term «unroundness» is not recommended) is the greatest deviation of an actual surface form from an adjoining circle. The adjoining circle is a draw-out circle for an external surface (shaft) and a draw-in circle for an internal surface (hole.) The draw-out circle is a circle, which has minimum diameter but does not cross an actual profile of a part, Fig. 6-2. The draw-in circle is a circle, which has maximum diameter but does not cross the actual profile of the part, Fig. 6-3. This measurement is made on any cross section of the part if it is not specified. Measured deviation is compared with a tolerance of deviation from a round. The tolerance of deviation from a round depends on an geometrical accuracy of a part and a diameter size. The geometrical accuracy of a part depends on a purpose of a part, Table 6-2.

Deviation from a cylinderness (the term «uncylinderness» is not recommended) is greatest deviation of an actual surface form from an adjoining cylinder. The adjoining cylinder is a draw-out cylinder for an external surface (shaft) and a draw-in cylinder for an internal surface (hole.) The draw-out cylinder is the cylinder which has minimum diameter but does not cross the actual profile of the part. The draw-in cylinder is the cylinder which has maximum diameter but does not cross the actual profile of the part.

Table 6-2. Tolerances of deviation from a round, a cylinder, a profile of a longitudinal section, μm.

Size, mm	Degree of accuracy							
	5	6	7	8	9	10	11	12
0 - 3	2	3	5	8	12	20	30	50
3 - 10	2.5	4	6	10	16	25	40	60
10 - 18	3	5	8	12	20	30	50	80
18 - 30	4	6	10	16	25	40	60	100
30 - 50	5	8	12	20	30	50	80	120
50 - 120	6	10	16	25	40	60	100	160
120 -250	8	12	20	30	50	80	120	200
250 - 400	10	16	25	40	60	100	160	250
400 - 630	12	20	30	50	80	120	200	300
630 - 1000	16	25	40	60	100	160	250	400
1000 - 1600	20	30	50	80	120	200	300	500
1600 - 2500	25	40	60	100	160	250	400	600

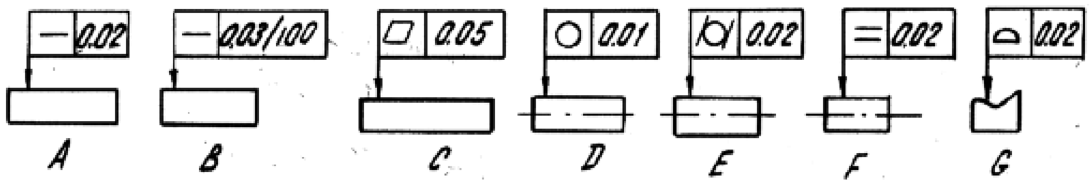


Fig. 6-3. Deviations: A - deviation from a straight on the full length of the part; B - deviation from a straight on the length of 100 mm; C - deviation from a plane; D - deviation from a rounder; E - deviation from a cylinder; F - deviation from a profile of longitudinal section; G - deviation of a form of specified surface.

6.2. Deviations of Disposition of Surfaces

Deviation of disposition of surface or profile is a deviation of actual surface (profile) from it base disposition. Deviations of examine surfaces and base surfaces must be excepted. These surfaces is replaced by adjoining surfaces and axes, planes of symmetry and centers of adjoining surfaces are considered as axes, plates of symmetry and centers of actual surfaces.

Deviation from parallelism of plates is the difference of the largest and the smallest distances between the examined and base surfaces into the limits of specified section, Fig. 6-4.

Deviation from parallelism of axes in the space is the geometry sum of deviations from parallelism of projections of axes in two mutually perpendicular plates. One of these plates passes through the base axis and point of crossing of the examined and base axes, Fig. 6-4.

Deviation from perpendicular of surfaces can be measured as in millimeters as in angular value. In the most cases this deviation is measured in millimeters using a dial indicator with tool post holder and sleeve, Fig. 6-4.

Deviation from coaxiality is greatest distance between an axis of a examined surface of rotation on a length of the normalized site, Fig 6-4. A tolerance zone of coaxiality is area in space limited by the cylinder, which diameter is equal to the tolerance of coaxiality of diametrical expression T, and the axis coincides with a base axis, Fig. 6-4.

Deviation from symmetricity to a base plane is greatest distance between a plane of symmetry of examined surface and a base plane of symmetry in the limits of the normalized site, Fig. 6-4.

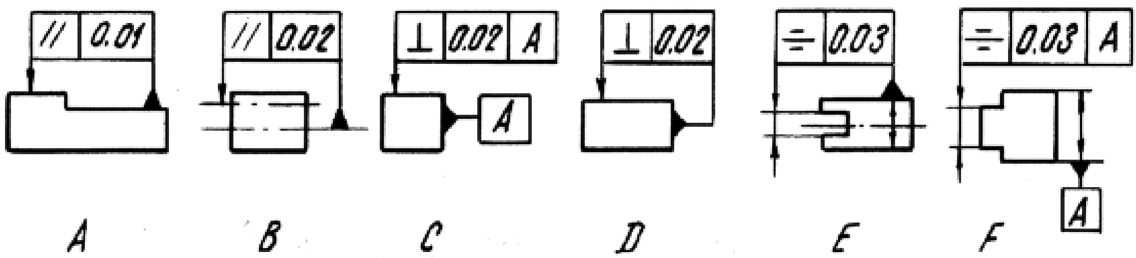


Fig. 6-4. Deviation of disposition of surface or profile: A - deviation from parallelism of plates; B - deviation from parallelism of axes in the space; C - deviation from perpendicular of surfaces; D - deviation from perpendicular of surfaces; E - deviation from symmetry to a base plane; F - deviation from symmetry to a base plane.

Item deviation is the greatest deviation of a real disposition of an element (its center, axis or plate of symmetry) from its nominal disposition, Fig. 6-4.

Radial palpitation of a rotation surface relatively a base axis grows out of joint display of a deviation from roundness of examined section and a deviation of its center relatively the base axis. It is equal to a difference of the greatest and least distances from points of a real profile of rotation surface to a base axis in section, perpendicular this axis. The complete radial palpitation is determined within the limits of the normalized site, Fig. 6-5.

Table 6-3. Tolerances of coaxiality, symmetricity, axes cross and radial palpitation.

Base size, mm	Degree of accuracy							
	5	6	7	8	9	10	11	12
	Tolerance, µm							
0 - 3	5	8	12	20	30	50	80	120
3 - 10	6	10	16	25	40	60	100	160
10 - 18	8	12	20	30	50	80	120	200
18 - 30	10	16	25	40	60	100	160	250
30 - 50	12	20	30	50	80	120	200	300
50 - 120	16	25	40	60	100	160	250	400
120 - 250	20	30	50	80	120	200	300	500
250 - 400	25	40	60	100	160	250	400	600
400 - 630	30	50	80	120	200	300	500	800
630 - 1000	40	60	100	160	250	400	600	1000
1000 - 1600	50	80	120	200	300	500	800	1200
1600 - 2500	60	100	160	250	400	600	1000	1600
Note: 1. Tolerances of coaxiality, symmetricity and axes cross are showed in diametrical expression. 2. Base size is examined size.								

Face palpitation (complete) is a difference of the greatest and least distances from points of a real face surface to a plane, which is perpendicular to a base axis. Sometimes the radial palpitation is determined in face surface section by the cylinder of specified diameter d, Fig. 6-6.

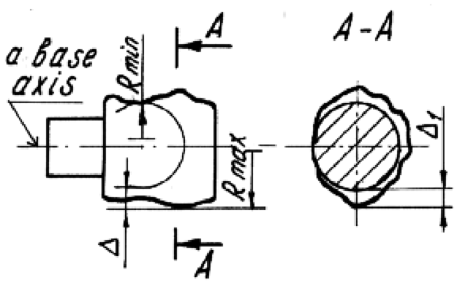


Fig. 6-5. Complete radial palpitation of a rotation surface relatively a base axis.

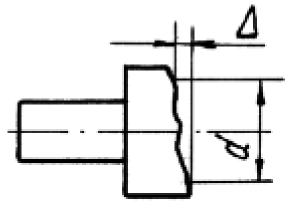


Fig. 6-6. Face palpitation relatively a common axis.

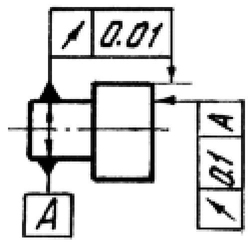


Fig. 6-7. Radial and face palpitations relatively a common axis.

Radial palpitation of a rotation surface relatively a common axis grows out of joint display of a deviation from roundness of examined section and a deviation of its center relatively the common axis, Fig. 6-7.

Dependent tolerance is the variable tolerance of disposition of surface or form. Its minimal value is specified on the drawing and which is supposed to be exceeded on value appropriate to a deviation of the actual size of a surface of a detail from a maximum material limit (the greatest limiting size of the shaft or the least limiting size of the hole).

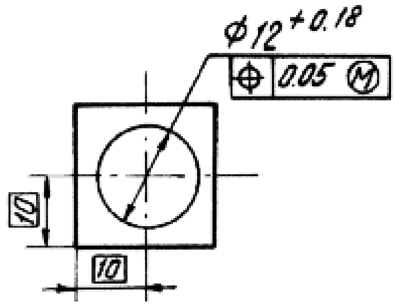


Table for explanation of dependent tolerance

Possible actual diameter of the hole, mm	12.0	12.01	12.02	12.10	12.18
Full value of dependent item tolerance, mm	0.05	0.06	0.07	0.15	0.23

Fig. 6-8. Designation of dependent item tolerance and table for it explanation.

The dependent tolerances of disposition of surface and form nominate basically when it is necessary to supply assembly of matting details simultaneously on several surfaces. The dependent tolerances are usually controlled by complex gages.

6.3. Designation of Tolerances of the Form and of the Surface Disposition

Kind of the tolerance of the form and the disposition of surface are designated on the drawing by marks (graphic symbols). The mark (Table 6-4) is entered in a framework on the first place, the numerical meaning of the tolerance in millimeters is written on the second place, Fig. 6-9. If it is necessary the letter designation of base (base surface) is written on the third place. The framework is united to an element, to which the tolerance is concerned, by continuous line, which comes to an end by a pointer. If the tolerance concerns to an axis or plane of symmetry the connecting line should be continuation of a size line. If the tolerance concerns to a common axis or plane of symmetry the connecting line is drawn to a common axis.

Before numerical meaning of the tolerance it is necessary to write a symbol: Ø - the tolerance is specified by its diameter; R - the tolerance is specified by its radius; T - the tolerance is specified in diameter expression (for the tolerances of symmetry, crossing of axes, form of the specified surface, item); T/2 - the same tolerances are specified in radius expression; the word "sphere" and symbols Ø or R if the tolerance zone is spherical. If the tolerance concerns to a site of a surface of specified length (area), its meaning is underlined after the tolerance, separating by a inclined line. If the tolerance is specified to all length and to a site of a surface of specified length the tolerance of specified length is underlined below the tolerance of all length.

The total tolerances of the form and of the disposition of surface, for which special symbols are not established, are designated by marks of the compound tolerances: the tolerance symbol of the disposition of surface is marked first, then the tolerance symbol of the form is marked. The base is designated by a black triangle, which is united to a framework of the tolerance. In most cases a base is designated by the letter. In this case it is necessary: sections and kinds are designated first by the letters in alphabetic order without the misses and recurrences, and then the bases are designated, continuing the alphabet.

Table 6-4. Marks of tolerances of form and surface disposition.

Group of tolerances	Kind of tolerance	Mark
Tolerances of form	Tolerance of straightness	—
	Tolerance of planeness	□
	Tolerance of roundness	O
	Tolerance of cylinderness	/O/
	Tolerance of profile of longitudinal section	═
Tolerances of surface disposition	Tolerance of parallelness	//
	Tolerance of perpendicularity	⊥
	Tolerance of inclination	∠
	Tolerance of coaxiality	◎
	Tolerance of symmetricyty	÷
	Tolerance of item	⊗
	Tolerance of axes cross	×
Total tolerances of form and surface disposition	Tolerance of radial palpitation	↗
	Tolerance of face palpitation	↗
	Tolerance of full radial palpitation	↗↗
	Tolerance of full face palpitation	↗↗
	Tolerance of form of specified profile	⌒
	Tolerance of form of specified surface	⌓

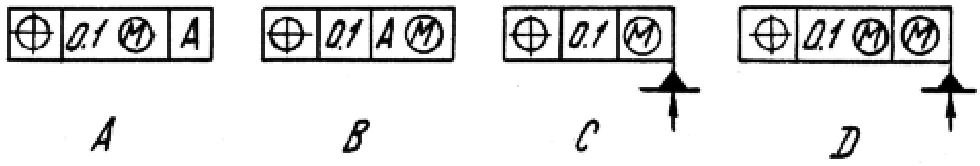


Fig. 6-10. Designation of dependent tolerances.

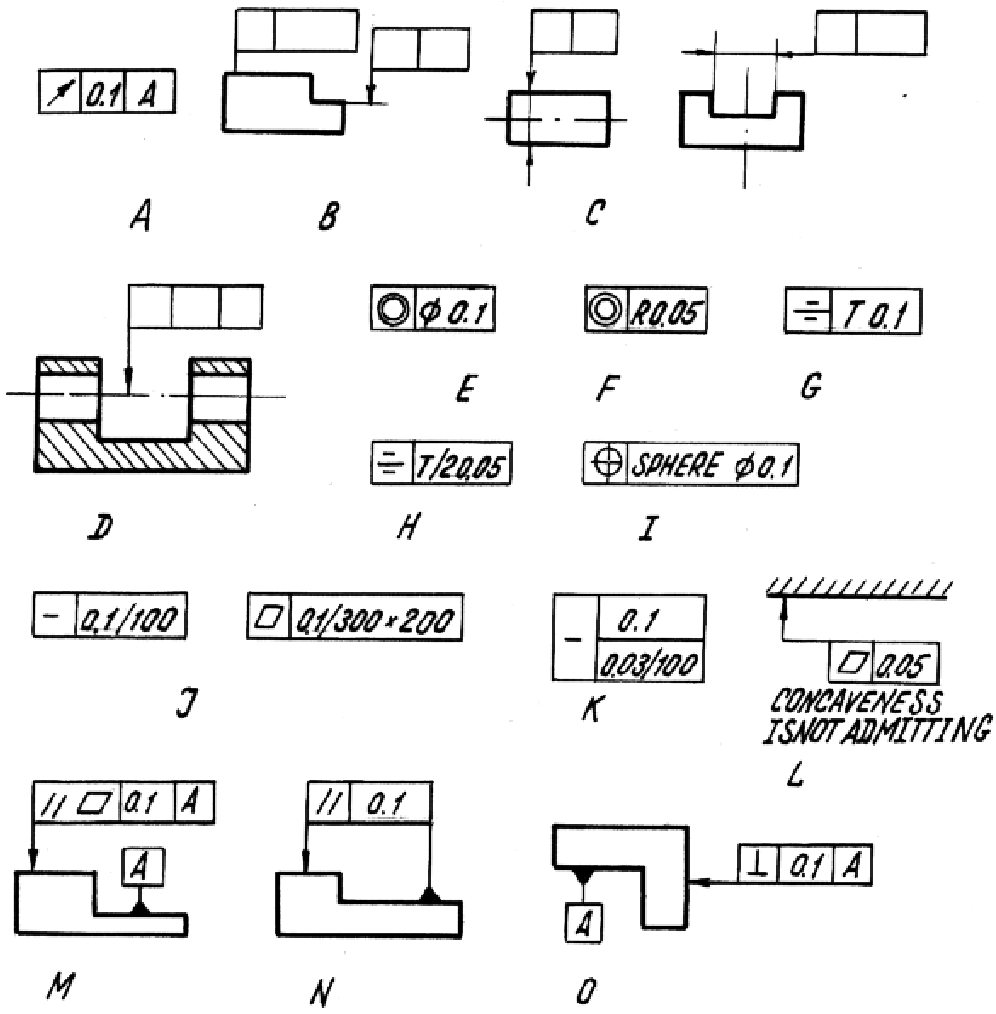


Fig. 6-9. Designation of tolerances of the form and of the disposition of surface.

The dependent tolerance is designated by the letter M in a circle. It settles down: after numerical meaning of the tolerance, if the dependent tolerance is connected to the actual sizes of a surface, Fig. 6-10A; after a letter designation of base (Fig. 6-10B) or without a letter designation of base (Fig. 6-10C) in the third part of a framework, if this tolerance is connected to the actual sizes of a base surface; after numerical meaning of the tolerance and after a letter designation of base or without a letter designation of base if the dependent tolerance is connected to the examined and to the base element, Fig. 6-10D.

6.4. Measurements of Deviations of the Form and Surfaces Disposition

The deviations of the form are measured by universal and special measuring devices. Special measuring surface plate from cast iron and firm stone breeds (in the basic granite), **straight-edges**, master straight-edges, **parallel-plane** precision gage blocks, optic-mechanical devices are used for these purposes. For measurement of a deviation from straightforwardness a parallel-plane ruler (plate) is put on a checked surface. This plate carries out a role of adjoining straight line. A tool post holder (rack) with an indicator is put on this plate, and a tip of the indicator concerns the checked surface. The tool post holder with the indicator is moved on the plate along a detail. A difference of the greatest deviations of an indicator hand is a deviation from straightforwardness.

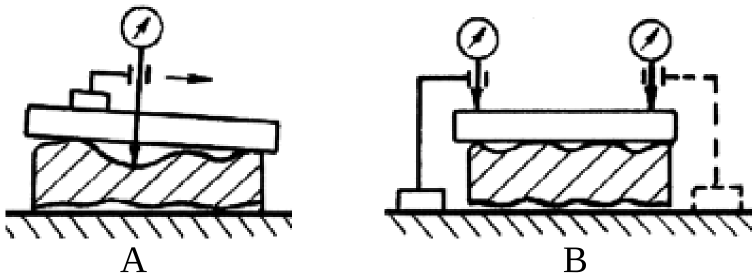


Fig. 6-11. Schemes of measurements of deviations from straightness (A) and parallelism (B).

For measurement of deviations from **parallelism** the similar scheme is used, but detail and rack are put on a parallel-plane plate, and the tip of the indicator concerns a surface of the plate. A deviation from parallelism is a difference of the greatest and least deviations of the indicator hand at moving along a detail. The deviation from a plane can be supervised on a stain of contact.

The beam of the laser is used in optic-mechanical devices as an exemplary straight line. The coordinate measuring machine is applied to exact measurements deviations from parallelism. The tip of the machine scans checked surface. The coordinates of the surface are entered in computer, which creates virtual adjoining plane to virtual created scanned surface. The maximal deviation of these virtual surfaces is a deviation from a plane. The round-gauge works similarly, the record is done in polar coordinates.

Indicators with prisms and special support are used for measurement of deviations from a roundness. One-point and two-points devices can not measure the deviation from a roundness with odd number of sides. The basing on a prism (V-block) is used in this case. Indications of the device are multiplied by the reproduction factor, which depends on the number of sides and the prism corner.

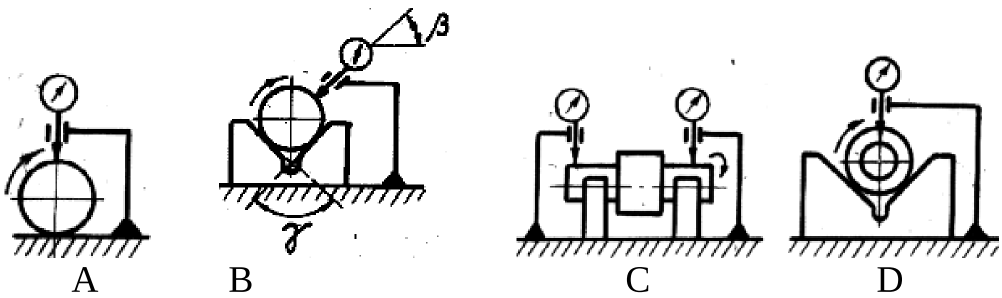


Fig. 6-12. Schemes of measurements of deviations from roundness (A, B), coaxiality(C) and radial palpitation (D).

The fastening of shaft at centers is used for measurement of a deviation from roundness, radial palpation and coaxiality. Two indicators are mounted on the checked and base surfaces simultaneously. The radial palpation relatively a base surface is the difference between simultaneous indications of indicators. The deviation from coaxiality of holes in a case details is made with the help of mandrel, mounted in these holes. Set square or the vertical moving of the indicator on the rack is used for measurement of deviations from perpendicular.

6.5. Control of Surface Finish

The smoothness of a machined surface is determined by a combination of factors involved in the machining process. Some of the most significant of them include: type and condition of cutting tool used, rigidness of the machine and setup, type of material being cut, depth of cut, rate of feed, cutting speed, and kind of cutting fluid used. Figure 6-13 shows surface characteristics involved in measurements of surface finish quality.

The type of surface required for a given product is determined by the designer. Such items as bearings, gear teeth, and pistons must have controlled surface quality. For example, the surface on a bearing can be excessively rough or smooth. If it is too rough, it will wear rapidly, resulting in limited life. If it is too smooth, it will not have adequate provision for oil pockets and it will be difficult to keep lubricated, thus again resulting in limited life.

To require a high surface quality where it is not necessary is expensive and unprofitable. Where detailed specifications concerning surface quality are not indicated, it means that the surface normally produced by that particular kind of machine operation is adequate.

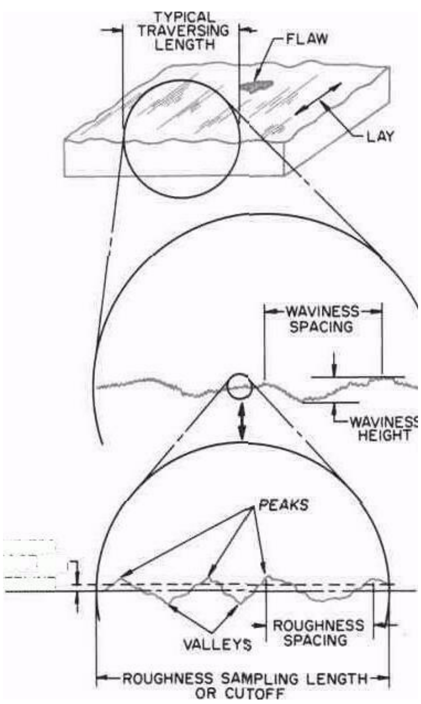


Fig. 6-13. Surface characteristics.

The machinist must produce machined surfaces which meet specified standards of quality, and must be able to interpret the surface quality specifications indicated on drawings and blueprints. The machinist also must know how to determine whether machined surfaces meet surface quality specifications.

All machined surfaces, including those which appear to be very flat and smooth, have surface irregularities. Under high magnification, scratches or grooves in the form of peaks and valleys are revealed. These irregularities may or may not be superimposed on larger waves. Such complex factors as height, width, and direction of surface irregularities determine surface texture. They are specified with standard symbols on drawings.

The following terms related to surface texture have been defined by the American National Standards Institute (ANSI) and have been extracted from ANSI B46.1-1978 with the permission of the publisher:

Surface texture. Repetitive or random deviations from the nominal surface which form the three-dimensional topography of the surface. Surface texture includes roughness, waviness, lay, and flaws.

Profile. The profile is the contour of a surface in a plane perpendicular to the surface, unless some other angle is specified.

Centerline. The centerline is the line about which roughness is measured and is a line parallel to the general direction of the profile within the limits of the sampling length, such that the sum of the areas contained between it and those parts of the profile which lie on either side are equal.

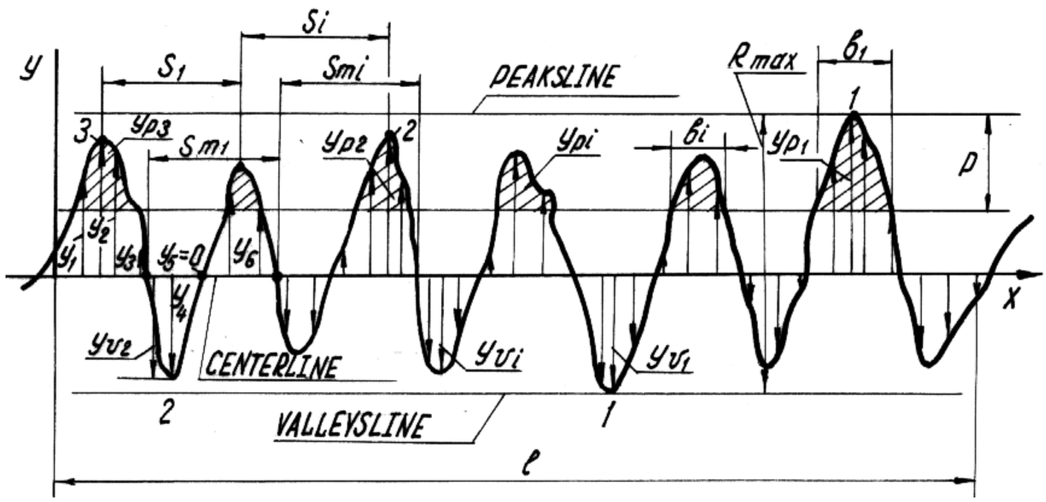


Fig. 6-14. Short section of hypothetical profile divided into increments.

Roughness. Roughness consists of the finer irregularities in the surface texture, usually including those irregularities which result from the inherent action of the production process. These are considered to include the traverse feed marks and other irregularities within the limits of the roughness sampling length.

Roughness average. Roughness average is the arithmetic average of the absolute values of the measured profile height deviations taken within the sampling length and measured from the graphical centerline, Fig. 6-1. Roughness average is expressed in micrometers.

$$Ra = \frac{1}{n}\sum_{i=1}^{n}|yi| = \frac{1}{l}\int_{0}^{l}|y(x)|dx, \qquad (6.1)$$

Roughness sampling length. The roughness sampling length **l** is the sampling length within which the roughness average is determined. This length is chosen, or specified, to separate the profile irregularities which are designated as roughness from those irregularities designated as waviness. Roughness sampling length is measured in millimeters. Standard values are (mm): 0.08 (Ra<0.025μm), 0.25 (0.025<Ra<0.4), 0.8 (0.4<Ra<3.2), 2.5 (3.2<Ra<12.5), 8 (12.5<Ra<100).

Height of the profile roughness on ten points. Height of the profile roughness on ten points is a sum average absolute values of the 5 highest peaks and of the 5 depthest valleys taken within the sampling length and measured from the graphical centerline. Roughness is expressed in micrometers.

$$Rz = \frac{1}{5}\left[\sum_{i=1}^{5}|yi| + \sum_{j=1}^{5}|yj|\right], \qquad (6.2)$$

Roughness spacing. Roughness spacing is the average spacing between adjacent peaks of the measured profile within the roughness sampling length. Roughness spacing average is expressed in millimeters.

$$S = \frac{1}{n}\sum_{i=1}^{n} Si, \qquad (6.3)$$

Roughness spacing on the centerline. Roughness spacing on the centerline is the average spacing measured on the graphical centerline of the measured profile within the roughness sampling length. Roughness spacing average is expressed in millimeters.

$$Sm = \frac{1}{n}\sum_{i=1}^{n} Smi, \qquad (6.4)$$

Relative profile length. This parameter is measured on the specified level **p** relatively the centerline, and **p** is calculated in percents (5%, 10%, 15%, 20%, 25%, 30%, 40%, etc.) relatively R_{max}. Relative profile length is expressed in %.

$$tp = \frac{1}{l}\sum_{i=1}^{n} bi, \qquad (6.5)$$

Cutoff. The cutoff is the electrical response characteristic of the roughness average measuring instrument which is selected to limit the spacing of the surface irregularities to be included in the assessment of roughness average. The cutoff is rated in millimeters.

Waviness is the more widely spaced component of the surface texture. Unless otherwise noted, waviness is to include all irregularities whose spacing is greater than the roughness sampling length and less than the waviness sampling length. Waviness may result from such factors as machine or work deflections, vibration, chatter, heat treatment, or warping strains. Roughness may be considered superimposed on a «wavy» surface.

Waviness height. The waviness height is the peak-to-valley height of the modified profile from which the roughness and flaws have been removed by filtering, smoothing, or other means. The measurement is to be taken normal to the normal profile within the limits of the waviness sampling length and expressed in millimeters.

Waviness spacing. The waviness spacing is the average spacing between adjacent peaks of the measured profile within the waviness sampling length.

Lay. Lay is the direction of the predominant surface pattern, ordinarily determined by the production method used.

Flaws. Flaws are unintentional irregularities which occur at one place or at relatively infrequent or widely varying intervals on the surface. Flaws include such defects as cracks, blow holes, checks, ridges, scratches, etc. Unless otherwise is specified, the effect of flaws shall not be included in the roughness average measurements. Where flaws are to be restricted or controlled, a special note as to the method of inspection should be included on the drawing or in the specifications.

Application of Surface Finish Symbols

Surface quality is designated with a surface finish symbol and ratings. The symbol is similar to a check mark, but with a horizontal extension line added, Fig.6-15. The long leg of the check-like symbol is to the right as the drawing is read. If only the roughness height is designated, the horizontal extension line may be omitted.

The point of the surface symbol is located on the line indicating the surface specified. It also may be located on an extension line or leader pointing to the surface specified, as in Fig. 6-15. Symbols used with the surface symbol to indicate lay are shown in Table 6-1.

Surface quality ratings for various characteristics such as roughness, waviness, and lay are positioned specifically in relation to the surface symbol. The relative location of these specifications and ratings is indicated in Table 6-2.

Designations for Roughness Height

The roughness average, according to the ISO standard, is expressed in micrometers as the simple **arithmetical average** (AA) deviation, measured normal to the centerline. In previous standards the roughness height was expressed in micrometers as the **root mean square average** (RMS) deviation, measured normal to the centerline. Certain instruments are equipped with a selector switch for selecting either the RMS or the AA reading.

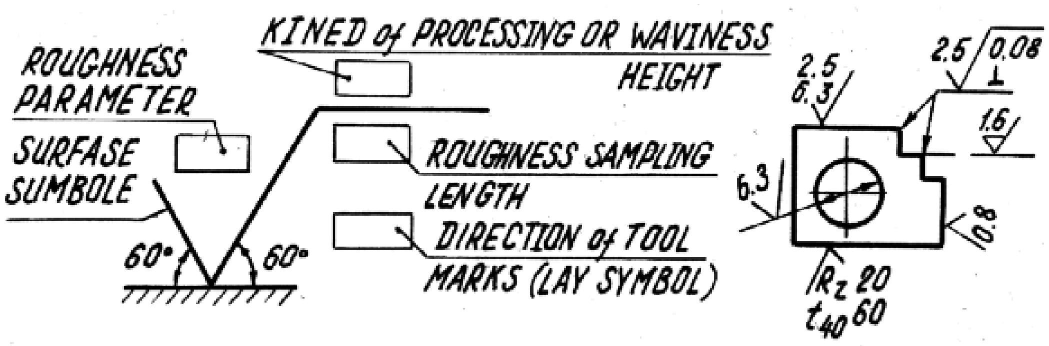


Fig. 6-15. Application of surface finish symbols.

Table 6-1. Lay symbols.

Lay symbol	Meaning
$=$	Lay approximately parallel to the line representing the surface to which the symbol is applied.
$\perp$	Lay approximately perpendicular to the line representing the surface to which the symbol is applied.
$\times$	Lay angular in both directions to the line representing the surface to which the symbol is applied.
M	Lay multidirectional.
C	Lay approximately circular relative to the center of the surface to which the symbol is applied.
R	Lay approximately radial relative to the center of the surface to which the symbol is applied.
P	Lay particulate, nondirectional, or protuberant.

Roughness measuring instruments calibrated for AA values will indicate approximately 11 percent lower for a given surface than those calibrated for RMS average values. However, because the absolute limit of roughness for satisfactory functioning of a surface is indefinite, many manufacturers adopt AA ratings without changing the RMS values indicated on older drawings. For most surface measurement applications, the difference between the two values is of no consequence.

In order to eliminate error or confusion in the use of various stylus instruments, standards are included in ANSI B46.1-1978. For instruments indicating a numerical value only, a spherical-tip stylus with a 10 micrometer (400 microinch) radius tip is standard. The accuracy of instruments for surface roughness measurement should be checked periodically by measuring a precision reference specimen.

Table 6-2. Application of surface texture values to symbol.

Symbol	Meaning
	Roughness average rating is placed on the left of the long leg. The specification of only one rating shall indicate the maximum value and any lesser value shall be acceptable. Specify in micrometers.
	The specification of maximum and minimum roughness average values indicates permissible range of roughness. Specify in micrometers.
	Maximum waviness height rating is the first rating placed above the horiszontal extension. Any lesser rating shall be acceptable. Specify in millimeters. Maximum waviness spacing rating is the second rating placed above the horiszontal extension and to the right of the waviness height rating. Any lesser rating shall be acceptable. Specify in millimeters.
	Material removal by machining is required to produce the surface. The basic amount of stock provided for material removal is specified on the left of the short leg of the symbol. Specify in millimeters.
	Removal of material is prohibited.
	Lay designation is indicated by the lay symbol placed at the right of the long leg.
	Roughness sampling length or cutoff rating is placed below the horizontal extension. When no value is shown, 0.8 mm applies. Specify in millimeters.
	Where required maximum roughness spacing shall be placed at the right of the lay symbol. Any lesser rating shall be acceptable. Specify in millimeters.

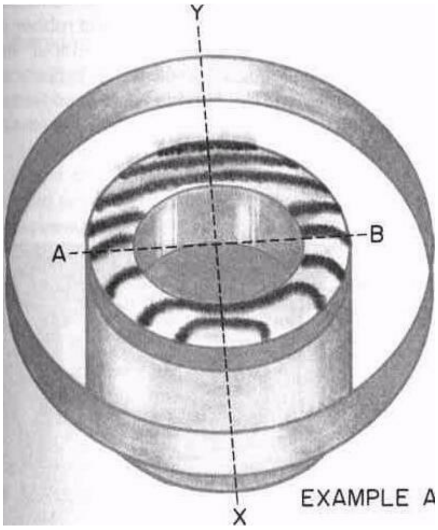
6.6. Flatness Measurement

Flatness is very important to the accuracy of gage blocks and certain other gages, micrometers, parallels, and other precision tools. It is equally important to proper functioning of flat metal-to-metal assemblies which must be leak-free without use of sealing materials. This degree of flatness goes well beyond the capacity of dial indicators or other conventional measuring tools to detect.

Interferometry, using the interference of two beams of light for measurement, is capable of measuring flatness to 25 millionths of a millimeter. Optical flats utilize interferometry and are relatively inexpensive tools used for measuring flatness.

Optical flats are quartz or glass lenses that have been polished accurately flat on one or both surfaces. When both surfaces are polished, they are known as **optical parallels.** They range in size from 25 mm diameter, 13 mm thick, to 300 mm diameter, 70 mm thick, and larger. A choice of three grades provides accuracies of 25, 50, or 100 millionths of a millimeter.

The precision surface of the optical flat facing the work is both transparent and light reflecting. Because of this, light waves striking this surface are effectively split into two light waves, one passing through and one reflecting back. When two reflected light waves cross or interfere with each other, they become visible as dark bands, Fig. 6-16. This occurs when the surface of the optical flat and the surface being measured are out of parallel by one half of a wavelength, or multiples thereof, of the light being used. Helium light, with a predominant wavelength of 589 millionths of a millimeter is ordinarily used with optical flats. Thus, each dark band provides a measuring unit of 294.5 millionths of a millimeter.



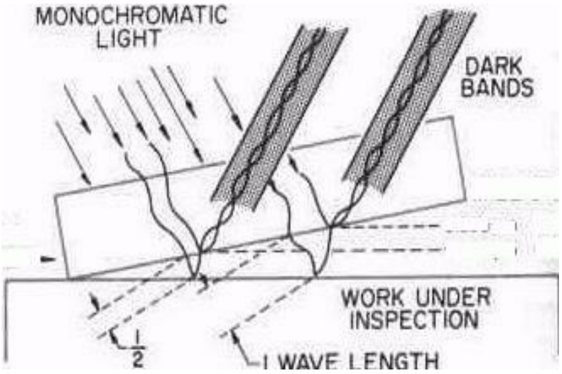


Fig. 6-16. Pattern of dark bands produced by a rotating seal that is not flat. The greatest variation from flatness is across the approximate center, where the center is higher than the outer edges by 1 '/2 scale lines or 442 millionths of a millimeter.

Flat surfaces out of parallel in one axis produce a pattern of dark bands at right angles to that axis. Diagonal bands result when the surfaces are out of parallel in both axes. Convex or concave surfaces produce a pattern of curved bands.

Chapter 7. Methods of Measurements. Tolerances and Fits

Production operations would not be possible without measurements. In most cases it is necessary to reduce errors in measurements. Errors in measurements depend on the accuracy of the measuring equipment. These errors can result from the following:

1. Inherent errors in the measuring instrument.
2. Errors in the « master gage » used to set the instrument.
3. Errors resulting from temperature variations and different coefficients of linear expansion of instrument and part being gaged.
4. Errors due to the « human element » of the inspector.
5. Errors of a measurement method.

7.1. Methods of Measurements

All methods of measurements can be divided into 2 groups: the methods of absolute measurements and methods of relative measurements.

When we deal with the method of absolute measurements we obtain a dimension

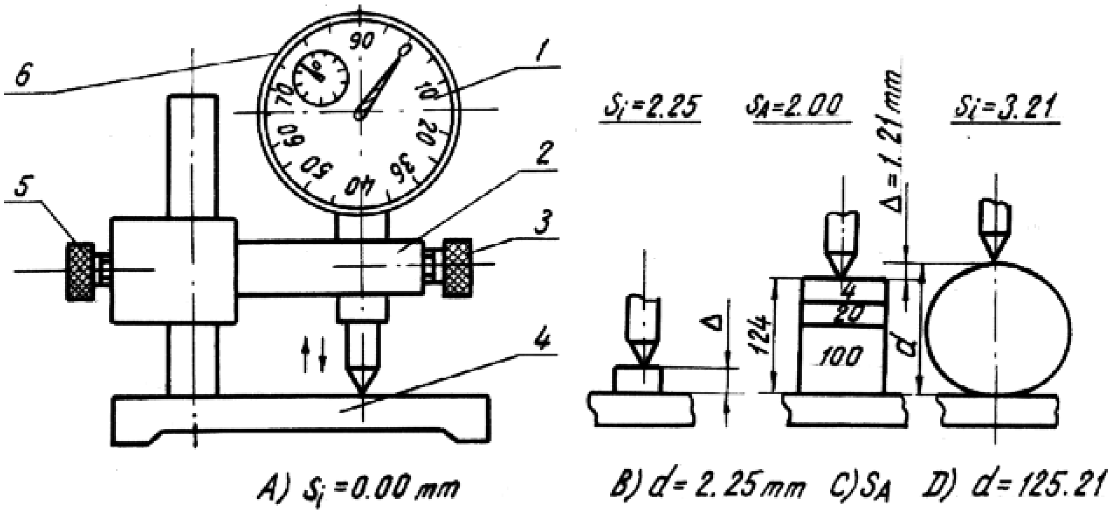


Fig. 7-1.Measuring a part with a dial indicator.

at once, reading the scale. For example, when we use a ruler, vernier caliper, micrometer. This method is available when we measure a small part with a dial indicator as shown in Fig. 7-1A. First of all, we must lower a bracket 2 together with the indicator 6 in order to touch a part-table 4 (further we shall call it the table) and indicator hands must turn a little. It is better for us if the short hand is set against a stroke under the number 0 (zero). After that we have to fasten the bracket. In order to make reading easier, we have to turn the main face 1 of the indicator until the stroke of 0 is set against the long hand. We have adjusted the instrument, and after that we must not turn the face of the indicator. Our reading is 0.00 mm.

In order to measure a part, we must carefully lift the spindle and place our part on the table. Then we lower the spindle until it touches the surface of the part. We are reading the result. For example: the short hand shows 2, and the long hand shows 25. The dimension is 2.25 mm (Fig. 7-1 B).

However, we can’t use this method when the dimension of a part is larger than the range of an indicator. Then we can use the method of relative measurements. First of all, we must know the approximate dimension of a part in order to adjust the instrument. We measure a part with the help of a simple instrument, such as a ruler or a vernier caliper. For example, our dimension is about 124 mm.

The second, we take gage blocks and compose the block 124 mm, using blocks 100, 20 and 4 mm. It is necessary to compose the block attentively, looking at working polished sides of the blocks: they must contact each other. Before adjusting the blocks it is necessary to clean their working sides with alcohol.

The third, we place the composed block on the table and lower bracket, repeating adjusting as we have said before. However, it is better for us to adjust the indicator short hand not against 0, but against 1 or 2 mm in order that the reading should be possible if the dimension of a part proves to be less than we thought. For example, we have adjusted the indicator for 2.00 mm when we placed the block of 124 mm (Fig. 7-1C).

The fourth: we replace the part on the table, lower the spindle and look at the face. For example, indicator shows 3.21 mm (Fig. 7-1D).

The fifth, we define the relative transference: 3.21-2.00=1.21 mm.

The sixth , we obtain the general dimension : 124+1.21=125.21 mm.

This method is called «Method of relative measurements» as the measurement is made relatively to the block surface.

The advantages of this method are:

1. Higher accuracy, especially with small relative transference as errors of measuring instrument will not be accumulated.
2. Short time required for the inspection process.

The disadvantages are:

1. Necessity of adjustment and measurements of only a definite adjusted size.
2. A long time of adjustment.

7.2. Tolerances and Fits

The terms used in dimensioning limits of size are so interrelated that they should be understood clearly for correct interpretation of dimensions. The following definitions correspond to the system ISO and are adapted from those by ANSI (American National Standards Institute).

Actual size is a size measured with permissible accuracy.

To understand it better we place the details (parts) so that the lower part of the surface must be set against a certain level without looking at the actual size

(Fig. 7-2). The size of the **shaft** is marked **d**, the size of the **hole** is marked **D** (Fig. 7-3). As a shaft we consider all elements of the part which are external, and a hole is an element that is internal.

In accordance with ISO standards the shaft size is marked, for example, 30g6. Figure 30 shows the basic size in millimeters. This size is calculated by engineers in accordance with the requirements of solidness, rigidness, etc. That is why they account only whole millimeters of size. But the movement of the mating parts (the hole and the shaft) is obtained the difference of the actual sizes which makes up only hundredth parts of a millimeter. The **basic** size is the size from which limits of size are derived by the application of fundamental deviation and tolerances. The basic size is the figure and in the picture we draw a line from which limits of size are derived by the application of fundamental deviation and tolerances too. This line is called a **zero line** (Fig. 7-2).

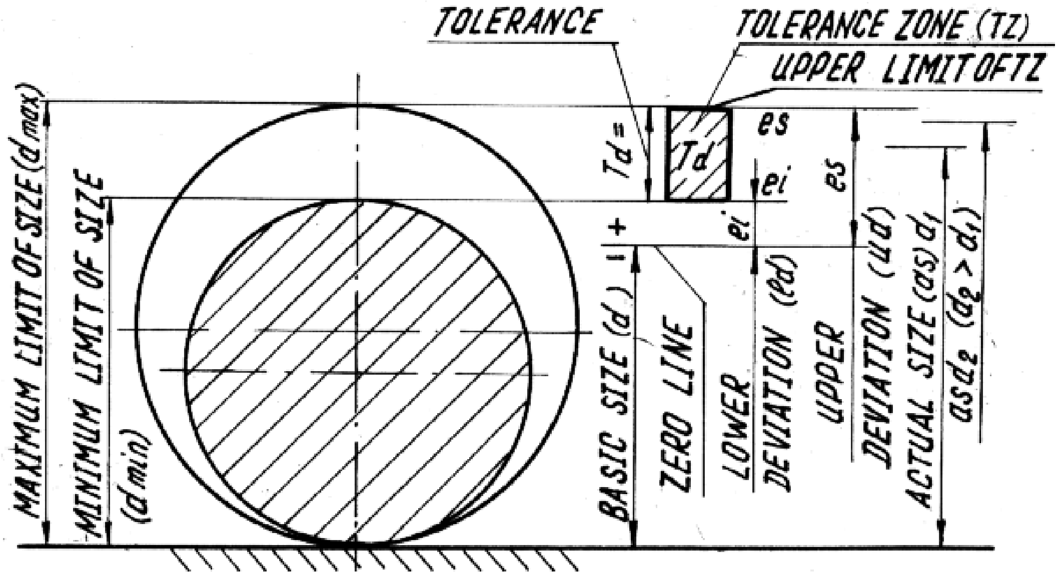


Fig. 7-2. A scheme of the shaft tolerance zone.

A letter determines a **fundamental deviation**: the deviation between basic size (zero line) and the nearest limit of a tolerance zone (Fig.7-4). We read the fundamental deviation value in the table of fundamental deviations (Table 7-1). The

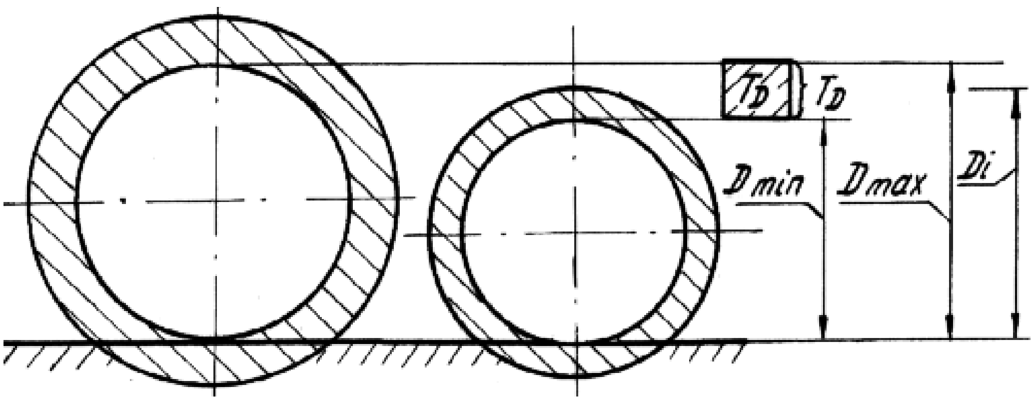


Fig. 7-3. A scheme of the hole tolerance zone.

positive sign shows that the fundamental deviation is laid up from the zero line. The **negative sign** shows that the fundamental deviation is laid low from the zero line.

A figure after the letter shows a **grade of tolerance** that determines the value of tolerance (Table 7-2). **Tolerance** is the total permissible variation of a size. It is the difference between the maximum and minimum limits of size. Numbers 6 and 7 grades of tolerance are accuracy and are obtained by grinding, accuracy turning processing, etc. Numbers 8,9,10 are middle accuracy; 11, 12 are rugged enough; 14 is very rugged.

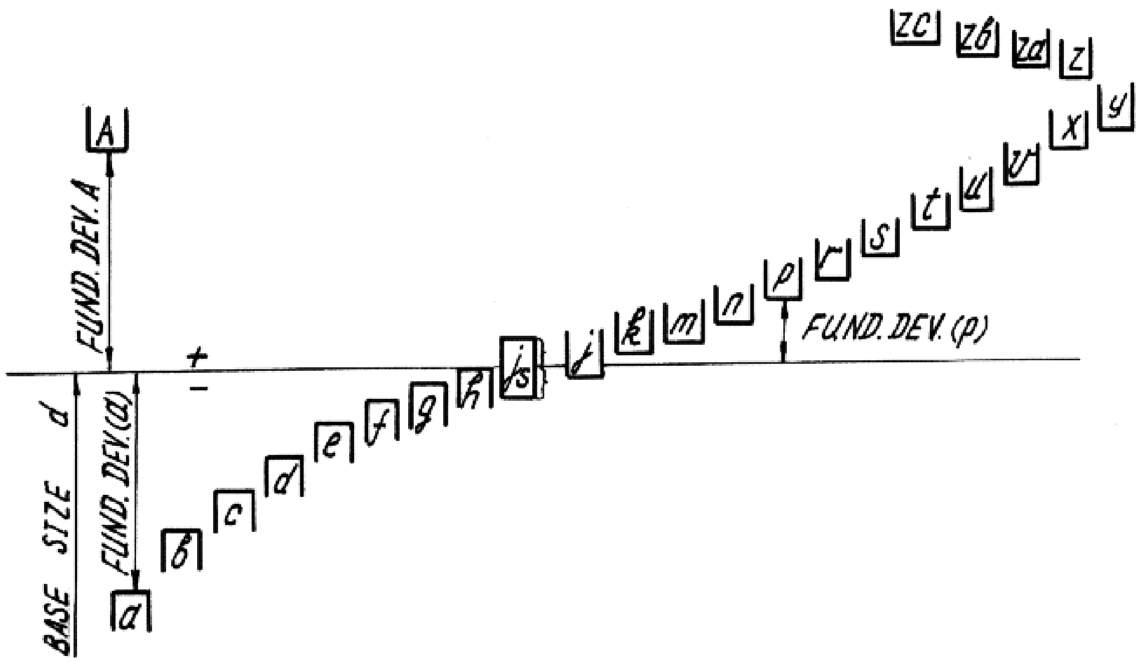


Fig. 7-4. A scheme of fundamental deviations of shafts.

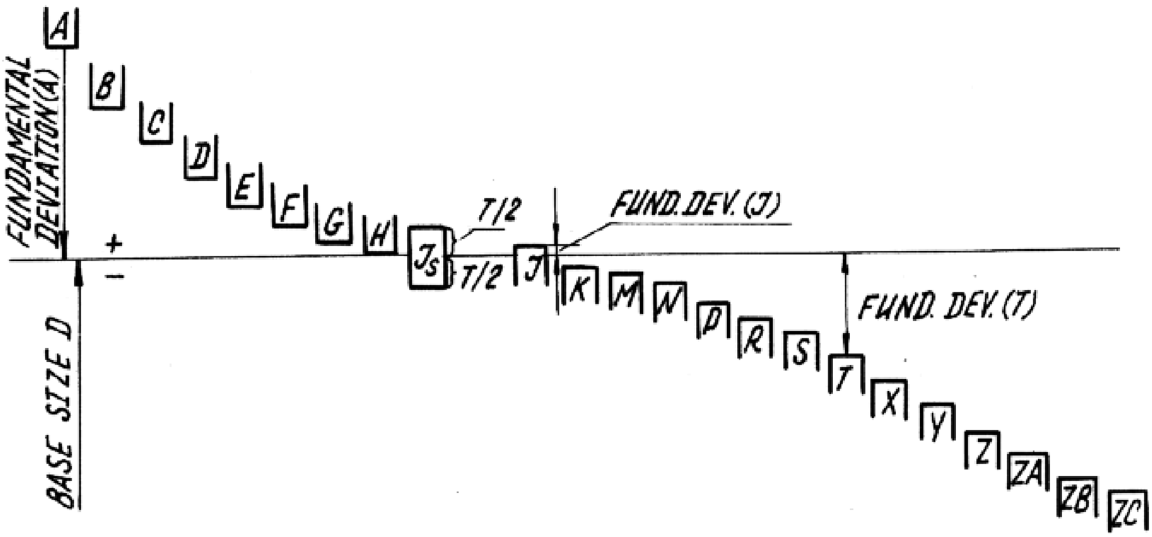


Fig. 7-5. A scheme of fundamental deviations of holes.

Table 7-1. Fundamental deviations, μm (1μm = 0.001 mm).

A basic size, mm	a	b	c	d	e	f	g	j		k		m	n	p	r	s	t	u	v	x	y	z
	For all numbers of grades of tolerance							5,6	7	4 -7	<3, >7	For all numbers of grades of tolerance										
from 1- befor 3	-270	-140	-60	-20	-14	-6	-2	-2	-4	0	0	+2	+4	+6	+10	+14	---	+18	---	+20	---	+26
3 - 6	-270	-140	-70	-30	-20	-10	-4	-2	-4	+1	0	+4	+8	+12	+15	+19	---	+23	---	+28	---	+35
6 - 10	-280	-150	-80	-40	-25	-13	-5	-2	-5	+1	0	+6	+10	+15	+19	+23	---	+28	---	+34	---	+42
10 - 14	-290	-150	-95	-50	-32	-16	-6	-3	-6	+1	0	+7	+12	+18	+23	+28	---	+33	---	+40	---	+50
14 - 18	-290	-150	-95	-50	-32	-16	-6	-3	-6	+1	0	7	+12	+18	+23	+28	---	+33	+39	+45	---	+60
18 - 24	-300	-160	-110	-65	-40	-20	-7	-4	-8	+2	0	8	+15	+22	+28	+35	---	+41	+47	+54	+63	+73
24 - 30	-300	-160	-110	-65	-40	-20	-7	-4	-8	+2	0	8	+15	+22	+28	+35	+41	+48	+55	+64	+75	+88
30 - 40	-310	-170	-120	-80	-50	-25	-9	-5	-10	+2	0	9	+17	+26	+34	+43	+48	+60	+68	+80	+94	+112
40 - 50	-320	-180	-130	-80	-50	-25	-9	-5	-10	+2	0	9	+17	+26	+34	+43	+54	+70	+81	+97	+114	+136
50 - 65	-340	-190	-140	-100	-60	-30	-10	-7	-12	+2	0	11	+20	+32	+41	+53	+66	+87	+102	+122	+144	+172
65 - 80	-360	-200	-150	-100	-60	-30	-10	-7	-12	+2	0	11	+20	+32	+43	+59	+75	+102	+120	+146	+174	+210
80 - 100	-380	-220	-170	-120	-72	-36	-12	-9	-15	+3	0	13	+23	+37	+51	+71	+91	+124	+146	+178	+214	+258
100 -120	-410	-240	-180	-120	-72	-36	-12	-9	-15	+3	0	13	+23	+37	+54	+79	+104	+144	+172	+210	+254	+310
120 - 140	-460	-260	-200	-145	-85	-43	-14	-11	-18	+3	0	15	+27	+43	+63	+92	+122	+170	+202	+248	+300	+365
140 - 160	-520	-280	-210	-145	-85	-43	-14	-11	-18	+3	0	15	+27	+43	+65	+100	+134	+199	+228	+280	+340	+415
160 - 180	-580	-310	-230	-145	-85	-43	-14	-11	-18	+3	0	15	+27	+43	+68	+108	+146	+210	+252	+310	+380	+465

A basic size, mm	**a**	**b**	**c**	**d**	**e**	**f**	**g**	**j**		**k**		**m**	**n**	**p**	**r**	**s**	**t**	**u**	**v**	**x**	**y**	**z**
	For all numbers of grades of tolerance							5,6	7	4 -7	<3, >7	For all numbers of grades of tolerance										
180 - 200	-660	-340	-240	-170	-100	-50	-15	-13	-21	+4	0	17	+31	+50	+77	+122	+166	+236	+284	+350	+425	+520
200 - 225	-740	-380	-260	-170	-100	-50	-15	-13	-21	+4	0	17	+31	+50	+80	+130	+180	+258	+310	+385	+470	+575
225 - 250	-820	-420	-280	-170	-100	-50	-15	-13	-21	+4	0	17	+31	+50	+84	+140	+196	+284	+340	+425	+520	+640
250 - 280	-920	-480	-300	-190	-110	-56	-17	-16	-26	+4	0	20	+34	+56	+94	+158	+218	+315	+385	+475	+580	+710
280 - 315	-1050	-540	-330	-190	-110	-56	-17	-16	-26	+4	0	20	+34	+56	+98	+170	+240	+350	+425	+525	+650	+790
315 - 355	-1200	-600	-360	-210	-125	-62	-18	-18	-28	+4	0	21	+37	+62	+108	+190	+268	+390	+475	+590	+730	+900
355 - 400	-1350	-680	-400	-310	-125	-62	-18	-18	-28	+4	0	21	+37	+62	+114	+208	+294	+435	+530	+660	+820	+1000
400 - 450	-1500	-760	-440	-230	-135	-68	-20	-20	-32	+5	0	23	+40	+68	+126	+232	+330	+490	+595	+740	+920	+1100
450 - 500	-1650	-840	-480	-230	-135	-68	-20	-20	-32	+5	0	23	+40	+68	+132	+252	+360	+540	+660	+820	+1000	+1250

Table 7-2. Tolerances, µm.

A basic size, mm	A number of grades of tolerance																			
	01	**0**	**1**	**2**	**3**	**4**	**5**	**6**	**7**	**8**	**9**	**10**	**11**	**12**	**13**	**14**	**15**	**16**	**17**	**18**
1 - 3	0.3	0.5	0.8	1.2	2	3	4	6	10	14	25	40	60	100	140	250	400	600	1000	1400
3 - 6	0.4	0.6	1	1.5	2.5	4	5	8	12	18	30	48	75	120	180	300	480	750	1200	1800
6 - 10	0.4	0.6	1	1.5	2.5	4	6	9	15	22	36	58	90	150	220	360	580	900	1500	2200
10 - 18	0.5	0.8	1.2	2	3	5	8	11	18	27	43	70	110	180	270	430	700	1100	1800	2700
18 - 30	0.6	1	1.5	2.5	4	6	9	13	21	33	52	84	130	210	330	520	840	1300	2100	3300
30 - 50	0.6	1	1.5	2.5	4	7	11	16	25	39	62	100	160	250	390	620	1000	1600	2500	3900
50 - 80	0.8	1.2	2	3	5	8	13	19	30	46	74	120	190	300	460	740	1200	1900	3000	4600
80 - 120	1	1.5	2.5	4	6	10	15	22	35	54	87	140	220	350	540	870	1400	2200	3500	5400

A basic size, mm	A number of grades of tolerance																			
	01	**0**	**1**	**2**	**3**	**4**	**5**	**6**	**7**	**8**	**9**	**10**	**11**	**12**	**13**	**14**	**15**	**16**	**17**	**18**
120 - 180	1.2	2	3.5	5	8	12	18	25	40	63	100	160	250	400	630	1000	1600	2500	4000	6300
180 - 250	2	3	4.5	7	10	14	20	29	46	72	115	185	290	460	720	1150	1850	2900	4600	7200
250 - 315	2.5	4	6	8	12	16	23	32	52	81	130	210	320	520	810	1300	2100	3200	5200	8100
315 - 400	3	5	7	9	13	18	25	36	57	89	140	230	360	570	890	1400	2300	3600	5700	8900
400 - 500	4	6	8	10	15	20	27	40	63	97	155	250	400	630	970	1550	2500	4000	6300	9700
500 - 630	4.5	6	9	11	16	22	30	44	70	110	175	280	440	700	1100	1750	2800	4400	7000	11000
630 - 800	5	7	10	13	18	25	35	50	80	125	200	320	500	800	1250	2000	3200	5000	8000	12500
800 - 1000	5.5	8	11	15	21	29	40	56	90	140	230	360	560	900	1400	2300	3600	5600	9000	14000
1000 - 1250	6.5	9	13	18	24	34	46	66	105	165	260	420	660	1050	1650	2600	4200	6600	10500	16500
1250 - 1600	8	11	15	21	29	40	54	78	125	195	310	500	780	1250	1950	3100	5000	7800	12500	19500
1600 - 2000	9	13	18	25	35	48	65	92	150	230	370	600	920	1500	2300	3700	6000	9200	15000	23000
2000 - 2500	11	15	22	30	41	57	77	110	175	280	440	700	1100	1750	2800	4400	7000	11000	17500	28000
2500 - 3150	13	18	26	36	50	69	93	135	210	330	540	860	1350	2100	3300	5400	8600	13500	21000	33000
3150 - 4000	16	23	33	45	60	84	115	165	260	410	660	1050	1650	2600	4100	6600	10500	16500	26000	41000
4000 - 5000	20	28	40	55	74	100	140	200	320	500	800	1300	2000	3200	5000	8000	13000	20000	32000	50000
5000 - 6300	25	35	49	67	92	125	170	250	400	620	980	1550	2500	4000	6200	9800	15500	25000	40000	62000
6300 -8000	31	43	62	84	115	155	215	310	490	760	1200	1950	3100	4900	7600	12000	19500	31000	49000	76000
8000-10000	38	53	76	105	140	195	270	380	600	940	1500	2400	3800	6000	9400	15000	24000	38000	60000	94000
Value of unit of tolerance (a).	---	---	---	----	---	---	7	10	16	25	40	64	100	160	250	400	640	1000	1600	2500

The values of tolerance and fundamental deviation depend on the part size. All the range of sizes is divided into intervals of sizes within which the values of tolerance and fundamental deviation are equal. In tables the values of tolerance and fundamental deviation are written in micrometers.

In our example 30g6 the fundamental deviation **fd** in accordance with the letter **g** is -7μm = -0.007mm (Table 7-1). The size tolerance is T_d=13μm in accordance with the number **6** of the grade of tolerance (Table 7-2). The upper deviation **es** is equal to the fundamental deviation **fd** in this example: es=-7μm, Fig. 7-6. We define the lower deviation **ei**: ei=-7-13=-20μm. A lower limit of the tolerance zone corresponds to the lower deviation. An upper limit of the tolerance zone corresponds to the upper deviation. We can write: $30g6\left(^{-0.007}_{-0.020}\right)$ or $30^{-0.007}_{-0.02}$ mm. The maximum limit of the size: $d_{max}=d_{basic}+es=30+(-0.007)=29.993$mm.

The minimum limit of the size: $d_{min}=d_{basic}+ei=30+(-0.02)=29.98$ mm. The right size is settled between 29.98 and 29.993 mm, including size limits.

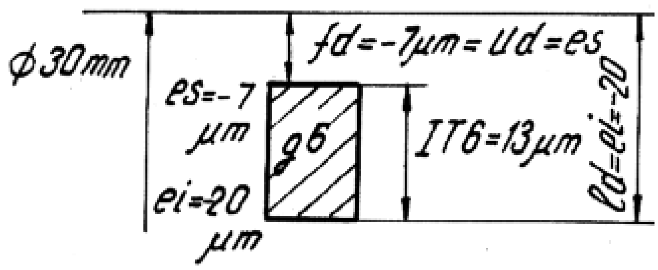


Fig. 7-6. A scheme of the tolerance zone 30g6.

If the fundamental deviation proves positive we will draw analogy between what we have written before.

For example 30s7: the fundamental deviation **fd** in accordance with the letter **s** is +35μm = +0.035mm (Table 7-1). The size tolerance is T_d=21μm in accordance with the number 7 of the grade of tolerance (Table 7-2).The lower deviation **ei** is equal to the fundamental deviation **fd** in this example ei=+35μm. We define the upper deviation **es**: es=+35+21=+56μm. We can write: $30s7\left(^{+0.056}_{+0.035}\right)$ or $30^{+0.056}_{+0.035}$ mm. The maximum limit of size: $d_{max}=d_{basic}+es=$ 30+(+0.056)=30.056 mm. The minimum limit of size: $d_{min}=d_{basic}+ei=30+(+0.035)=30.035$ mm. The right size is settled between 30.035 and 30.056 mm including size limits.

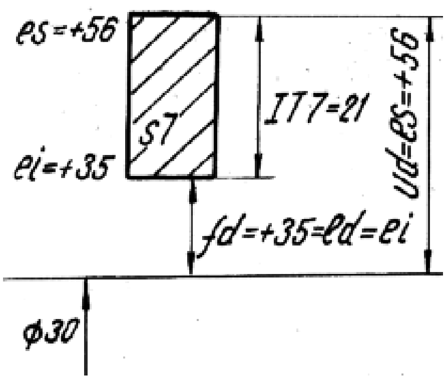


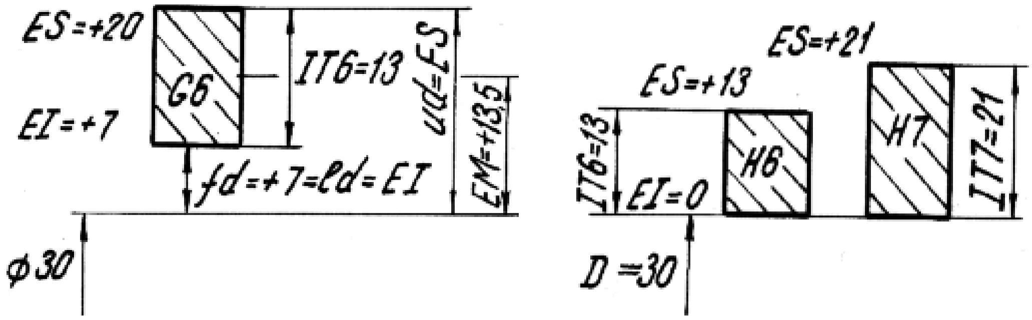
Fig. 7-7. A scheme of the tolerance zone 30s7.

Holes standard is analogous to shafts standard but the arrangement of fundamental deviations is like a mirror and fundamental deviations are marked with a capital letter (Fig.7-5).

For example, for 30G6 the fundamental deviation is +7μm likely -7μm for the shaft 30g6. Lower deviation coincides with the fundamental deviation: EI=+7μm.

The size tolerance is T_D=13 μm. The upper deviation is ES=+7+13=+20 μm. We can write: 30G6$\left({}^{+0.020}_{+0.007}\right)$ or $30^{+0.020}_{+0.007}$ mm.

There are standards where deviations are already defined. First of all we find the standard for a hole or a shaft, then we find the necessary grade of a tolerance number. After that we find the necessary letter of the fundamental deviation (for example, g6). On an intersection letter column with a size line we obtain both deviations (for example, 30g6: ${}^{-7}_{-20}$ μm).



A B

Fig. 7-8. A scheme of tolerances zones 30G6 (A) and basic holes H6 and H7 (B).

Sometimes we can use the mean deviation: em=(es+ei)/2 or EM=(ES+EI)/2. For example, for 30G6$\left({}^{+0.020}_{+0.007}\right)$ EM=(+0.02+0.007)/2=+0.0135 mm.

In order to reduce quantity of tolerance zone we have to use a tolerance zone for the preferable application. These zones are marked in standards.

It is better if the lower deviation of a hole will be equal to 0 to reduce the number of cutting tools such as drills, reamers and others. If the lower deviation is not equal to 0 we must have cutting tools of several sizes, for example, 10.1mm, 10.2mm, 10.3mm, etc. When we use the hole tolerance zone with a zero lower deviation we can use only the tools with whole millimeters, for example, 10mm, 11mm, 12mm, etc. These tolerance zones are H6, H7, H8, etc. This hole is called a **basic hole**.

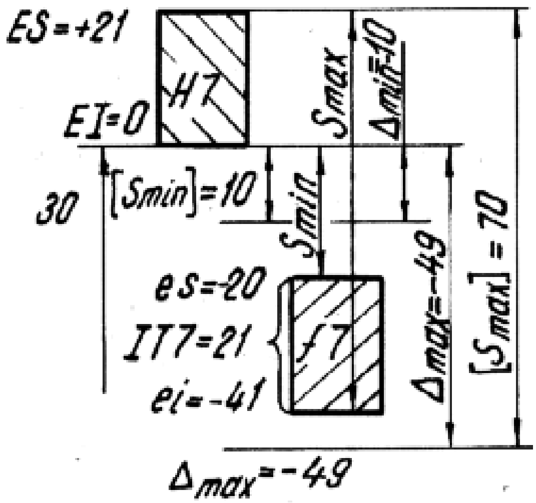


Fig. 7-9. A scheme of the clearance fit.

A hole together with a shaft makes a fit. Fits are divided into: 1) clearance fits; 2) interference fits; 3) transition fits.

When we deal with clearance fits a shaft is always smaller then a hole and we obtain only clearances between minimum clearance and maximum clearance (Fig. 7-9). We can define the tolerance of the fit: $T_F=T_S= S_{max}-S_{min}= T_D+T_d$.

When we deal with interference fits a shaft is always larger then a hole and we obtain only interferences between the minimum interference and maximum interferences (Fig. 7-10). We can define a tolerance of the fit: $T_F=T_N= =N_{max}-N_{min}= T_D+T_d$.

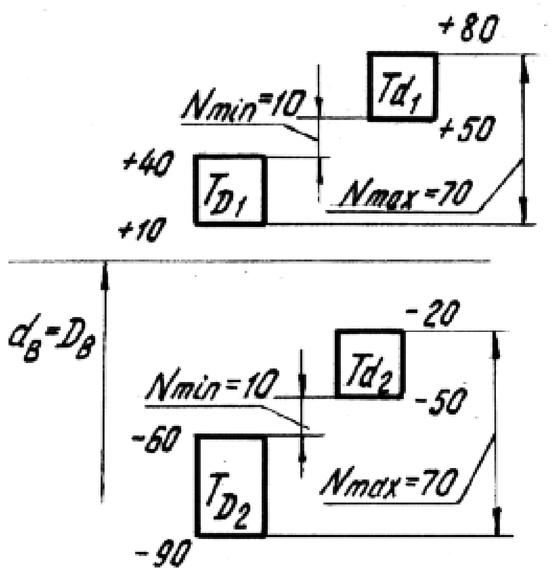


Fig. 7-10. A scheme of the interference fit.

When we deal with transition fits we can obtain clearances or interferences dependent on an actual size of a hole and a shaft (Fig. 7-11). These fits have S_{max} and N_{max}. We can define a tolerance of the fit: $T_F= N_{max}+S_{max}= =T_D+T_d$.

We can nominate a fit using the hole-basis system of fits or shaft-basis system of fits.

When we nominate a fit using the hole-basis system of fits the hole is made as the basic hole (H) and we select (pick out) a shaft tolerance zone to this basic hole in order to obtain the necessary fit.

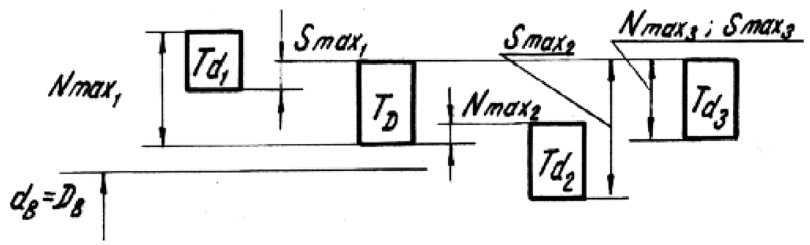


Fig. 7-11. A scheme of transitions fits: 1) T_D/T_{d1}; 2) T_D/T_{d2}; 3) T_D/T_{d3}.

When we nominate a fit using the shaft-basis system of fits, the shaft is made as the basic shaft (h) and we select a hole tolerance zone to this basic shaft in order to obtain the necessary fit.

The hole-basis system of fits is more preferable than the shaft-basis system of fits.

For example, it is necessary to pick out a fit if the basic size of the fit is 30 mm, the maximum clearance must be less than 70μm ($S_{max}<70\mu m$), the minimum clearance must be more than 10μm ($S_{min}>10\mu m$).

$d_b=30$ mm, $[S_{max}]=70\mu m$, $[S_{min}]=10\mu m$.

1. We define an approximate tolerance of the fit: $T_F = T_S= [S_{max}] - [S_{min}] = =70-10=60\mu m$.
2. We define an approximate tolerance of the hole and shaft considering $T_D= T_d$: if $T_D+ T_d=T_F$, that is $T_D=T_F/2=60/2=30\mu m$. $[T_D]=30\mu m$.
3. We define the grade of tolerance: a tolerance must be less than 30μm. For d=30 mm that is number 7 (IT7=21μm). For number 8 IT8=33$\mu m>[T_D]=30\mu m$.

4. We draw the tolerance zone for the hole 30H7. The letter H we have nominated as we use the basic hole (a hole-basis system). The lower deviation is 0, the upper deviation is +21 μm.
5. We lay $[S_{max}]$=70 μm and $[S_{min}]$=10 μm as it is shown in Fig.7-12.

6. The fundamental deviation must be less than -10 μm and the lower deviation must be larger than -49 μm. A shaft tolerance zone must settled between -10 μm and -49 μm, Fig. 7-12.

7. Looking at the fundamental deviation table we define the letter of the shaft tolerance zone. This is the letter f (the fundamental deviation fd=-20 μm).

8. We define shaft tolerance: IT7=21 μm.

9. We define lower deviation:

ei=-20-21=-41 μm.

10. We define clearances:

S_{min}=20 μm > $[S_{min}]$=10 μm; S_{max}=ES-ei= =+21-(-41)=62 μm<$[S_{max}]$=70 μm.

All conditions are executed.

11. We have defined the fit: H7/f7 or 30H7/f7.

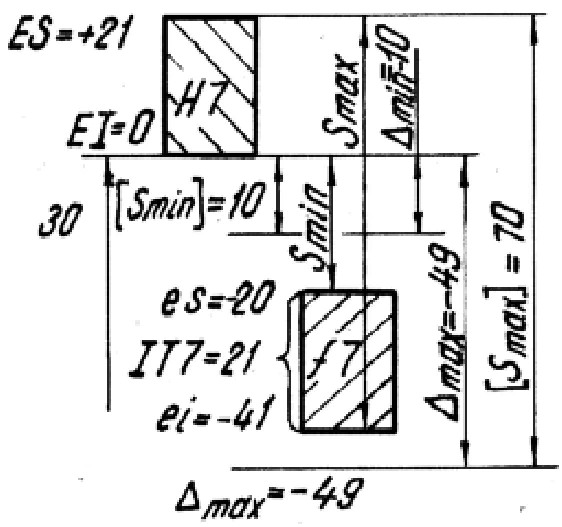


Fig. 7-12. A scheme of the fit 30H7/f7.

Chapter 8. Special Forms of Fits

8.1. Fits for Rolling Contact Bearing

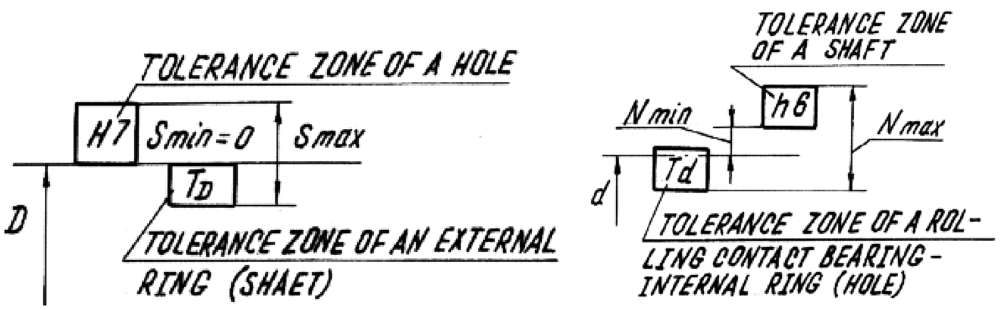
Bearings differ according to the class of accuracy: 0; 6; 5; 4; 2. The less the number of the class the higher is the accuracy. Although the size of inner race is a «hole», the tolerance zone is nominated inside of the basic size. This permits us to use tolerance zones of shaft such as k6, m6, n6 to make interference fits.

When a load is directed to the unchangeable place of the inner surface of race we have local loading of race and this race is mounted on the shaft or into the hole with clearance. This is done in order that loading place may change and wear of race inner surface is more uniform and work life increases. Very often the outside race is loaded this way and a fit is used usually H7/b0 where b0 is the tolerance zone of the outside race (b means «bearing », 0 is the class of accuracy of bearing). In accordance with the class of accuracy of the bearing we define the value of deviations in standard of bearings (see Table 8-1).

Table 8-1. Deviations of bearing races.

A basic diameter d (D), mm	Deviations of **inner** race, μm				Deviations of **outside** race, μm			
	The class of accuracy is **0**		The class of accuracy is **6**		The class of accuracy is **0**		The class of accuracy is **6**	
	es	ei	es	ei	ES	EI	ES	EI
0.6 - 2.5	+1	-9	+1	-8	---	----	---	---
2.5 - 10	+2	-10	+1	-8	+1	-9	+1	-8
10 - 18	+3	-11	+1	-8	+2	-10	+1	-8
18 - 30	+3	-13	+1	-9	+2	-11	+1	-9
30 - 50	+3	-15	+1	-11	+3	-14	+2	-11
50 - 80	+4	-19	+2	-14	+4	-17	+2	-13
80 - 120	+5	-25	+3	-18	+5	-20	+2	-15
120 - 180	+6	-31	+3	-21	+6	-24	+3	-18

The other race has no local loading and the fit with interference is used. Usually the B0/n6 fit is used where B0 is the tolerance zone of the inner race (B means «bearing », 0 is the class of accuracy of the bearing). In accordance with the class of accuracy of the bearing we define the value of deviations in standard of bearings (see Table 8-1).



A B

Fig. 8-1. A scheme of the bearing fit.

Bearings are industrial goods which are produced at specialized plants and never remade. That is why in these bearing fits we may show only the tolerance zone of a shaft or a hole with which bearing contact: 30n6 (we show only the tolerance zone of the shaft on which bearing is mounted); 60H7 (we show only the tolerance zone of the hole into which bearing is mounted).

8.2. Selective Assembly

Sometimes the requirement of the accuracy of a fit is so high that it can not be achieved using the ordinary methods. In this case one can use the method of selective assembly. The tolerance zones of a hole and shaft are divided into groups as shown in Fig. 8-2. After that we assemble the hole and the shaft using only the equal numbers of groups.

The tolerance of the group fit is:

$T_{F.g}=T_{S.g}=S_{max.g}-S_{min.g}=T_{D.g}+T_{d.g}$ (8.1)

$T_{S.g}<T_S=S_{max}-S_{min}$ (8.2)

Usually $T_D=T_d$ and $T_{D.g}=T_{d.g}$. Then $T_{S.g}=2T_{D.g}$.

$S_{min.g}> S_{min}$

$S_{min.g}=S_{min}+T_D-T_D/n$, (8.3)

where **n** is a quantity of groups.

For interference fits:

$N_{max.g}=N_{max}-T_D+T_D/n$. (8.4)

Usually n=2...3, but sometimes n=10 (in a bearing industry).

For example, it is necessary to make the fit with $S_{max}< [S_{max}]=20$ μm and $S_{min}> [S_{min}]=5$ μm for the basic size 50 mm. But we can produce parts with tolerance T=25 μm.

If we used an ordinary method we could not meet the requirements (Fig.8-2 A), $S_{max}> [S_{max}]$.

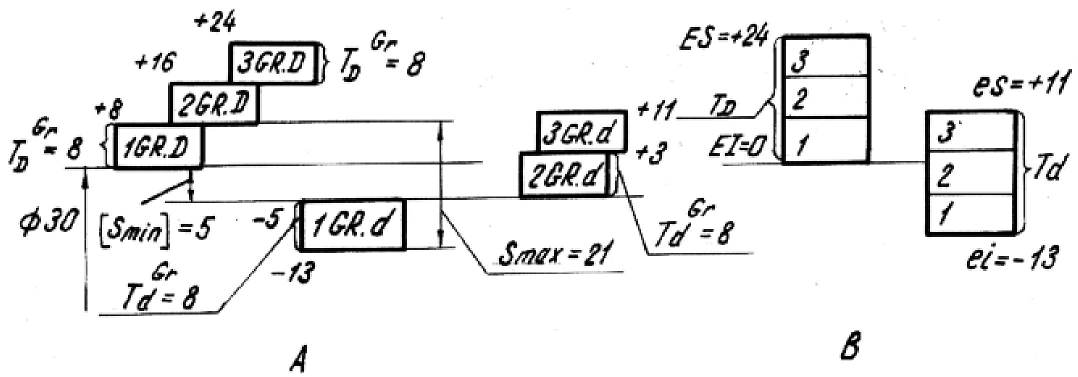


Fig. 8-2. A scheme of tolerance zones definition for the selective assembly.

We can use the method of selective assembly.

1. We draw the tolerance zone of a hole with the tolerance T=25 μm. It is preferable to use fits of hole-basis system of fits that is why we choose the tolerance zone of the hole as H (Fig.8-2 B).
2. We lay $[S_{min}]=5$ μm. This is the upper limit of the group tolerance zone of a shaft ($es_g=5$μm).
3. We define the tolerance of the group fit: $T_{S.g}=[S_{max}] - [S_{min}]=20-5=15$μm.
4. We define the tolerance of the group: $T_{D.g}= T_{d.g}= =T_{S.g}/2=15/2=7.5$μm.
5. We lay $ES_g=+7$μm into the tolerance zone of the hole.
6. We lay $[S_{max}]=20$ μm and draw the lower limit of the group tolerance zone of the shaft ($ei_g=13$μm).
7. We define the number of the groups: $n=T_D/T_{D.g}=25/7=3.5$. We nominate n =4.
8. We draw other groups tolerance zones of the hole.
9. We draw other groups tolerance zones of the shaft.

The task is made.

8.3. Gages for Testing Parts

The principles involved in testing a hole with gages are illustrated in Fig.8-3.

We have to make gage with the size equal D_{max}. If the size of the hole is smaller than D_{max} (a good size) this gage will not go through this hole. If the size of the hole is larger then D_{max} (a bad size) this gage will go through this hole. These gages do not go through a good hole that is why they are called **Not-Go** gages.

We have checked only the upper limit of the hole tolerance zone. Then we are to check a lower limit. We make the size of the gage equal to the lower possible size of the hole (D_{min}). If the size of the hole is larger than D_{min} (a good size) this gage will go through this hole. These gages go through a good hole that is why they are called **Go** gages. If the size of the hole is smaller than D_{min} (a bad size) this gage will not go through this hole.

Of course, we can not make gages absolutely exactly. That is why we nominate the tolerance of the gage **H** using the standard for gages . The **Go** gage will often go that is why the gage tolerance zone is increased by **z** and the possible wear size is reduced by **y** in order to increase work life of the gage.

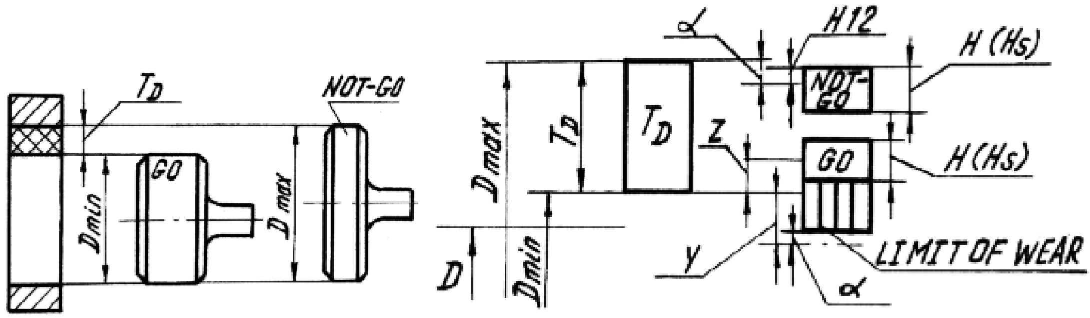


Fig. 8-3. A scheme of gages tolerances zones for testing a hole.

It must admit that the points of count for gage deviations are tolerance zone limits of the part.

The principles involved in testing shafts with gages are the same as for holes and are illustrated in Fig.8-4. If the size of the shaft is smaller then d_{max} (a good size) that gage is made with size equal d_{max} will go through this hole. If the size of a shaft is larger then d_{max} (a bad size) that gage will not go through this hole. These gages go through good shafts that is why they are called **Go** gages.

We have checked only the upper limit of the shaft tolerance zone. Then we are to check a lower limit. We make the size of the gage equal to the lower possible size of the shaft (d_{min}). If the size of the shaft is larger than d_{min} (a good size) this gage will not go through this hole. These gages do not go through good shafts that is why they are called **Not-Go**.

Value α is equal to **0** for the basic size is **smaller** than 180 mm.

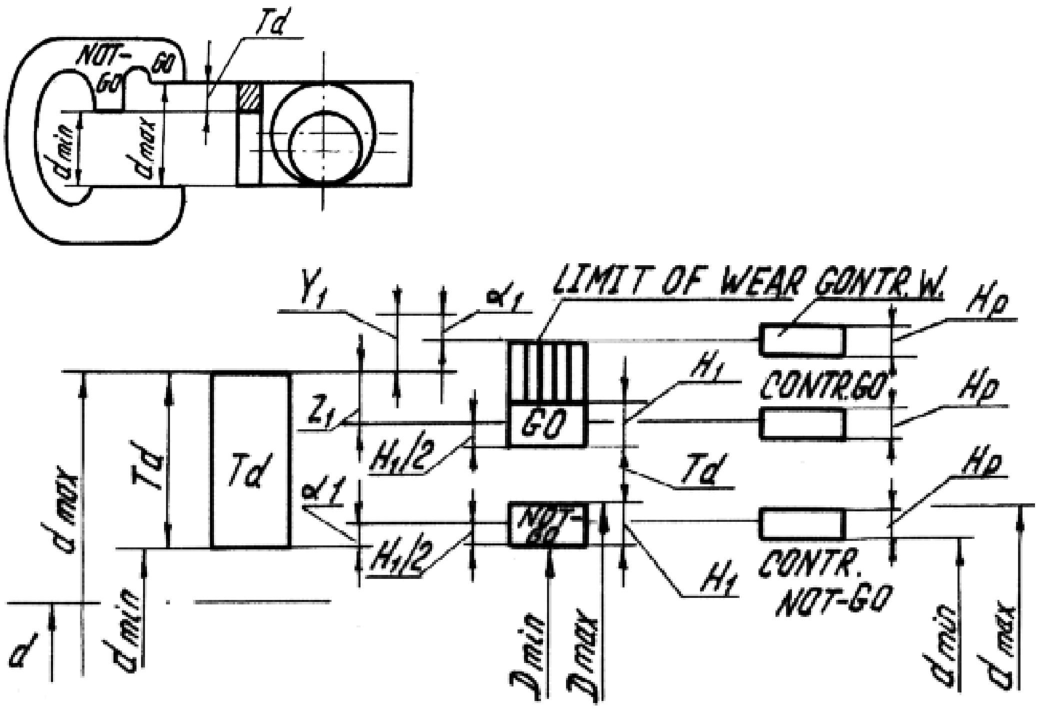


Fig. 8-4. A scheme of gages tolerance zones for testing a shaft.

Table 8-2. Tolerance of gages, μm.

Grade of tolerance	Symbol	Basic size, mm							
		1 - 3	3 - 6	6 - 10	10 - 18	18 - 30	30 - 50	50 - 80	80-120
6 gr.	z	1	1.5	1.5	2	2	2.5	2.5	3
	y	1	1	1	1.5	1.5	2	2	3
	z_1	1.5	2	2	2.5	3	3.5	4	5
	y_1	1.5	1.5	1.5	2	3	3	3	4
	H, H_s	1.2	1.5	1.5	2	2.5	2.5	3	4
	H_1	2	2.5	2.5	3	4	4	5	6
	H_p	0.8	1	1	1.2	1.5	1.5	2	2.5
7 gr.	z, z_1	1.5	2	2	2.5	3	3.5	4	5
	y, y_1	1.5	1.5	1.5	2	3	3	3	4
	H, H_1	2	2.5	2.5	3	4	4	5	6
	H_s	---	---	1.5	2	2.5	2.5	3	4
	H_p	0.8	1	1	1.2	1.5	1.5	2	2.5
8 gr.	z, z_1	2	3	3	4	5	6	7	8
	y, y_1	3	3	3	4	4	5	5	6
	H	2	2.5	2.5	3	4	4	5	6
	H_1	3	4	4	5	6	7	8	10
	H_s, H_p	1.2	1.5	1.5	2	2.5	2.5	3	4
9 gr.	z, z_1	5	6	7	8	9	11	13	15
	y, y_1	0	0	0	0	0	0	0	0
	H	2	2.5	2.5	3	4	4	5	6
	H_1	3	4	4	5	6	7	8	10
	H_s, H_p	1.2	1.5	1.5	2	2.5	2.5	3	4

The advantages of gages are:
1. Short time required for the inspection process.
2. Easy inspection process.
The disadvantages are:
1. Testing of only a definite size.
2. High cost of gage.

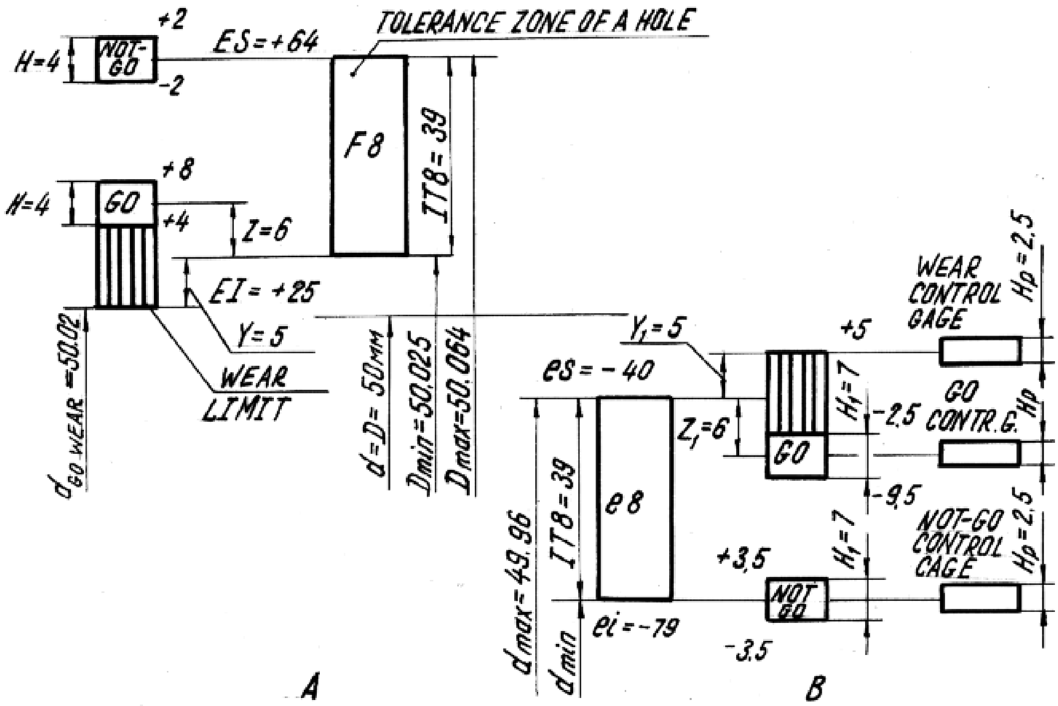


Fig. 8-5. A scheme of gages tolerance zones for testing a hole 50F8 and a shaft 50e8.

8.4. Key Fits

Key junctions are used for transmission of rotary load. For these purposes we can use: 1) straight (feather) key; 2) Woodruff key; 3) taper key. We will view the standard of a straight key (Fig. 8-6).

The key junction has 3 fits: on a width, on a length, on a height. The most important fit is the fit on the width. Fits are different: the fit on the width between the slot of a shaft and the key and between the slot of a sleeve and the key.

The key is always made with the tolerance zone h9. But the tolerance zones of the slot are different. Key fits are divided into:

1. A free junction. The fit between the shaft slot and the key (SS/k) is H9/h9. The fit between the hole slot and the key (HS/k) is D10/h9.
2. Normal junction: SS/k is N9/h9, HS/k is J_S9/h9.
3. Strong junction: SS/k is P9/h9, HS/k is P9/h9.

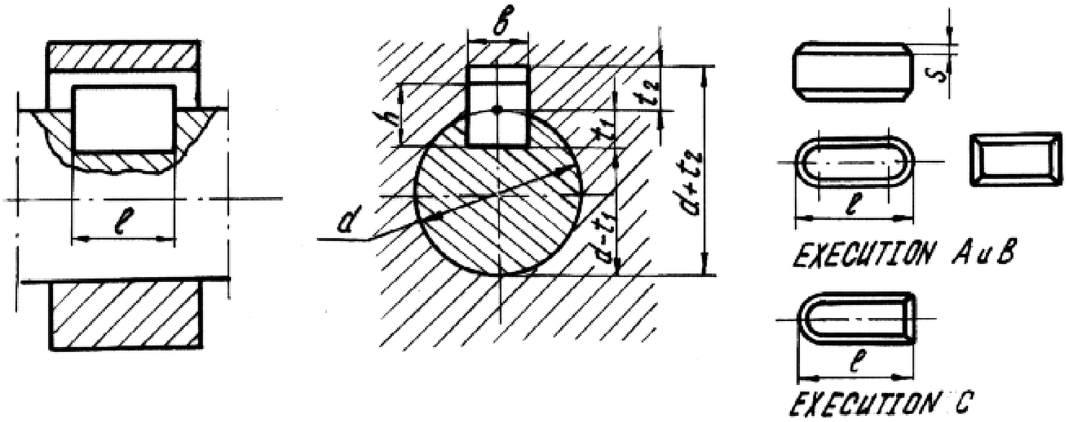


Fig. 8-6. A scheme of tolerance zones for the key junction.

The free junction is usually used. This type is always used for the move junction.

Strong junction is used for exchangeable loads.

The fit H15/h14 is always used on the length fit.

A height of the key is produced with the tolerance zone h9. The depth of slots is produced in accordance with a special standard for the key. There is always a clearance between the key and slots on the height.

8.5. Slit Fits

Slit junctions are used for the transmission of a large rotary load and for the transmission of a rotary load when there is an axle (shift) transference.

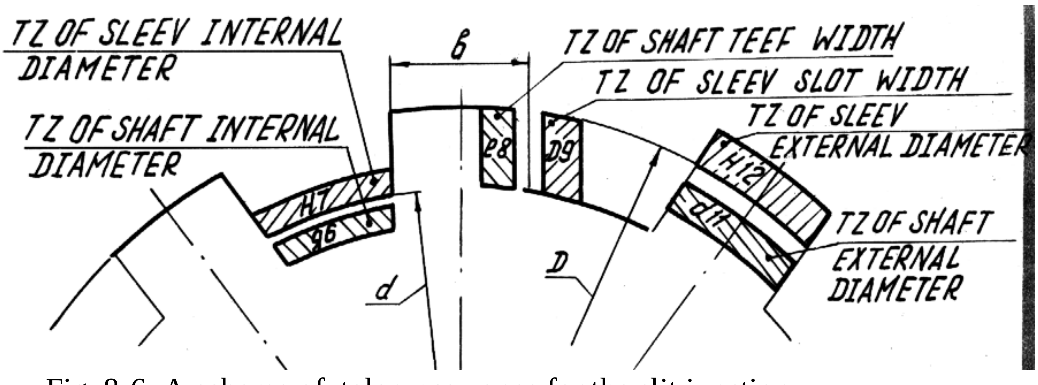


Fig. 8-6. A scheme of tolerance zones for the slit junction.

There are several types of slit junctions but usually the straight side slits are used. There are 2 types of centring: on an external diameter and on an internal diameter.

Internal diameter centring is usually used because of easier production: grinding of the internal diameter of the slit hole is more technological in comparison with that of the external diameter of the slit hole.

A designation of the slit junction is, for example: d-8 x 36H7/e8 x 40H12/a11 x 7D9/f8. For the hole: d-8 x 36H7 x 40H12 x 7D9. For the shaft: d-8 x 36e8 x 40a11 x 7f8.

8.6. Tolerances of Angles and Cones

There are 17 grades of tolerance for angles: from 1 to 17. Designation is AT1, AT2, ..., AT17, where: AT - Angle Tolerance; 1 - grade of tolerance. The numerical values of angle tolerances (difference between the maximum limit α_{max} and the minimum limit α_{min} of angles) one degree to another is changed with factor of increase 1.6. Degrees of accuracy 12 - 14 are coarse accuracy, degrees of accuracy 8 - 10 are middle accuracy, degrees of accuracy 6 - 7 are accurate. The tolerance is smaller for a greater size as easy to measure angle for greater length of angle. Tolerances are nominated depending on the smaller party of an angle.

The conecity is $C = (D - d)/L = 2tg\alpha/2$, where: D – the diameter of the large base of a cone; d – the diameter of the small base of a cone; α - corn of cone; $\alpha/2$ - corn of slope.

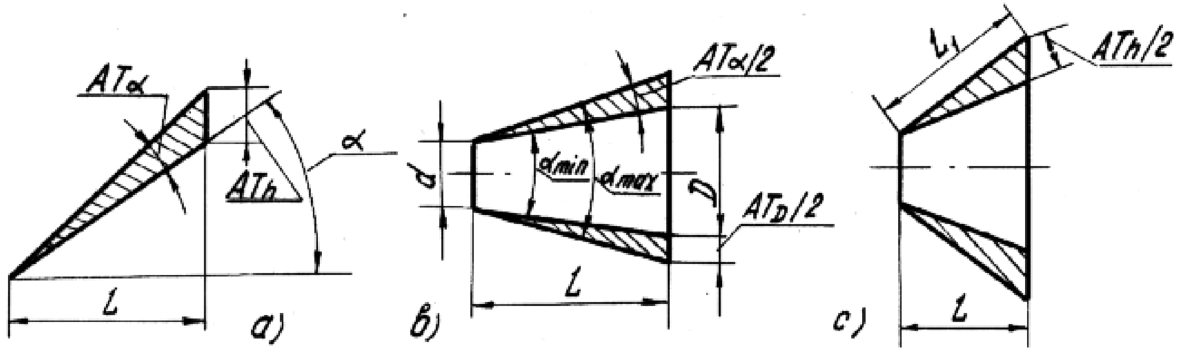


Fig.8-7. Disposition of tolerance zones of cones.

For each degree tolerances are established:

1. Angle tolerance AT_α, which is expressed in angle units as microradian (μrad) as angle degree and angle minute (^{o} , ′). Angle degrees are used chiefly. Round of values AT_α' are recommended to use on drawings.

2. Angle tolerance AT_h, which is expressed in millimeters (mm). This tolerance is expressed by a piece on a perpendicular to the party of a corner on distance L_1 from top of this corner.

3. Angle tolerance AT_D, which is expressed in millimeters by the tolerance on a difference of diameters in two perpendicular to an axis of a cone sections on specified distance between them (it is determined on a perpendicular to an axis of a cone) is used for conecity less than 1:3. Tolerances are nominated depending on the base length of cone L. When conecity greater tolerances are nominated depending on the forming length of cone L_1.

There are two ways to nominate the tolerance of a cone diameter:

1. The tolerance is nominated for a diameter. This tolerance is identical in any cross section of a cone and is limiting two cone limits, between which all points of a surface of the actual cone should settle down. The tolerance limits also deviation of a corner of a cone, if these deviations are not limited by smaller tolerance.
2. The tolerance is specified only for a specified section of a cone. The tolerance of the form is determined by the sum of tolerances of a roundness and straightness its forming. These tolerances are nominated on the diameter of the greater basis of a cone or diameter of a specified section of a cone.

Table 8-3. Tolerance of corners.

Size, mm	Symbol	Unit of measurement	Grade of tolerance							
			6	7	8	9	10	11	12	17
10 - 16	AT_α	μrad	500	800	1250	1600	2500	4000	6300	63000
	AT_α	...°	1′ 43″	2′ 45″	4′ 18″	5′ 30″	8′ 35″	13′ 44″	21′ 38″	3° 36′34″
	AT_h, AT_D	μm	5	8	12.5	16 - 25	25 - 40	40 - 63	63 - 100	630 - 1000
16 - 25	AT_α	μrad	315	500	800	1250	2000	3150	5000	50000
	AT_α	...°	1′ 5″	1′ 43″	2′ 45″	4′ 18″	6′ 52″	10′ 49″	17′ 10′	2° 51′ 53″
	AT_α'	...°	1′	1′ 40″	2′ 30″	4′	6′	10′	16′	2°
	AT_h, AT_D	μm	5 - 8	8 - 12.5	12.5 - 20	20 - 32	32 - 50	50 - 80	80 - 125	800 - 1250
25 - 40	AT_α	μrad	250	400	630	1000	1600	2500	4000	40000
	AT_α	...°	52″	1′ 22″	2′ 10″	3′ 26″	5′ 30″	8′ 35″	13′ 44″	2° 17′ 30″
	AT_α'	...°	50″	1′ 20″	2′	3′	5′	8′	12′	---
	AT_h, AT_D	μm	6.3 - 10	10 - 16	16 - 25	25 - 40	40 - 63	63 - 100	100-160	1000 - 1600

The deviations for corners are nominated chiefly symmetric, ally and can be both in plus or in minus. The conic fits can be with a clearance, interference and transition fits, depending on the axial arrangement. The fits are divided by the way of fixing of axial arrangement: 1) Fits with fixing by combining the constructive elements of cones (base planes); 2) Fits with fixing of a specified axial displacement of cones; 3) Fits with fixing of a specified axial distance between base planes of mating cones; 4) Fits with fixing of the specified force of pressing.

The clearance fits are applied in junctions, in which it is necessary to adjust a clearance between mating parts (for example, for bearings of spindle). The interference fits are applied to maintenance of tightness and self-centering, transfer of the twisting moment. Conic fits provide easier disassembly in comparison with cylindrical junctions, allow to adjust interference during work.

Methods and Tools for Control of Corners and Cones.There are comparative and trigonometric methods of control. The comparison of a checked corner with angle gage blocks, set squares and angle patterns are used in the first method. Conic gages are applied to the control of cones when conecity is used from 1:3 to 1:50 and tolerance degrees of corners and cones are used from 4 to 9. The control of a stain of contact is used for an estimation of a degree of cones adjoining. The application of sine bar, gage blocks and indicator allows to measure and to check corners and cones with a high accuracy. An universal bevel protractor are applied for measurements with accuracy 5′.

Chapter 9. Screw Threads

A screw thread is a helical or spiral ridge of uniform section on the surface of a cylinder or a cone, either external or internal. Threads on bolts and screws are external threads, Fig. 9-1. Threads on nuts are internal threads, Fig. 9-2. External threads and internal threads have the same basic pitch diameters. Threads on a cylindrical surface (such as bolts, machine screws, and nuts) are **straight or parallel** threads. Threads on a conical surface are **tapered** threads.

Threads may be **right-hand** (RH) or **left-hand** (LH). A right-hand thread advances away from the observer when turned clockwise. A left-hand thread advances away from the observer when turned counterclockwise. A grinder with two grinding wheels, one mounted on each end of the arbor, has a right-hand thread and nut on one end and a left-hand thread and nut on the other end. Taps and dies are available for use in cutting right-hand and left-hand threads. Unless a thread is otherwise designated, it is assumed to be right-hand.

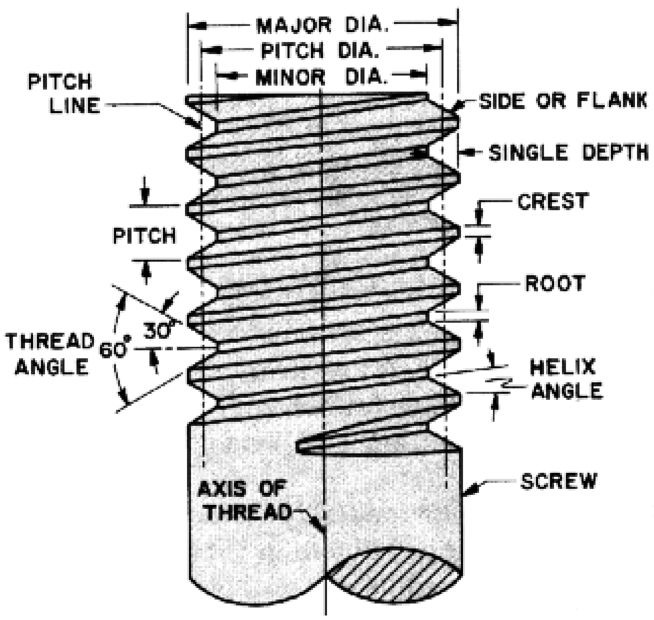


Fig. 9-1. Principal parts of an external screw thread.

Screw threads are widely used on fasteners such as nuts, bolts, and screws. They permit easy assembly and dismantling for replacement of parts.

Screw threads are also widely used to transmit motion, transmit power, increase mechanical advantage, control movement accurately and uniformly, and permit adjustments on machines.

The lead screw on a lathe transmits power. The screw on a vise provides for increasing mechanical advantage. The screw on a micrometer provides accurate and uniform control of movement, thus making accurate measurement possible. Screw threads allow for adjustments on tools, machines, instruments, and control devices.

External thread: the thread on the external surface of a cylinder or cone, Fig.9-1. **Internal thread:** the thread on the internal surface of a cylinder or cone, Fig. 9-2.

Major diameter: the largest diameter of a straight external or internal thread, Fig. 9-1. **Minor diameter:** the smallest diameter on a straight external or internal screw thread, Fig. 9-1. **Pitch diameter:** on a straight thread, the diameter of an imaginary cylinder which passes through the thread profile at points where the width of the groove and the width of the thread are equal, Fig. 9-1. The pitch diameter may be measured with a thread micrometer. The amount of clearance permitted between two mating threads is controlled by maintaining close tolerances on their pitch diameters.

The pitch diameter on a taper thread, at a given position on the thread axis, is the diameter of the pitch cone at that position.

Pitch: the distance from a point on one screw thread to a corresponding point on the adjacent thread, measured parallel to the thread axis, Fig. 9-1. The pitch of a thread is a measure of the size of the thread form. For metric threads, pitch is expressed in millimeters. For inch-based threads, pitch is equal to 1 divided by the number of threads per inch.

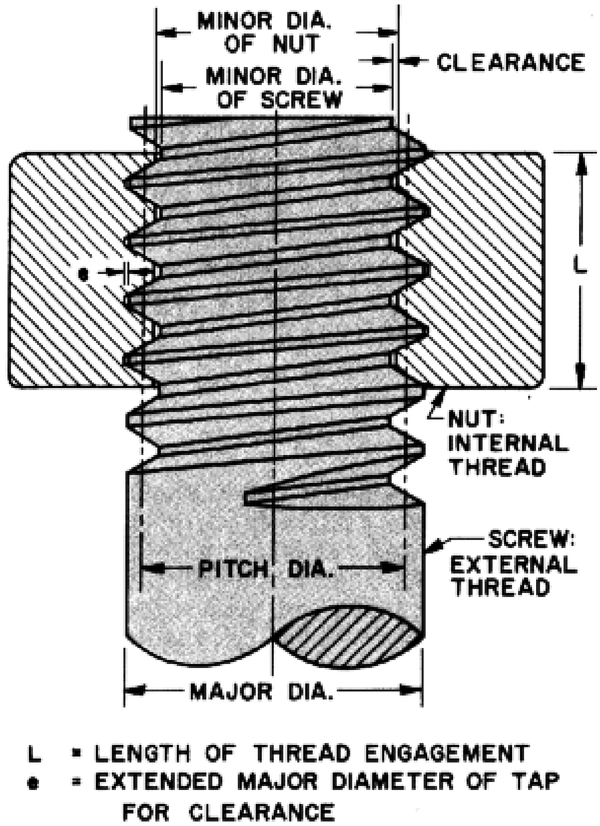
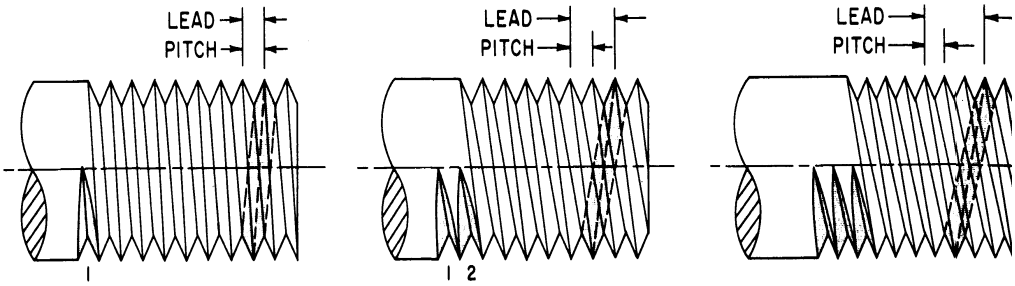


Fig. 9-2. Comparison between the minor diameters of a screw and a nut, showing clearance.

Lead: the distance a thread moves along its axis, with respect to a mating part, in one complete revolution. On a single thread, the lead and the pitch are the same. On a double thread, the lead is equal to twice the pitch. On a triple thread, the lead is equal to three times the pitch. See Fig. 9-3. (A single thread has one groove; a double thread, two grooves; a triple thread, three grooves; and so on, Fig. 9-3).

Multiple thread: the thread having the same form produced with two or more helical grooves, such as a double, triple, or quadruple thread, Fig. 9-3.

Angle of thread: the included angle between the sides of flanks of the thread, measured in an axial plane, Fig. 9-1.



Single thread Double thread Triple thread

Fig. 9-3. Relationship of pitch and lead on multiple threads.

Lead angle (sometimes called helix angle)**:** the angle made by the helix of a thread at the pitch diameter, measured in a plane perpendicular to the axis of the thread.

Axis of a screw thread: the axis of the pitch cylinder or cone on which the screw thread appears, Fig. 9-1.

Crest: the top surface which joins the two sides of a thread. The crest of an external thread is at its major diameter. The crest of an internal thread is at its minor diameter, Figs. 9-1 and 9-2.

Root: the bottom surface which joins the sides of two adjacent threads. The root of an external thread is at its minor diameter. The root of an internal thread is at its major diameter, Figs. 9-1 and 9-2.

Flank: the surface which connects the crest with the root on either side of the thread, Fig. 9-1.

Clearance: the distance between the crest of a thread and the root of the mating thread, measured perpendicular to the thread axis, Fig. 9-2.

Depth of engagement: the depth to which one thread is engaged with a mating thread, measured perpendicular to the thread axis.

Length of engagement: the contact distance between an external and internal thread, measured parallel to the axis along the pitch cylinder or cone, Fig. 9-2.

Height of thread (sometimes called depth of thread)**:** the distance between the major and minor cylinders or cones of the thread, measured perpendicular to the axis of the thread.

Form: the profile for a length of one pitch in an axial plane, Fig. 9-4.

Maximum material limits: the maximum limit of size for an external dimension, or the minimum limit of size for an internal dimension.

Minimum material limits: the minimum limit of size for an external dimension, or the maximum limit of size for an internal dimension.

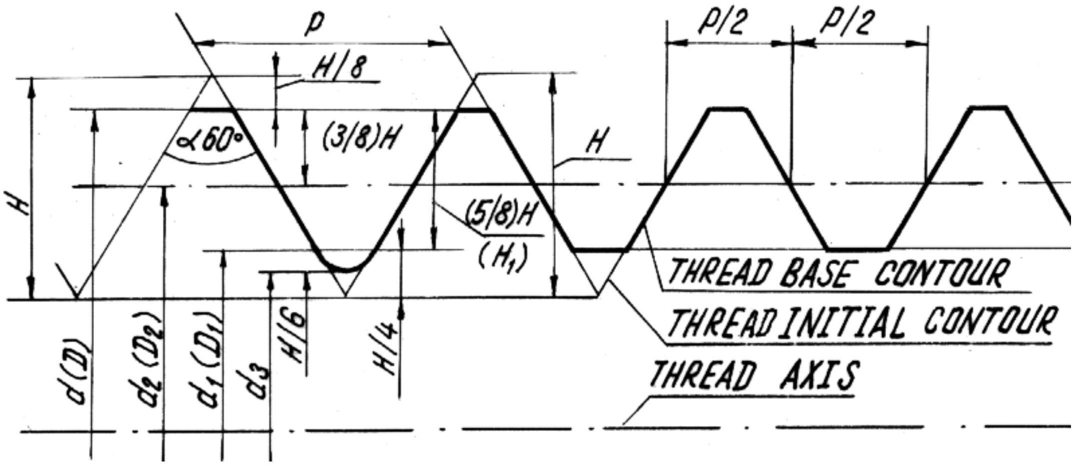


Fig. 9-4. Cross section of thread form.

Table 9-1. Tolerance grades for ISO Metric threads.

External Thread		Internal Thread	
Major Diameter	Pitch Diameter	Major Diameter	Pitch Diameter
	3		
4	4	4	4
	5	5	5
6	6	6	6
	7	7	7
8	8	8	8
	9		

Basically, there are three classes of fit: **fine, medium,** and **coarse.** The classes of fit are more accurately identified by specification of the **tolerance grade** and **tolerance position** of the mating external and internal threads. Tolerance grades are specified by a number, and may be applied to both the major diameter and pitch diameter. Table 9-1 lists the range of tolerance grades for external and internal threads. Grade 6 is recommended for medium fits on general purpose threads, and is closest to Unified class 2A and 2B fits.

Tolerance position is specified with a lowercase letter for external threads, and a capital letter for internal threads as follows:

External threads: e - large allowance; g - small allowance; h - no allowance.

Internal threads: G - small allowance; H - no allowance.

The combination of tolerance grade and tolerance position constitutes the **tolerance class** of the thread. Fine - 5H/4h; medium -6H/6g; coarse - 7H/8g.

Basic designations for all ISO Metric threads begin with the capital letter "M". Next, the nominal size (basic major diameter) in millimeters is given. This is followed by the pitch in millimeters, separated by an "X". ISO practice calls for the pitch to be omitted when designating coarse series threads. Therefore, an ISO Metric 10 mm coarse series thread with a pitch of 1.5 mm (large pitch) would simply be designated M10, whereas the same diameter in the fine series would be designated M10 X 1.25. See Table 9-2.

Table 9-2. Basic diameters and pitches of Metric thread for rows 1 and 2.

Basic major diameter, mm		Pitch P, mm		Basic diameter, mm		Pitch, mm	
Row 1	Row 2	Large	Small	Row 1	Row 2	Large	Small
3		0.5			33	3.5	2; 1.5
4		0.7	0.5	36		4	3; 2; 1.5
5		0.8	0.5		39	4	3; 2; 1.5
6		1		42		4.5	3; 2; 1.5
8		1.25	1		45	4.5	3; 2; 1.5
10		1.5	1.25; 1	48		5	3; 2
12		1.75	1.5; 1.25		52	5	3; 2
	14	2	1.25	56		5.5	4; 3
16		2	1.5		60	---	4; 3
	18	2.5	2; 1.5	64		6	4; 3
20		2.5	2; 1.5		68	6	4; 3
	22	2.5	2; 1.5	72		---	6; 4; 3
24		3	2; 1.5		76	---	6; 4; 3
	27	3	2; 1.5	80		---	6; 4; 3
30		3.5	2; 1.5		85	---	6; 4; 3
				90		---	6; 4; 3

Table 9-3. Value of basic pitch and minor diameters of Metric thread.
(ISO 68 and ISO R1501-1970.)

Pitch P, mm	Basic diameters of a fread		Pitch P, mm	Basic diameters of a fread	
	Pitch diameter d_2 and D_2	Minor diameter d_1 and D_1		Pitch diameter d_2 and D_2	Minor diameter d_1 and D_1
0.075	d - 1 + 0.951	d - 1 + 0.919	0.7	d - 1 + 0.546	d - 1 + 0.242
0.08	d - 1 + 0.948	d - 1 + 0.913	0.75	d - 1 + 0.513	d - 1 + 0.188
0.09	d - 1 + 0.942	d - 1 + 0.903	0.8	d - 1 + 0.480	d - 1 + 0.134
0.1	d - 1 + 0.935	d - 1 + 0.892	1	d - 1 + 0.350	d - 2 + 0.918
0.125	d - 1 + 0.919	d - 1 + 0.865	1.25	d - 1 + 0.188	d - 2 + 0.647
0.15	d - 1 + 0.903	d - 1 + 0.838	1.5	d - 1 + 0.026	d - 2 + 0.376
0.175	d - 1 + 0.886	d - 1 + 0.811	1.75	d - 2 + 0.863	d - 2 + 0.106
0.2	d - 1 + 0.870	d - 1 + 0.783	2	d - 2 + 0.701	d - 3 + 0.835
0.225	d - 1 + 0.854	d - 1 + 0.756	2.5	d - 2 + 0.376	d - 3 + 0.294
0.25	d - 1 + 0.838	d - 1 + 0.730	3	d - 2 + 0.051	d - 4 + 0.752
0.3	d - 1 + 0.805	d - 1 + 0.675	3.5	d - 3 + 0.727	d - 4 + 0.211
0.35	d - 1 + 0.773	d - 1 + 0.621	4	d - 3 + 0.402	d - 5 + 0.670
0.4	d - 1 + 0.740	d - 1 + 0.567	4.5	d - 3 + 0.077	d - 5 + 0.129
0.45	d - 1 + 0.708	d - 1 + 0.513	5	d - 4 + 0.752	d - 6 + 0.587
0.5	d - 1 + 0.675	d - 1 + 0.459	5.5	d - 4 + 0.428	d - 6 + 0.046
0.6	d - 1 + 0.610	d - 1 + 0.350	6	d - 4 + 0.103	d - 7 + 0.505

Table 9-4. Fundamental deviations of Metric thread.
(ISO 68 and ISO R1501-1970.)

Pitch P, mm	Fundamental deviation of a thread, μm								
	External thread (**es** for d, d_2, d_1.)					Internal thread (**EI** for D_1, D_2, D.)			
	d	e	f	g	h	E	F	G	H
0.5	---	-50	-36	-20	0	+50	+36	+20	0
0.7	---	-56	-38	-22	0	+56	+38	+22	0
0.8	---	-60	-38	-24	0	+60	+38	+24	0
1	-90	-60	-40	-26	0	+60	+40	+26	0
1.25	-95	-63	-42	-28	0	+63	+42	+28	0
1.5	-95	-67	-45	-32	0	+67	+45	+32	0
1.75	-100	-71	-48	-34	0	+71	+48	+34	0
2	-100	-71	-52	-38	0	+71	+52	+38	0
2.5	-106	-80	-58	-42	0	+80	---	+42	0
3	-112	-85	-63	-48	0	+85	---	+48	0
3.5	-118	-90	---	-53	0	+90	---	+53	0
4	-125	-95	---	-60	0	+95	---	+60	0
4.5	-132	-100	---	-63	0	+100	---	+63	0
5	-132	-106	---	-71	0	+106	---	+71	0
5.5	-140	-112	---	-75	0	+112	---	+75	0
6	-150	-118	---	-80	0	+118	---	+80	0

Complete designations for ISO Metric threads include identification of the tolerance class. The tolerance class follows the basic designation, separated by a dash. The tolerance grade and position for the pitch diameter are given first, followed by the tolerance grade and position for the major diameter. If the pitch and major diameter tolerance are the same, then the symbols need only be given once.

Examples: M10 X 1.25-6g8g, where 6g- tolerance class of a pitch diameter tolerance symbol; 8g- tolerance class of a major diameter tolerance symbol. 6 or 8- tolerance grade, g- tolerance position.

M16 X 1.5-6g , - tolerance grade and position for both pitch and major diameter are identical.

Table 9-5. Tolerances for major and minor diameters of Metric thread.
(ISO 68 and ISO R1501-1970.)

Pitch P, mm	Td for **external** thread, μm			TD_1 for **internal** thread, μm				
	Tolerance grade							
	4	6	8	4	5	6	7	8
0.5	67	106	---	90	112	140	180	---
0.7	90	140	---	112	140	180	224	---
0.8	95	150	236	125	160	200	250	315
1	112	180	280	150	190	236	300	375
1.25	132	212	335	170	212	265	335	425
1.5	150	236	375	190	236	300	375	475
1.75	170	265	425	212	265	335	425	530
2	180	280	450	236	300	375	475	600
2.5	212	335	530	280	335	450	560	710
3	236	375	600	315	400	500	630	800
3.5	265	425	670	355	450	560	710	900
4	300	475	750	375	475	600	750	950
4.5	315	500	800	425	530	670	850	1060
5	335	530	850	450	560	710	900	1120
5.5	335	560	900	475	600	750	950	1180
6	375	600	950	500	630	800	1000	1250

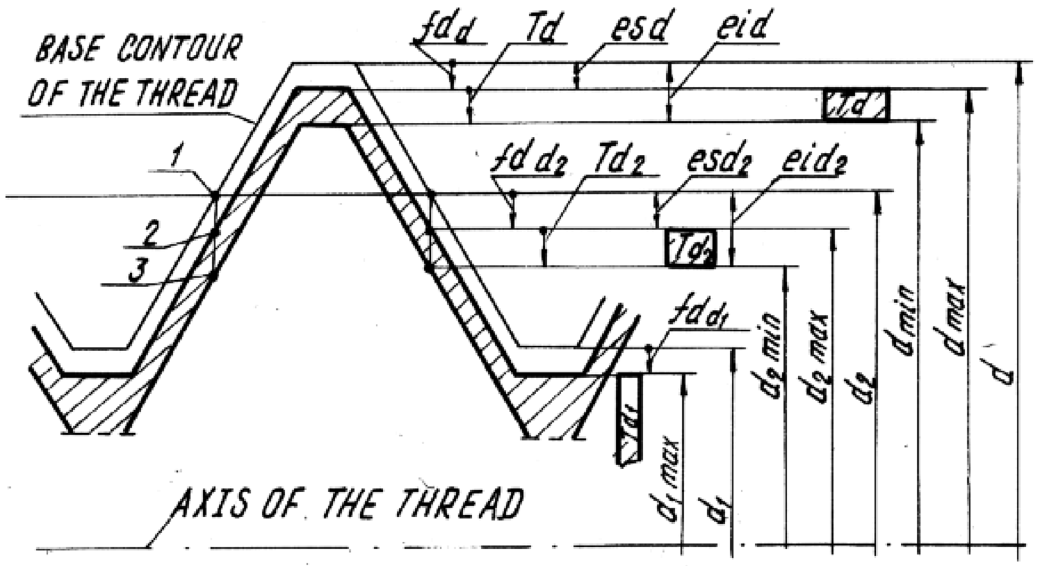


Fig. 9-5. A scheme of tolerance zones of an external screw thread.

Table 9-6. Tolerances for pitch diameter of Metric external thread.
(ISO 68 and ISO R1501-1970.)

Basic diameter (d_2) of a thread, mm	Pitch P, mm	Td_2, µm, for the tolerance grade						
		3	4	5	6	7	8	9
2.8 -5.6	0.5	38	48	60	75	95	---	---
	0.7	45	56	71	90	112	---	---
	0.8	48	60	75	95	118	150	190
5.6 - 11.2	1	56	71	90	112	140	180	224
	1.25	60	75	95	118	150	190	236
	1.5	67	85	106	132	170	212	265
11.2 - 22.4	1.25	67	85	106	132	170	212	265
	1.5	71	90	112	140	180	224	280
	1.75	75	95	118	150	190	236	300
	2	80	100	125	160	200	250	315
	2.5	85	106	132	170	212	265	335
22.4 - 45	1.5	75	95	118	150	190	236	300
	2	85	106	132	170	212	265	335
	3	100	125	160	200	250	315	400
	3.5	106	132	170	212	265	335	425
	4	112	140	180	224	280	355	450
	4.5	118	150	190	236	300	375	475
45 - 90	3	106	132	170	212	265	335	425
	4	118	150	190	236	300	375	475
	5	125	160	200	250	315	400	500
	5.5	132	170	212	265	335	425	530
	6	140	180	224	280	355	450	560

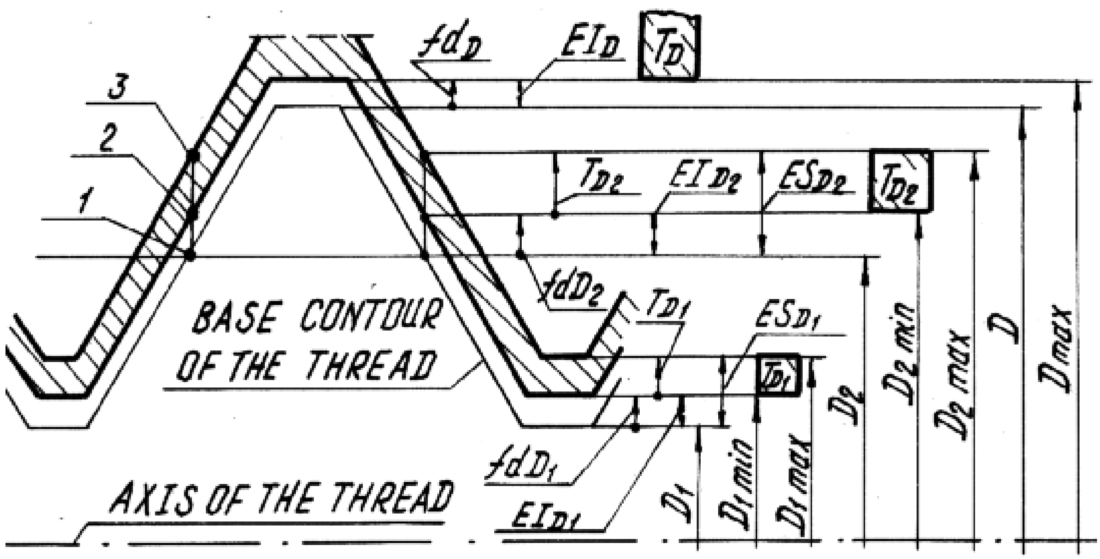


Fig. 9-6. A scheme of tolerance zones of an internal thread.

Table 9-7. Tolerances for pitch diameter of Metric internal thread.
(ISO 68 and ISO R1501-1970.)

Basic diameter (D_2) of a thread, mm	Pitch P, mm	TD_2, μm, for the tolerance grade				
		4	5	6	7	8
2.8 - 5.6	0.5	63	80	100	125	---
	0.7	75	95	118	150	---
	0.8	80	100	125	160	200
5.6 - 11.2	1	95	118	150	190	236
	1.25	100	125	160	200	250
	1.5	112	140	180	224	280
11.2 - 22.4	1.25	112	140	180	224	280
	1.5	118	150	190	236	300
	1.75	125	160	200	250	315
	2	132	170	212	265	335
	2.5	140	180	224	280	355
22.4 - 45	1.5	125	160	200	250	315
	2	140	180	224	280	355
	3	170	212	265	335	425
	3.5	180	224	280	355	450
	4	190	236	300	375	475
	4.5	200	250	315	400	500
45 - 90	3	180	224	280	355	450
	4	200	250	315	400	500
	5	213	265	335	425	530
	5.5	224	280	355	450	560
	6	236	300	375	475	600

Example: it is nesessary to define limit diameters of a screw and a nut for the thread junction M12 x 1.5 -7F8F/6g8g and draw a scheme of tolerance zones.

1. Internal thread: M12 x 1.5 -7F8F.

1.1. The basic major diameter D = d = 12 mm, pitch P = 1.5 mm.

1.2. We define the basic pitch diameter D_2 = D - 1 + 0.026 = 11.026 mm. (See

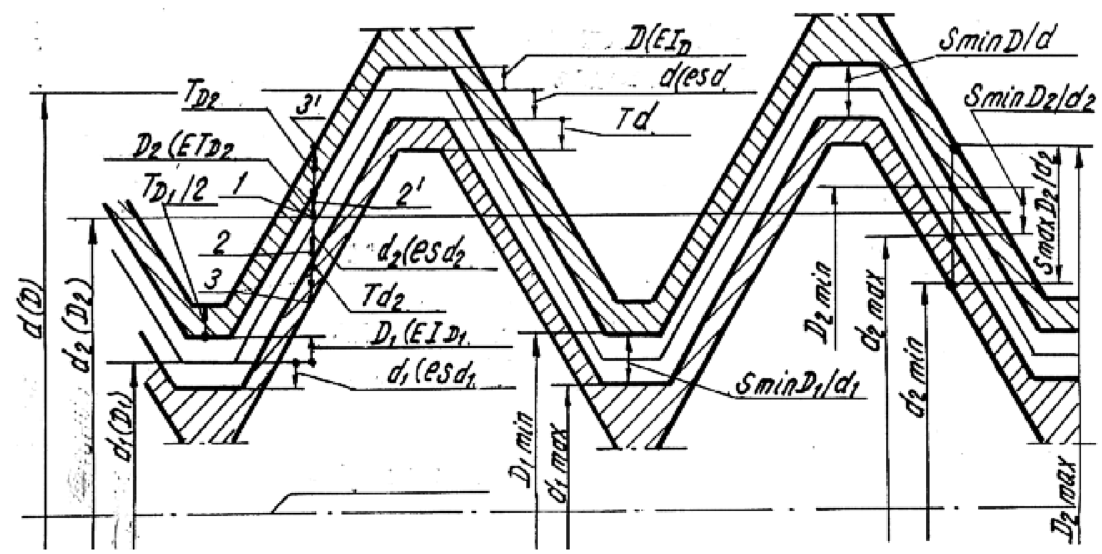


Fig. 9-7. Tolerances zones of thread fit.

Table 9-3.)

1.3. We define the basic minor diameter $D_1 = D - 2 + 0.376 = 10.376$ mm. (See Table 9-3.)

1.4. We define the fundamental deviation FD of the pitch diameter which is equal to the lower deviation EI in this case: $FD_{D2} = EI_{D2} = +45$ μm = +0.045 mm. (See Table 9-4: letter F, pitch P = 1.5 mm.)

1.5. We draw the basic profile of the thread and the basic diameters D, D_2, D_1. (See Fig. 9-8.)

1.6. We lay the lower deviation of the pitch diameter $EI_{D2}/2$ from the point of intersection the pitch diameter line D_2 with the basic thread side, Fig. 9-8. We draw a line parallel to the basic thread side. This is the **lower** limit of the tolerance zone of the pitch diameter.

1.7. We define the **minimum** limit pitch diameter of the internal thread: $D_{2\,min} = D_2 + EI_{D2} = 11.026 + (+0.045) = 11.071$ mm.

1.8. We define the tolerance of the pitch diameter: $T_{D2} = 236$ μm = 0.236 mm. (See Table 9-7: tolerance grade 7, D = 12 mm, P = 1.5 mm.)

1.9. We lay the tolerance of the pitch diameter $T_{D2}/2$, Fig. 9-8. We draw a line parallelly to the basic thread side. This is the **upper** limit of the tolerance zone of the pitch diameter.

1.10. We define the **maximum** limit pitch diameter of the internal thread: $D_{2\,max} = D_{2\,min} + T_{D2} = 11.071 + 0.236 = 11.307$ mm.

1.11. We define the **upper** deviation ES of the pitch diameter of the internal thread: $ES_{D2} = EI_{D2} + T_{D2} = +0.045 + 0.236 = +0.281$ mm.

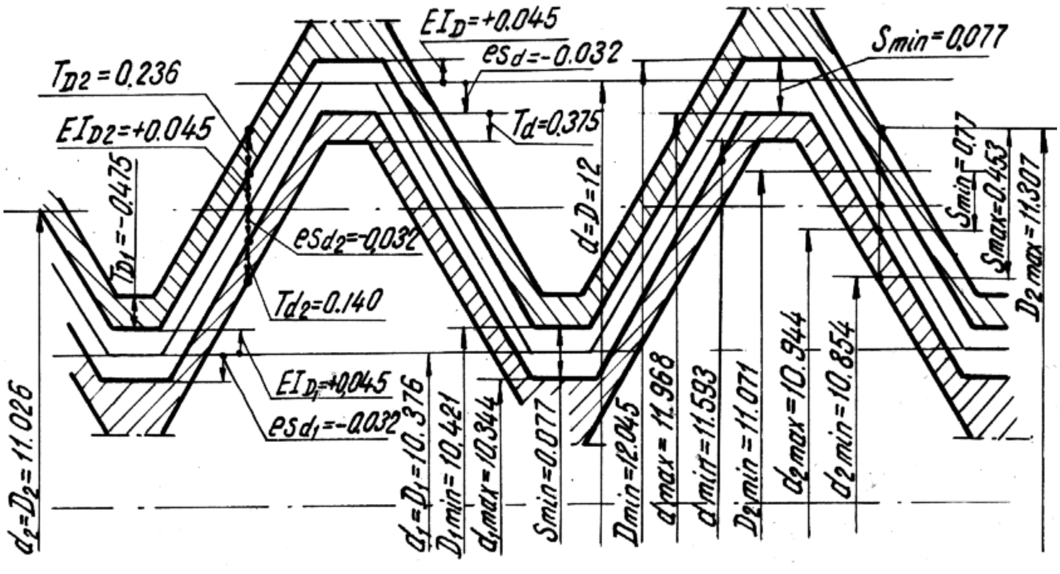


Fig. 9-8. A scheme of tolerance zones of the external and the internal threads for thread fit M12 X 1.5 -7F8F/6g8g.

1.12. We define the fundamental deviation FD of the minor diameter which is equal to a lower deviation EI in this case: $FD_{D1} = EI_{D1} = +45$ μm = +0.045 mm. (See Table 9-4: letter F, pitch P = 1.5 mm.)

1.13. We lay the lower deviation of the minor diameter $EI_{D1}/2$ from the basic minor diameter line, Fig. 9-8. We draw a line parallel to the axis of thread. This is the **lower** limit of the tolerance zone of the minor diameter.

1.14. We define the **minimum** limit minor diameter of the internal thread: $D_{1\ min} = D_1 + EI_{D1} = 10.376 + (+0.045) = 10.421$ mm.

1.15. We define the tolerance of the minor diameter: $T_{D1} = 475\ \mu m = 0.475$ mm. (See Table 9-7: tolerance grade 8, P = 1.5 mm.)

1.16. We lay the tolerance of the minor diameter $T_{D1}/2$ from the **minimum** limit minor diameter line, Fig. 9-8. We draw a line parallelly to the axis of thread. This is the **upper** limit of the tolerance zone of the minor diameter.

1.17. We define the **maximum** limit minor diameter of the internal thread: $D_{1\ max} = D_{1\ min} + T_{D1} = 10.421 + 0.475 = 10.896$ mm.

1.18. We define the fundamental deviation FD of the major diameter which is equal to a lower deviation EI in this case: $FD_D = EI_D = +45\ \mu m = +0.045$ mm. (See Table 9-4: letter F, pitch P = 1.5 mm.)

1.19. We lay the lower deviation of the major diameter $EI_D/2$ from the basic major diameter line, Fig. 9-8. We draw a line parallel to the axis of thread. This is the **lower** limit of the tolerance zone of the major diameter.

1.20. We define the **minimum** limit major diameter of the internal thread: $D_{min} = D + EI_D = 12 + (+0.045) = 12.045$ mm.

1.21. The **maximum** limit **major** diameter of the **internal** thread **is not specified**.

2. External thread: M12 X 1.5- 6g8g.

2.1. Basic major diameter d = 12 mm, pitch P = 1.5 mm.

2.2. We define basic pitch diameter $d_2 = D_2 = d - 1 + 0.026 = 11.026$ mm. (See Table 9-3.)

2.3. We define basic minor diameter $d_1 = D_1 = d - 2 + 0.376 = 10.376$ mm. (See Table 9-3.)

2.4. We define the fundamental deviation fd of the pitch diameter which is equal to the upper deviation es in this case: $fd_{d2} = es_{d2} = -32\ \mu m = -0.032$ mm. (See Table 9-4: letter g, pitch P = 1.5 mm.)

2.5. We draw the basic profile of the thread and basic diameters d = D, $d_2 = D_2$, $d_1 = D_1$. (See Fig. 9-8.)

2.6. We lay the upper deviation of the pitch diameter $es_{d2}/2$ from the point of intersection the pitch diameter line d_2 with the basic thread side, Fig. 9-7. We draw a line parallel to the basic thread side. This is the **upper** limit of the tolerance zone of the pitch diameter.

2.7. We define the **maximum** limit pitch diameter of the external thread: $d_{2\ max} = d_2 + es_{d2} = 11.026 + (-0.032) = 10.994$ mm.

2.8. We define the tolerance of the pitch diameter: $T_{d2} = 140\ \mu m = 0.140$ mm. (See Table 9-6: tolerance grade 6, d = 12 mm, P = 1.5 mm.)

2.9. We lay the tolerance of the pitch diameter $T_{d2}/2$, Fig. 9-8. We draw a line parallel to the basic thread side. This is the **lower** limit of the tolerance zone of the pitch diameter.

2.10. We define the **minimum** limit pitch diameter of the external thread: $d_{2\,min} = d_{2\,max} - T_{D2}$ = 10.994 - 0.140 = 10.854 mm.
2.11. We define the **lower** deviation of the pitch diameter: $ei_{d2} = es_{d2} - T_{d2}$ = -0.032 - 0.140 = -0.172 mm.
2.12. We define the fundamental deviation fd of the major diameter which is equal to the upper deviation es in this case: $fd_d = es_d$ = -32 μm = -0.032 mm. (See Table 9-4: letter g, pitch P = 1.5 mm.)
2.13. We lay the upper deviation of the major diameter $es_d/2$ from the basic major diameter line, Fig. 9-8. We draw a line parallel to the axis of thread. This is the **upper** limit of the tolerance zone of the major diameter.
2.14. We define the **maximum** limit major diameter of the external thread: $d_{max} = d + es_d$ = 12 + (-0.032) = 11.968 mm.
2.15. We define the tolerance of the major diameter: T_d = 375 μm = 0.375 mm. (See Table 9-6: tolerance grade 8, P = 1.5 mm.)
2.16. We lay the tolerance of the major diameter $T_d/2$ from the **maximum** limit major diameter line, Fig. 9-8. We draw a line parallel to the axis of thread. This is the **lower** limit of the tolerance zone of the major diameter.
2.17. We define the **minimum** limit major diameter of the external thread: $d_{min} = d_{max} - T_d$ = 11.968 - 0.375 = 11.593 mm.
2.18. We define the fundamental deviation fd of the minor diameter which is equal to the upper deviation es in this case: $fd_{d1} = es_{d1}$ = -32 μm = -0.032 mm. (See Table 9-4: letter g, pitch P = 1.5 mm.)
2.19. We lay the upper deviation of the minor diameter $es_{d1}/2$ from the basic minor diameter line, Fig. 9-8. We draw a line parallel to the axis of thread. This is the **upper** limit of the tolerance zone of the minor diameter.
2.20. We define the **maximum** limit minor diameter of the external thread: $d_{1\,max} = d_1 + es_{d1}$ = 10.376 + (-0.032) = 10.344 mm.
2.21. The **minimum** limit **minor** diameter of the **external** thread **is not specified**.
3. Clearances:
3.1. On the pitch diameter: $S_{D2\,max} = D_{2\,max} - d_{2\,min}$ = 11.307 - 10.854 = 0.453 mm or $S_{D2\,max} = ES_{D2} - ei_{d2}$ = +0.281 - (-0.172) = 0.453 mm; allowance (possitive): $S_{D2\,min} = D_{2\,min} - d_{2\,max}$ = 11.071 - 10.994 = 0.077 mm.
3.2. On the major diameter: $S_{D\,max} = D_{max} - d_{min}$ - is not calculated (D_{max} is not specified); $S_{D\,min} = D_{min} - d_{max}$ = 12.045 - 11.968 = 0.077 mm.
3.3. On the minor diameter: $S_{D1\,max} = D_{1\,max} - d_{1\,min}$ - is not calculated ($d_{1\,min}$ is not specified); $S_{D1\,min} = D_{1\,min} - d_{1\,max}$ = 10.421 - 10.344 = 0.077 mm.

Chapter 10. Other Forms of Screw Threads and Screw Threads Measurement

10.1. Other Forms of Screw Threads

Square thread sometimes is used on vise screws, heavy jack screw and similar items. The depth, width, and space between square threads are equal, Fig.10 -1A. The square thread cannot be made efficiently with dies, taps or milling cutters unless the thread form is modified. Acme threads have largely replaced the use of square threads in machine design and construction. **Acme threads** have a 29^{o} thread angle, as shown in Fig.10-1B. They are used to produce traversing movements on machine tools, vises, etc. Although acme threads are not quite as strong as the square threads, they are much easier to machine. They may be cut with dies, taps, milling cutters, grinders, and a single-point tool in a lathe. The three classes (designated 2G, 3G, and 4G) are provided with clearances on all diameters for free movement.

Brown and Sharp Worm Thread. Worm threads are used in worm gear drive systems which have three principal uses: (1) to effect a large reduction in shaft R.P.M.; (2) to provide a sharp increase in mechanical advantage; (3) for steady and efficient power transmission.

The general form and the basic dimensions for the Brown and Sharp worm thread are similar to the acme thread, Fig. 10-IB. Both threads have a 29° included angle, but the Brown and Sharp thread is deeper. The widths of the crests and roots of the thread are also different. The formulas are: D= 0.6866 x P; C= 0.335 x P; R= 0.310 x P.

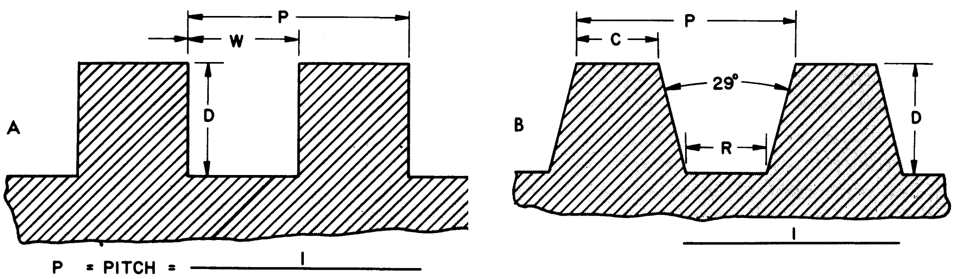


Fig.10-1. Cross sections of several forms of screw threads.
A- Square thread: P- pitch=1/ No. Threads per Inch;
D- depth = 0.5 x P; W- width = 0.5 x P.
B- Asme thread: D = 0.5 x P + 0.01 Inch; C- flat on top of thread = P x 0.3707; R- flat on bottom = (P x 0.3707) - 0.0052.

Pipe Threads. The most commonly used system of pipe threads is the American Standard or American National pipe thread. Two types of pipe threads are included in this system — the American Standard taper pipe thread (NPT) and the American Standard straight pipe thread (NPS). A third type which is a variation of the NPT is the American standard Dryseal pipe thread (NPTF). Tables in standard handbooks for the machinist include the basic pipe thread dimensions, nominal pipe sizes with the number of threads per inch, outside diameters, pitch diameters, and other dimensions.

A tapered pipe thread (NPT) is similar to the American National thread. It has a 60° angle between the sides of the threads, and it has flattened crests and roots. It differs in that the threads are tapered 3/4" per foot of length from the small end of the thread toward the large end.

Both the external and the internal threads are tapered. This permits drawing the joint up tightly for a rigid joint. However, a pipe compound must be used to seal the clearance space which exists between the crests and roots of the mating threads in order to prevent leakage of liquid or gas under pressure.

Straight pipe threads (NPS) sometimes are used in couplings which join pipes having tapered pipe threads. This type of pipe assembly often is satisfactory for low-pressure lines when a pipe compound is used and when the line is free of vibration.

American standard Dryseal pipe threads (NPTF) are used for those pressure joints where the use of a pipe sealing material is objectionable. The external thread is tapered, while the internal thread may be tapered or straight. The crest flats are equal to or less than the root flats of the mating thread, thus causing physical contact between the crests and the roots when turned up hand-tight. When turned wrench-tight, the threads mash together causing a pressure-tight seal.

Taps are used to cut internal threads in holes. The process of cutting internal threads is called **tapping.** External threads are cut with a die, and the process is called **threading.**

Taps are made of hardened tool steel, either carbon steel or high-speed steel. Carbon steel taps usually are used for hand-tapping operations. High-speed steel taps are used for both hand- and machine-tapping operations.

The flutes on the tap provide for the cutting edges and also provide space for the chips. Standard hand taps usually have four flutes, but some taps have two or three flutes. The square end on the tap is used for holding the tap with a wrench or other holding device. Taps are hard and brittle, and they break quite easily when excessive force is applied to them.

The three basic types of hand taps are the **taper tap,** the **plug tap,** and the **bottoming tap.** The only basic difference in the three taps is the number of threads which are ground to a taper on the end of the tap. Taps are ground tapered in order to start and cut the threads more easily. Most of the actual cutting of the thread is performed by the chamfered threads on the end of the tap.

As the name implies, **hand taps** are designed for hand-tapping operations. They can be purchased singly or in sets of three, including the **taper, plug,** and **bottoming** taps for each thread size and pitch.

The taper tap has 8 to 10 threads tapered. It is used to tap **open or through** type holes, as in Fig. 10-2A. It also is used as the first step in tapping closed or blind holes in hand-tapping operations, Fig. 10-2B and C. Tool life and breakage largely depend on the length of the tapered threads on the tap. Less power is required and less strain is exerted on a tapered tap than on a plug or bottoming tap. Therefore, the tapered tap should be used whenever possible. The plug tap has 3 to 5 threads tapered. The bottoming tap has 1 to I.5 threads tapered. It is used after the plug tap for tapping to the bottom of a hole as in Fig. 10-2B.

The tapered tap is used first in tapping to the bottom of the hole, and it is followed with the plug tap and the bottoming tap. For most materials, plug taps can be successfully used in place of taper taps. Plug taps are preferred for machine tapping because through holes and blind holes can be tapped in one operation, as in Fig. 10-2C, thus effecting considerable cost savings.

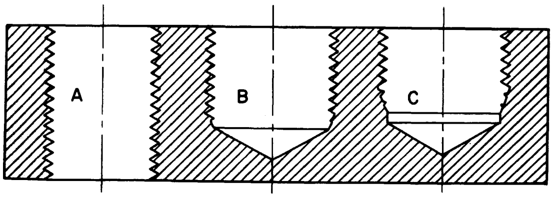


Fig.10-2. Types of holes commonly tapped. A-open or through; B-blind bottoming; C-blind but not bottoming.

Serial hand taps usually are produced in sets of three (numbered 1, 2, and 3) and are identified with one, two, or three rings on the shank of the tap near the square. Serial taps are used for hand tapping deep holes in tough metals. They are similar to the taper, plug, and bottoming taps in general appearance, but they differ in sizes of the pitch diameter and major diameter.

Each tap is designed to remove a certain portion of the metal in cutting a thread. This procedure eliminates excessive strain on the tap and aids in preventing tap breakage. The No. 1 tap is used first for cutting a shallow thread. This is followed by the No. 2 tap, which cuts deeper, and finally, the No. 3 tap is used to cut the thread to full depth.

Major diameters of machine screw sizes are listed in tables of tap drill sizes.

Tap wrenches are used to turn the tap for hand-tapping operations. The wrenches have adjustable jaws which enable the tap to be tightened in the wrench and held firmly. There are two common types of tap wrenches — the **T-handle** tap wrench and the **straight handle** tap wrench. Both types are available in several sizes. The T-handle tap wrench usually is used for holding smaller diameter taps than the straight handle tap wrench. Taps of a small size may be broken easily when used with tap wrenches which are too large. Tap wrenches are also used to drive reamers, screw extractors, and other tools which are turned by hand. Taps, tap wrenches, dies, and die stocks may be purchased in sets called **screw plates**.

10.2. Screw Thread Measurement

The size and accuracy of screw threads may be measured with thread micrometers (Fig. 10-5), ring thread gages (Fig. 10-6, A), screw thread plug gages (Fig. 10-6, B), roll thread snap gages (Fig. 10-7), thread comparators of various types, optical comparators, and by the three-wire method (Fig. 10-8).

The emphasis in thread measurement is always on the measurement of the pitch diameter. All methods of measurement of the pitch diameter must provide contact only on the pitch diameter and do not feel errors of the pitch and of the thread angle. That is why the **not-go gage** and the contact points have cutted down profile (Figs. 10-3, 10-4). The **not-go gage** is designed to check only the pitch diameter. The **go gage** is designed to check the maximum pitch diameter, flank angle, lead, and the clearance at the minor diameter simultaneously. The **go gage** must feel errors of the thread angle and the pitch. That is why the **go gage** has full profile and large amount of threads.

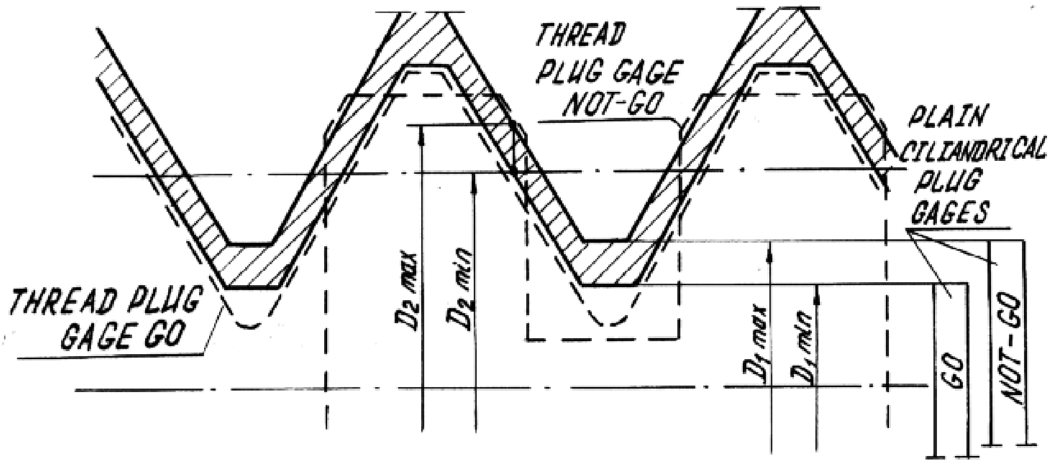


Fig. 10-3. Profiles of thread plug gages GO and NOT-GO.

The limits and tolerances on the pitch diameter largely determine the fit of screw threads. Since the pitch diameter of thread ring gages and snap roll gages is difficult to determine accurately by other methods, these gages are set or fitted with the use of accurate plug gages.

Tapped holes are checked for the correct fit with thread limits plug gages. The double-end gages provide a go gage on one end and a not-go gage on the opposite end. The go gage is always the longer of the two, and it has a chip groove for cleaning the threads in the tapped hole being measured.

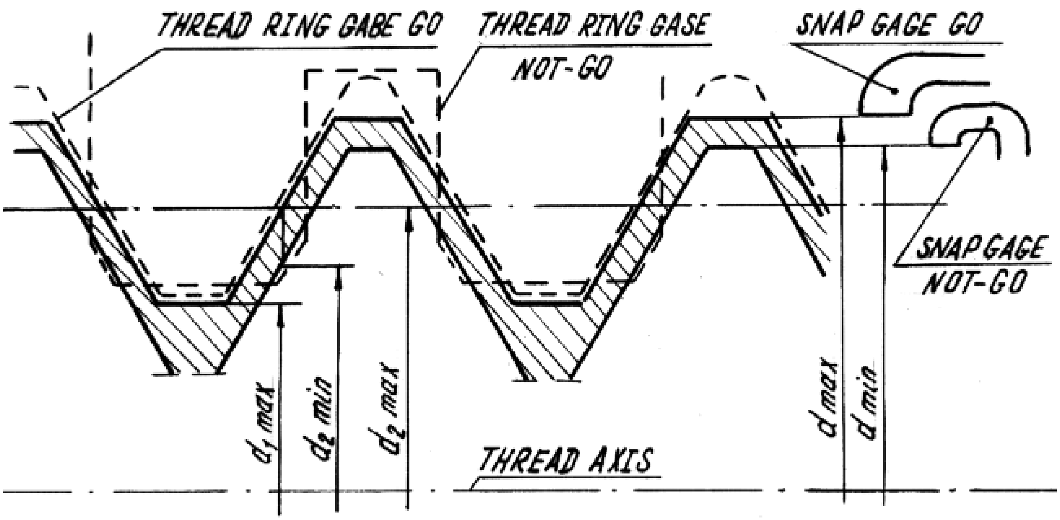


Fig. 10-4. Profiles of thread ring gages GO and NOT-GO.

The thread limits plug gage (Fig. 10-6, B) is used to determine whether the pitch diameter of a tapped thread is within limits for a specified class of thread. The diameter of the go gage is minimum diameter, usually basic (tolerance position H), and the not-go gage is maximum diameter.

The thread is gaged by having the go gage enter the tapped hole the full length of the gage. The not-go gage may or may not enter. If it does, it should have a snug fit on or before the third thread, thus indicating that the hole is the maximum size permitted for the specified fit. If this gage enters farther, the thread is oversize and will not fit properly.

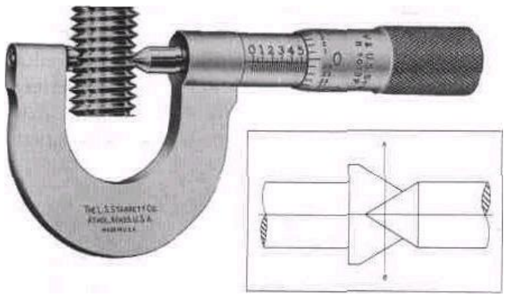
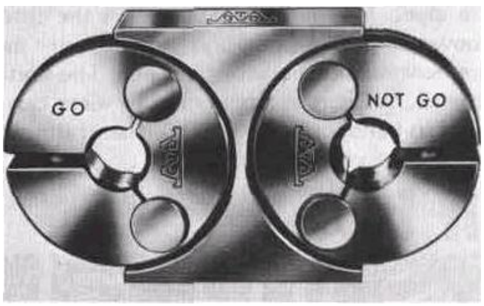


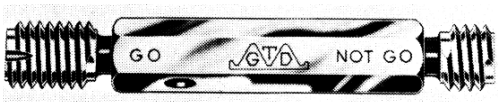
Fig. 10-5. Thread micrometer.

The pitch diameter of 60° V-threads may be measured directly with a **thread micrometer**. See Fig. 10-5. The spindle of the micrometer has a 60° conical point, and the anvil has a 60° groove. The anvil point swivels to enable measurement of different pitches.

A given thread micrometer is designed to measure a specific range of screw threads. Care should be taken to select a micrometer with the correct thread range when measuring a specific thread. The micrometer always should be checked for a zero reading before measuring threads. See the inset of Fig. 10-5.



A



B

Fig.10-6. Thread ring (A) and plug (B) gages.

The accuracy of an external thread may be checked with a pair of **thread ring gages**. The pair includes a go gage and a not-go gage, Fig. 10-6. The **go gage** is designed to check the maximum pitch diameter, flank angle, lead, and the clearance at the minor diameter simultaneously, Fig. 10-4. The **not-go gage** is designed to check only the pitch diameter to determine whether it is below minimum limits, Fig. 10-4.

To check a thread, both gages are used. If the go gage does not turn on freely, one of the thread elements is not accurate, and the thread will not assemble with the mating part. If the not-go gage turns on the thread, the pitch diameter of the thread is under the specified minimum limits, and the thread will not fit properly with its mating part.

Fig. 10-7. Roll thread snap gages.

External threads may be checked rapidly for accuracy with a **roll thread snap gage**, Fig. 10-7. The gage illustrated is the open-face type which may be used close to shoulders. Right- and left-hand threads can be checked with the same gage.

The outer or **go rolls** are set to the maximum pitch diameter limits, and they check all thread elements simultaneously. The inner or **not-go rolls** are set to minimum pitch diameter limits and check only the minimum pitch diameter. Screw threads which are within the correct size limits pass through the go rolls and are stopped by the not-go rolls.

A **three-wire method (**Fig. 10-8) is a method of measuring the pitch diameter of external threads. The three-wire method requires the use of an ordinary outside micrometer and three accurately sized wires. A different **best wire size** is recommended for each pitch and diameter combination. The best wire size is determined by calculation or by selection from a chart of recommended wire sizes. The best wire size is:

d_{wb}=0.5P/(cosα/2), (10.1)

where: P - pitch of the thread; α - thread angle.

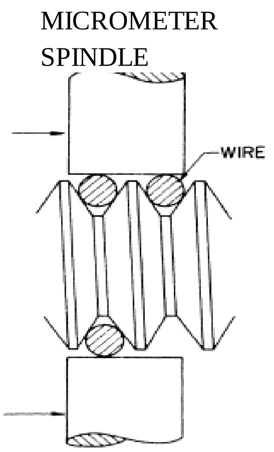


Fig. 10-8. Three-wire thread measurement.

The wires are placed in the thread grooves as in Fig. 10-8, and a micrometer measurement across the wires is made. The correct dimensions for measurement over wires can be found in handbooks for machinists, or the pitch diameter can be calculated:

$d_2 = M - 3d_w + 0.866P$, (10.2)

where: M - measured size; d_w - diameter of wire: P - pitch.

Because the three-wire method is more cumbersome and time consuming to use, many machinists prefer to check the pitch diameter of external threads with a ring gage, thread micrometer, or other instrument. However, the three-wire method is considered to be more accurate than the use of many gages designed for this purpose.

Numerous special thread gaging and measuring devices are available for use in measuring thread elements. Some of these devices measure only one element, such as the pitch diameter. Others measure several thread elements simultaneously.

An external thread comparator is used to inspect external threads by means of a single visual reading between indicator tolerance hands. The comparator checks for errors in lead, thread angle, and pitch diameter. The reading on the indicator dial shows whether the cumulative error of all these elements, combined, falls between the high and low limits for a class-of-thread tolerance. The comparator is set to a given size with the use of a master plug thread gage, and the tolerance hands are set for the desired class of thread. An **internal thread comparator** works on the same principle as the external comparator.

Chapter 11. Control of gears

11.1. Types of gears

A gear is a wheel into which teeth have been cut, see Fig. 11-1. Although friction wheels will transmit motion and power, they are inefficient because slippage occurs even under heavy loads.

Gears are used on nearly all types of machines, ranging from wristwatches and egg beaters to machine tools, automobiles, and heavy manufacturing machinery. Gears may be used to transmit motion between shafts that are parallel, intersecting, or neither parallel nor intersecting. The gear connected to the source of power is called the **driver**, and the one to which motion is transmitted is called the **driven**.

Gears normally are mass-produced on special machines. These machines are of several types, including **form-cutting** machines and **gear-generating** machines. Gear-generating machines usually are further classified according to two types: **gear-shaping** machines and **gear-hobbing** machines.

Machinists in maintenance, job, and experimental machine shops frequently are required to produce gears of various types on a milling machine. The machinist, therefore, should be familiar with some of the basic kinds of gears and the terminology that applies to them.

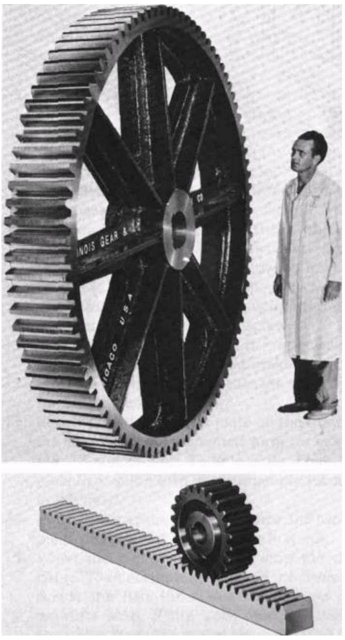


Fig. 11-1. A large spur gear above and a rack and pinion below.

When two gears of unequal size are mated, the smaller one is called a **pinion**. A **gear rack** (Fig. 11-1) is a gear that has teeth spaced along a straight line. This permits rotary motion to be changed to straight-line motion, or vice versa. A basic rack is one that is the basis for a system of interchangeable gears, see Fig. 11-13.

Many types of gears are used in industrial machines. The most common are spur, bevel and miter, internal, helical, and worm gears.

Internal gears have teeth on the inner surface of a cone or cylinder. This type may also have spur, bevel, or helical teeth. An advantage in combining internal and external gears is their compactness. Since the centers of the gears are closer together, less space is required. They also possess increased operational efficiency: more teeth are in mesh, the tooth lines curve in the same direction, and friction is reduced.

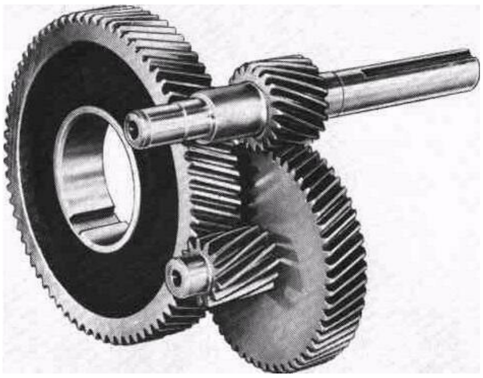


Fig. 11-2. A helical gear train with parallel shafts.

Helical gears are similar to spur gears, except that the teeth form a helix twisting around the body of the gear, Fig. 11-2. The helical teeth provide greater strength and smoother operation at high speeds. The teeth do not hit each other as in the case of spur gears. Instead, they slide across each other, thus reducing noise and vibration. Since several teeth are in contact at the same time, their strength is greater than for spur teeth of the same size.

A disadvantage of helical gear teeth is that increased friction and heat are caused by the sliding action of the teeth. This disadvantage may be overcome with good lubrication. Hence, helical gears that operate at high speeds generally are run in an oil bath.

Herringbone gears have the appearance of a pair of right- and left-hand helical gears located side by side. Since they have both right- and left-hand teeth, they frequently are called **double-helical gears**. Herringbone gears have greater surface contact than other gears of similar size. They therefore have greater load-carrying capacity. They generally withstand continuous, heavy-duty, high-speed operation better than other types of gears.

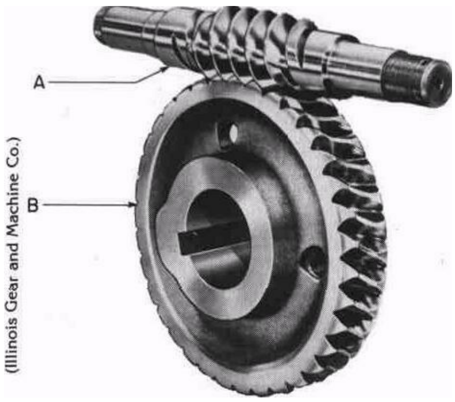


Fig. 11-3. A worm gear mechanism: (A) worm, (B) worm gear.

Worm gears are meshed with a **worm**; the gear and the worm constitute a worm gear mechanism, Fig. 11-3. The teeth on the worm gear are helical and conform with the helix angle of the tooth on the worm. The helical tooth on the worm is a form of thread, similar to an acme thread, and it often is called a **worm thread**.

Worms may have single, double, or triple threads. With the single thread, one revolution of the worm revolves the worm

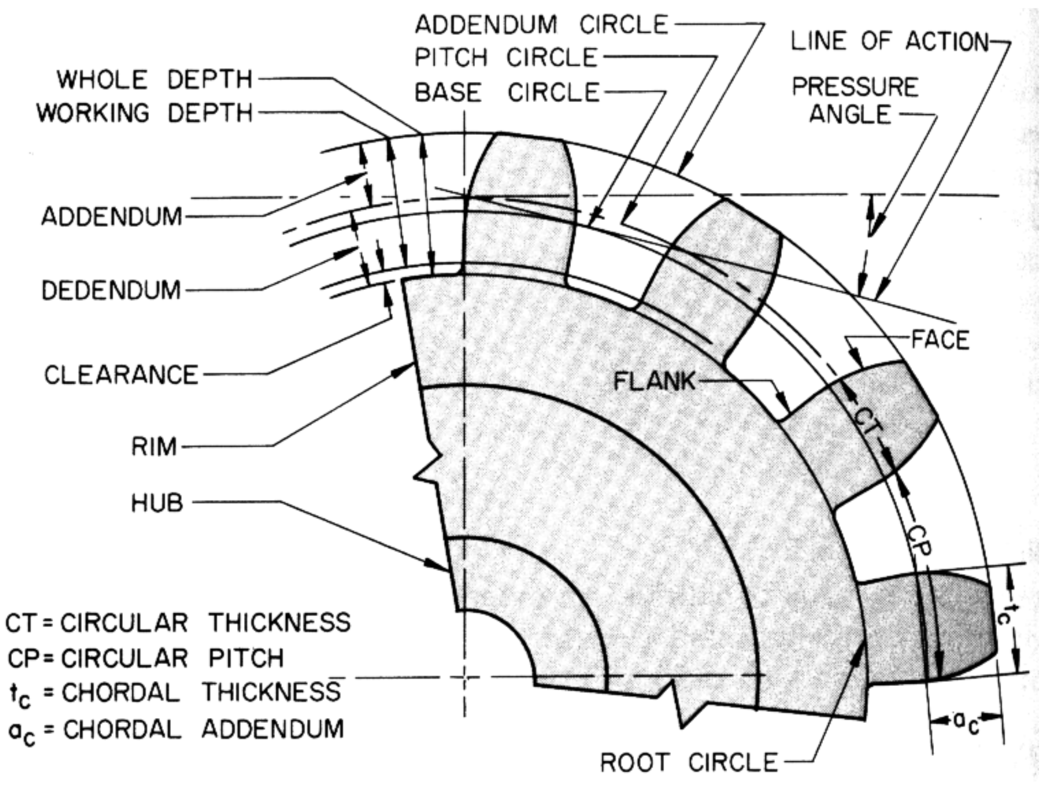


Fig. 11-4. Gear nomenclature.

gear a distance equal to that between a point on one tooth and a corresponding point on the next worm gear tooth, or one **circular pitch**. One revolution of a double-thread worm revolves the gear an amount equal to two teeth on the gear, and so on.

Worm gearing is used largely for speed reduction. The worm gear cannot turn the worm when a single-thread worm is used. This type of gear mechanism is **self-locking.** Engineers take advantage of this feature when they employ worm gears in steering mechanisms, in hoisting equipment, and in other devices.

11.2. Gear Terms and Definitions

Modern gears generally have involute teeth. This means that the shape of the tooth is generated or drawn with an involute curve. Such a curve may be drawn with a pencil inserted in the loop of a string wound about a cylinder and held taut as the string is unwound. This form of tooth has been found to give excellent results in terms of quietness and smoothness of operation.

The size of a gear is given in terms of its diameter at the pitch circle, which is called the pitch diameter d. See Fig. 11-4.

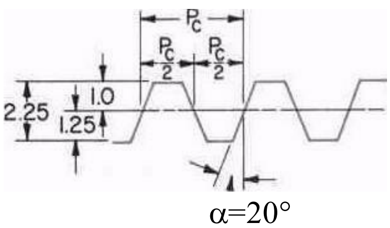
Metric gears are always made according to the module (m) system, with measurements in millimeters. The module represents the amount of pitch diameter per gear tooth, and also corresponds to the addendum, Fig. 11-6. Therefore, the higher the module number, the larger the size of the gear tooth. To say that a metric gear has a module of 2 means that it has 2 mm of pitch diameter for each gear tooth. Thus, a 2m, 40-tooth gear would have a pitch diameter of 2 x 40 = 80 mm (d=mz).

Inch gears are usually made according to the diametral pitch (P) system, with measurements in inches. Diametral pitch represents the number of gear teeth for each inch of pitch diameter.

The term **circular pitch** (p) refers to the distance along the pitch circle on a gear or along the pitch line on a rack, from a point on one tooth to a corresponding point on the next tooth, Fig. 11-6. Circular pitch corresponds to linear pitch on a gear rack.

Diametral pitch (P), pitch diameter (d), amount of teeth (z), and module (m) are all related as shown to the right: $P=m\pi$; $d=mz$.

Involute teeth of spur gears, helical gears, and worms are those in which the profile in a transverse plane (exclusive of the fillet curve) is the involute of a circle.



$\alpha=20°$

Fig.11-5. The Si metric 20° gear tooth form.

The **base circle** is the circle from which involute tooth profiles are derived. (See Fig. 11-4. Continue to refer to this figure throughout these definitions.)

A **pitch circle** is the curve of intersection of a pitch surface of revolution and a plane of rotation. According to theory, it is the imaginary circle that rolls without slipping with a pitch circle of a mating gear.

A **pitch line** corresponds in the cross section of a rack to the pitch circle in the cross section of a gear.

The **addendum circle** coincides with the tops of the teeth in a cross section.

The **root circle** is tangent to the bottoms of the tooth spaces in a cross section.

The **line of action** is the path of contact in involute gears. It is the straight line passing through the pitch point and tangent to the base circles.

Pressure angle (α) is the angle between a tooth profile and the line normal to a pitch surface, usually at the pitch point of profile. This definition is applicable to every type of gear. The term **pressure angle** originally meant an angle between the line of pressure and the pitch circle. In involute teeth, pressure angle is often described as the angle between the line of action and the line tangent to the pitch circles.

Center distance (C) is the distance between parallel axes of spur gears and parallel helical gears or between the crossed axes of crossed helical gears and worm gears. Also, it is the distance between the centers of pitch circles.

Addendum (a) is the height by which a tooth projects beyond the pitch circle or pitch line; also, it is the radial distance between the pitch circle and the addendum circle.

Dedendum (b) is the depth of a tooth space below the pitch circle or pitch line; also, it is the radial distance between the pitch circle and the root circle.

Clearance (c) is the amount by which the dedendum in a given gear exceeds the addendum of its mating gear.

Working depth (h_k) is the depth of engagement of two gears, that is, the sum of their addendums.

Whole depth (h_t) is the total depth of a tooth space, equal to addendum plus dedendum, also equal to working depth plus clearance.

Pitch diameter (D, d) is the diameter of the pitch circle.

Outside diameter (D_O, d_O) is the diameter of the addendum (outside) circle. In a bevel gear, it is the diameter of the crown circle. In a throated worm gear, it is the maximum diameter of the blank. The term applies to external gears.

Root diameter (D_R, d_R) is the diameter of the root circle.

Circular thickness (t_G, t_P) is the length of arc between the two sides of a gear tooth, on the pitch circle unless otherwise specified.

Chordal thickness (t_C) is the length of the chord subtending a circular-thickness arc.

Chordal addendum (a_C) is the height from the top of the tooth to the chord subtending the circular-thickness arc.

Number of teeth or **threads** (z or N) is the number of teeth contained in the whole circumference of the pitch circle.

Gear ratio (m_G) is the ratio of the larger to the smaller number of teeth in a pair of gears.

Full-depth teeth are those in which the working depth equals 2 x the metric module.

Stub teeth are those in which the working depth is less than 2 x the metric module.

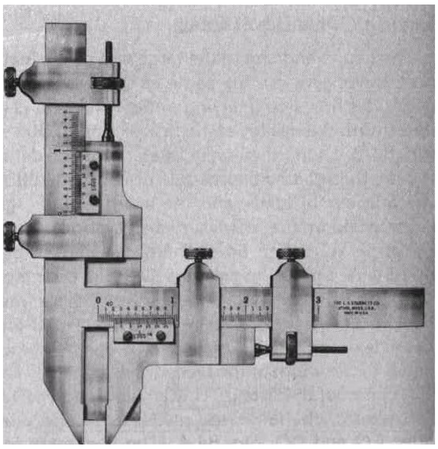


Fig. 11-6. A gear-tooth vernier caliper.

The chordal thickness and the chordal addendum of spur gear teeth may be accurately checked for size with a gear-tooth vernier caliper, Fig. 11-6. The values for these parts may be secured in standard handbooks for machinists. The vertical scale on the vernier catiper is first set at the handbook value for the chordal addendum (sometimes called corrected addendum). The caliper is then fit onto the tooth, and the chordal thickness is measured by using the sliding vernier scale.

11.3. Gear-Tooth Forms

In order for mating gears of the same module (or diametral pitch) to mesh with a smooth, quiet, rolling action, they must have the proper gear-tooth form. Two basic forms (or curves) for gear teeth are used - one with an involute curve, and a second that is a composite of the involute curve and cycloidal curves.

Mating gears with either of these forms will roll together smoothly and quietly without interference when operated at the prescribed pressure angle. Several systems of gear-tooth form have been standardized, and each system is designed to operate at a specified pressure angle, usually at 14-1/2°, 20°, or 25°.

Gears with the involute curve tooth design are the most widely used. The teeth of gears with the composite tooth form are very similar to those of the involute form, except for the design of basic rack teeth. This rack has an involute curve in the area of the pitch line, but it is modified slightly with cycloidal curves in areas above and below the pitch line. The modification prevents interference between mating gears that have a small number of teeth.

Gears with the composite tooth form normally are produced on milling machines, with form-type rotary milling cutters. The use of this tooth form largely is limited to the production of gears in small job shops, maintenance shops, and shops where small numbers of gears are produced. This form of gear also may be produced by hobbing or with other gear-generating machines.

The approved SI metric gear tooth form is shown in Fig. 11-5. Rack teeth with this form have straight sides and are full depth, providing the basis for a complete system of interchangeable gears. The pressure angle of 20° is in conformance with international agreement that 20° is the most versatile pressure angle. The addendum is equal to the module m, which corresponds to the reciprocal of the American diametral pitch. The dedendum is 1.250 m, corresponding to recent American practice. Root radius is some-what greater than in American practice, and the tip radius deviates from American practice, which does not specify rounding. This does not, however, prohibit the American practice of specifying a tip radius as near zero as practicable.

In spite of the nearly identical design of the SI metric 20° full-depth tooth form and the American Standard 20° full-depth tooth form, the gears **are not interchangeable**, due to differences in circular pitch.

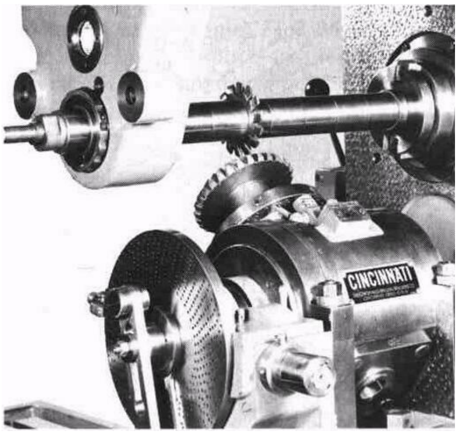


Fig. 11-7. Cutting bevel gear teeth on a gear blank mounted on a dividing head.

Normally, bevel gears are cut on special bevel gear-cutting machines. However, bevel gears can be cut with sufficient accuracy for many applications by using a standard milling machine. The machining setup is made by mounting the bevel gear blank in the dividing head chuck or spindle, Fig. 11-7.

Cutters used for cutting bevel gears on a milling machine are similar to those used for cutting spur gears. However, they are thinner, in order to conform to the profile of the teeth at the small end of the gear. Bevel gear cutters are made in sets, and are stamped with the word **bevel**. Instructions for the selection of bevel gear cutters generally are included in cutter manufacturer's catalogs or in standard handbooks for machinists.

11.4. Fixing of accuracy

12 degrees of accuracy of gears are established for gears and transfers designated in decreasing order of accuracy: 1, 2..., 12. The norms of admitted deviations of parameters are established for each degree of accuracy. They determine 4 norms of accuracy: (1) kinematics accuracy of gears and transfers, (2) smoothness of work, (3) contacts teeth of gears of transfer, (4) kinds of gears interfaces of transfer. The degree of accuracy on each norm of accuracy is underlined in a conditional designation of gears strictly under the order. Last two letters specify a kind of interface, which accuracy is designated not in figures, and letters. Each norm of accuracy does not depend on others, and they can have various degrees of accuracy, for example, 7-8-7-Bc. Each norm of accuracy can be controlled by several methods and devices, which choice depends on a

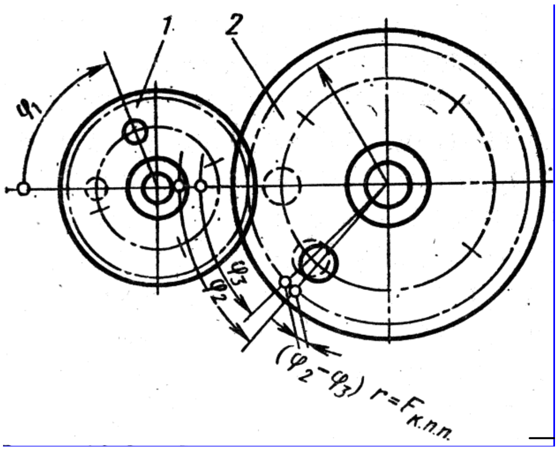


Fig. 11-8. A scheme of definition of gear transfer kinematic error.

degree of accuracy and purpose of transfer.

The **kinematics error of transfer** is a difference between the actual and nominal angles of turn of a driven gear of transfer. It is expressed in linear sizes of length of an arch of its pitch circle, $F_{k.e.t.}=(\varphi_2-\varphi_3)r$, where r is a radius of pitch circle of a driven gear; $\varphi_3=\varphi_3 z_1/z_2$; φ_1 is an actual angle of turn of a driver gear; z_1 and z_2 are teeth numbers of driver 1 and driven 2 gears. Greatest kinematics error of transfer F'_{ior} is the greatest algebraic difference of meanings of transfer kinematics error for a complete cycle of change of a relative situation of gears. The complete cycle occurs within the limits of number of revolutions of the large gear equals the quotient from division of the teeth number of a smaller gear by the greatest common divisor of teeth numbers of both transfer gears, i.e. on an angle $\varphi_2=2\pi z_1/x$. For example, for $z_1=20$ and $z_2=40$ the greatest common divisor is x=30 and $\varphi_2=2\pi 30\backslash 30=2\pi$.

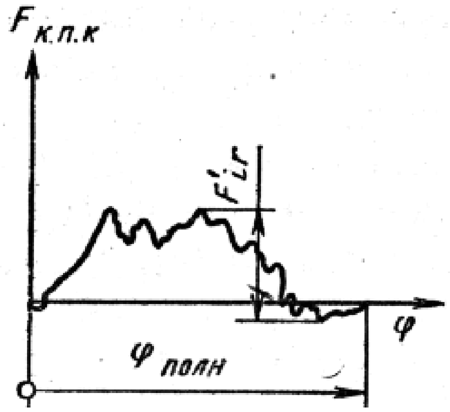


Fig. 11-9. A definition of the greatest gear wheel kinematic error F'_{ir}.

The greatest kinematics error of transfer is limited by the tolerance F'_{io}. This tolerance is equal to the sum of the kinematics errors tolerances of its gears, $F'_{io} = F'_{i1} + F'_{i2}$.

The kinematics error of gear $F_{k.e.g.}$ is the difference between the actual and nominal angles of turn of a gear on its work axes and driven with an accurate (a measuring) gear. It is expressed in linear sizes of length of an arch of its pitch circle.

The **kinematics error of gear** is determined from one revolution of a gear. The kinematics error of gear, made on machine tools by a method of round, occurs because of discrepancy of the center of the basic circle of gears to a working axis of its rotation, error of round circuits of the machine tool, discrepancy of the tool and its installation, etc. The kinematics accuracy of gears depends on errors, which total influence is found once from one revolution of a gear. It is a round error, saved error of a step F_{Pkr}, radial palpation of a gear ring F_{rr}, fluctuation of common normal length $F_{vWr\ ir}$ (Fig. 11-11,b) and of a distance measuring between axes for one revolution of a gear F''_{ir} (Fig. 11-10). These errors are measured by devices having corresponding names. One of the errors

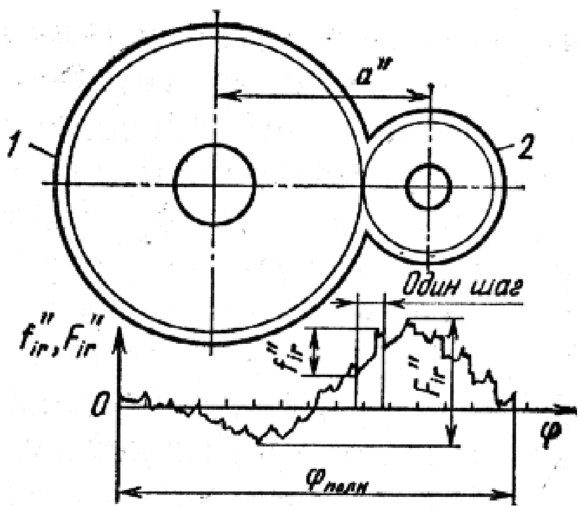


Fig. 11-10. A definition of fluctuation distance measuring between axes for one revolution of a gear F''_{ir} and its local kinematics errors f''_{ir}.

is supervised usually, others will correspond approximately to the same degree of accuracy. The choice of a kind of a checked error depends on a required degree of kinematics accuracy, kind of the mechanism and presence of the appropriate device. Usually radial palpation of a gear ring F_{rr} or fluctuation of a distance measuring between axes for one revolution of a gear F''_{ir} is supervised because of simplicity of the control.

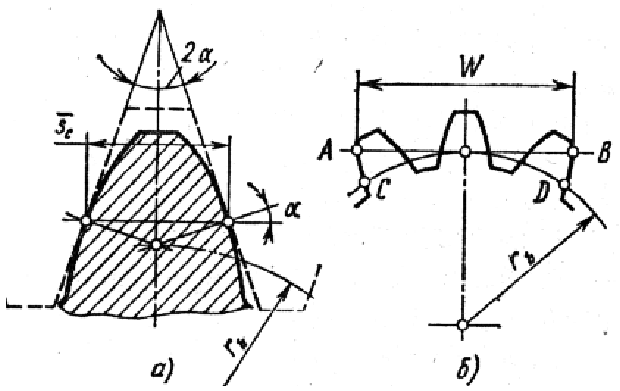


Fig. 11-11. A definition of parameters: constant hord S_c and common normal length W.

The **smoothness of work** of transfer is determined with parameters, which errors cyclically are shown for a revolution of a gear and also make a part of kinematics error. Analytically the kinematics error can be submitted as a spectrum of harmonic components, amplitude and frequency of which depend on the character of making errors. For example, the deviation of a step of gearing causes fluctuation of kinematics error with teeth frequency equal the frequency of an entrance of gears teeth. The cyclic character of errors and possibilities of harmonic analysis enable us to determine these errors by the spectrum of kinematics error.

The cyclic error of transfer f_{zko} and gears f_{zkr} is the double amplitude of harmonic making of kinematics error accordingly transfers or gears. Tolerances of a cyclic error of transfer and gears are defined by the formula:

$f_{zko}= f_{zk}=(k_c^{-0.6} + 0.13)F_r$, (11.1)

where k_c is a frequency of cycles; F_r is a radial palpation tolerance of a gear ring of the same degree, as f_{zk}.

The analysis of formula (11.1) shows, that the tolerances f_{zko} and f_{zk} decrease with increase of frequency k_c.

Helical transfers with large factor of axial overlapping ε_β have smaller amplitude in comparison with spur transfers, therefore the tolerance f_{zzo} is less as ε_β increases. Local kinematics errors of transfer f'_{ior} and gear f'_{ir} are the greatest difference between the local next limit data of kinematics error of transfer or gear for a complete cycle of rotation of transfer gears. Usually fluctuation of a distance measuring between the axes on one teeth of a gear f''_{ir} is supervised because of simplicity of the control (Fig. 11-10).

The deviation of a step f_{ptr} is kinematics error of a gear at its turn on one nominal angular step. The deviation of a step of gearing f_{pbr} is the difference between an actual P_a and nominal P_n steps of gearing. The actual step of gearing P_a is equal to the least distance between parallel planes, tangents to two same active lateral surfaces next teeth of gear. $|f_{pb}| = |f_{pt}|\cos\alpha = 0.94f_{pt}$. Instead of a deviation of a step

f_{ptr} it is possible to apply the difference of any steps f_{vptr}, and f_{vptr} =1.6|f_{pt}|. The error of a tooth profile f_{fr} is supervised for accurate gears.

The **contact of transfer teeth** should be whenever possible greatest. When the contact on lateral surfaces is incomplete, the bearing area decreases, contact loadings are increased and are allocated non-uniformly. It results in the intensive wear and the teeth may break easily.

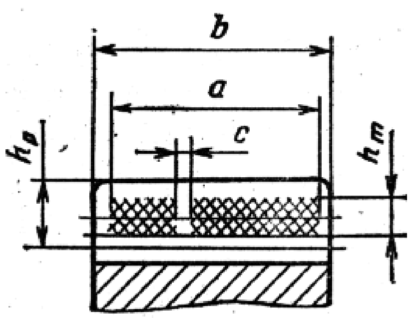


Fig. 11-12. A definition of total contact stain.

Usually a total stain of contact (Fig. 11-12) is supervised on traces of a paint in the assembled transfer after rotation under loading. The stain of contact is defined by relative sizes (percents): on length of a tooth by the ratio of distance **a** between extreme points of traces adjoin with a subtraction of breaks **c**, exceeding the module **m** in mm, to length of a tooth **b**, i.e. [(a-c)/b]100% ; on height of a tooth - by the ratio average (on length of a tooth) height of traces adjoin h_m to height of active lateral surfaces tooth h_p, i.e. (h_m/ h_p)100% .

The **total error of a contact line F_{kr}** is a distance on normal between two nearest nominal potential contact lines conditionally imposed on a surface of gearing, between which the actual potential contact line on an active lateral surface of a tooth is placed. The contact line is a line of crossing of a surface of a tooth by a surface of gearing. The deviation F_{Pxnr} influences on longitudinal, and error F_{kr} - on high-altitude contact of teeth.

The **error of a direction of a tooth $F_{\beta r}$** is a distance on normal between the nearest nominal pitch lines of a tooth in face section, between which actual pitch line of a tooth passes appropriate to working width of a ring. The actual pitch line of a tooth is a line crossing the actual lateral surface of a tooth of a gear by pitch cylinder, which axis coincides with the working axis.

The **deviation from parallelism of axes f_{xy}** is a deviation from parallelism of projections of working axes of gears of transfer to a plane, in which one of axes and a point of the second axis in an average plane of transfer lays. The **skew of axes f_{yr}** is a deviation from parallelism of projections of working axes of gears on the plane parallel by one of axes and perpendicular planes, in which this axis, and point of crossing of the second axis with an average plane of transfer lays. The **accuracy of installation of transfer** is determined also by deviations between-axes distance **f_{ar}**. For this error the limiting deviations are established: upper +f_a and lower -f_a.

When the total stain of contact corresponds to the requirements of the standard, the control of other parameters of gears contact is not required. It is supposed to determine a stain of contact with the help of a measuring gear.

Kinds of interfaces of gears teeth in transfer. The gear transfers should have a lateral clearance j_n (between non-working teeth profiles of the mating gears) for elimination of possible jamming at heating, maintenance of conditions of greasing course, compensation of errors of manufacturing, assembly and restriction of a dead course at reverse (Fig. 11-13). Such transfer is one-profile (contact only on one working profile). The lateral clearance is determined in section, perpendicular to a teeth direction, in a plane, tangent to the basic cylinders.

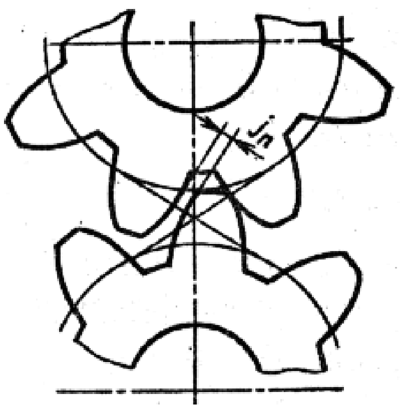


Fig. 11-13. A lateral clearance j_n

The system of tolerances on gear transfers establishes a guaranteed lateral backlash j_{nmin} , which is the least ordered lateral clearance not dependent on a degree of accuracy. For example, the most exact transfers of high-speed reducers of turbines are made with large lateral clearances for compensation of temperature deformations. Six kinds of interfaces determining various value for j_{nmin} , are established with reduction of a guaranteed clearance: A, B, C, D, E, H. These interfaces apply accordingly to degrees of accuracy on norms of smoothness of work: 3-12, 3-11, 3-9, 3-8, 3-7, 3-7. For interface H $j_{n\,min}$=0. The interface of a kind B guarantees the minimal lateral clearance, at which is excluded an opportunity of jamming of steel transfer from heating at a difference of temperatures of gears and case 25 ^oC.

Six classes of deviations of interaxial distance are established, designated in decreasing order of accuracy by the Roman figures from I up to VI. The guaranteed lateral clearance in each interface is provided at observance of the stipulated classes of deviations of interaxial distance (for interfaces H and E - II classes, for interfaces D, C, B, and A - classes III, IV, V and VI accordingly). The conformity of kinds of interfaces and specified classes is supposed to be changed.

The tolerance is established for a lateral clearance determined by a difference between the greatest and least clearances. Eight kinds of the tolerances T_{jn} are established for a lateral clearance: x, y, z, a, b, c, d, h. The conformity of kinds of interfaces and kinds of the tolerances is authorized to be changed, using thus and kinds of the tolerances z, y, x. The lateral clearance $j_{n\ min}$, necessary for compensation of temperature deformations, is defined by the formula

$$j_{n\,min} = V + \alpha_W(\alpha_1 \Delta t_1 - \alpha_2 \Delta t_2)2\sin\alpha, \qquad (11.2)$$

where V- thickness of a layer of greasing between gears teeth; α_W - interaxial distance; α_1 and α_2 - temperature factors of linear expansion of a material of gears and case; Δt_1 and Δt_2 - deviation of temperatures of gears and case from 20 ^oC; α- angle of profile of an initial contour.

The lateral clearance ensuring normal conditions of greasing is roughly accepted in the limits from 0.01mm (for low-speed transfers) up to 0.03 mm (for high-speed transfers).

The lateral clearance is provided by a radial displacement of the initial contour of a cutting tool from its basic site in a body of a gear (Fig. 11-14). The basic site of an initial contour is a site of the tool, at which the basic thickness of a tooth corresponds to dense of two-profile gearing.

The least additional displacement of an initial contour is nominated depending on a degree of accuracy on norms of smoothness and kind of interface and is designated $-E_{Hs}$ for gears with external teeth and $+E_{Hs}$ for gears with internal teeth. The tolerance T_H for an additional displacement of an initial contour is established depending on the tolerance on radial palpation F_r and kind of interface.

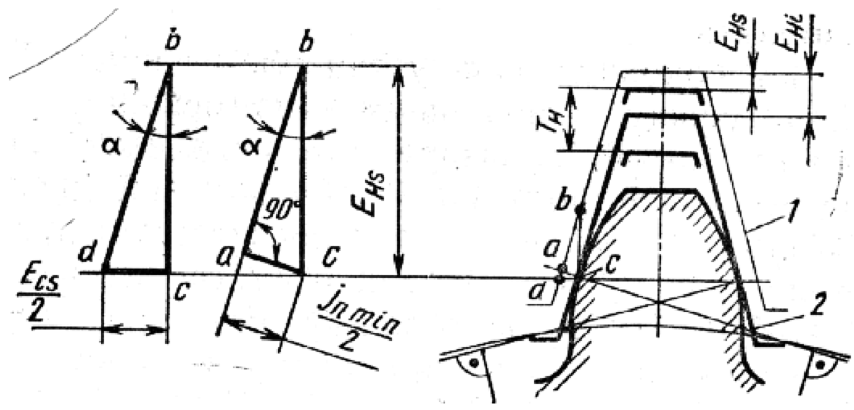


Fig. 11-14. An initial contour: 1- basic site; 2- actual site.

The least displacement of an initial contour at a clearance K_j , compensating errors of manufacturing, both installation of gears and reducing lateral clearance:

$E_{Hs} = 0.25(j_{n\,min} + K_j)/ \sin\alpha.$ (11.3)

Guaranteed lateral clearance is nominated also by least deviation of average length of common normal $-E_{Wms}$ ($+E_{Wmi}$) or least deviation of thickness of a tooth $-E_{cs}$, or limiting deviation of measuring interaxial distance $E_{a''s}$ ($E_{a''i}$).

The tolerances on average length common normal T_{Wm}, on thickness of a tooth on constant chord T_c, and limiting deviations measuring interaxial distance are accordingly established: upper $+E_{a''s}$ and lower $- E_{a''i}$.

Designation of accuracy of wheels and transfers.

Accuracy of manufacturing of gears and transfers are set by a degree of accuracy on the appropriate norms of accuracy, and requirement to a lateral clearance - by kind of interface on norms of a lateral clearance. An example of a designation: 8-B - cylindrical transfer with a degree of accuracy 8 on all three norms, with a kind of interface B and kind of the tolerance on a lateral clearance b. 8-7-6-Bc - degree of accuracy 8 on norms of kinematics accuracy, degree of accuracy 7 on norms of smoothness of work, degree of accuracy 6 on norms of teeth contact, kind of interface B, kind of the tolerance on a lateral clearance c and conformity between a kind of interface and class of deviations of interaxial distance.

An example of a designation of transfer with a degree of accuracy **7** on all norms, with a kind of interface of gears **C**, with a kind of the tolerance on a lateral tolerance **a** and more rough class of deviations of interaxial distance **V** and reduced guaranteed lateral clearance $j_{n\,min}$ =128 microns is: 7-Ca/V -128.

Degree of accuracy of gears and transfers are established depending on the requirements to kinematics accuracy, smoothness, transmitted capacity and circumferential speed of gears. At circumferential speed 10-15 mps degrees of accuracy 6-7 are applied; at speeds 20-40 mps - degrees of accuracy 4-5. For automobiles degrees of accuracy 7-6-6-C, 8-7-7-C are applied; for divide mechanisms 4-5-5-D, etc.

The complexes of the control are established for the control of gears. Each norm of accuracy can be controlled by several methods and devices, which choice depends on a degree of accuracy and purpose of transfer.

Chapter 12. Drawing requirements

The assembly drawings (Fig. 12-1) are necessary for understanding of an arrangement of details in the mechanism or unit, degree of mobility of mating parts, that is necessary for understanding of mechanism work. Fits of mating parts and their base sizes are specified on the assembly drawings. The technical requirements to assembly of the mechanism are specified in a right bottom corner above a stamp of the assembly drawing. Example: 1. A deviation from coaxiality of surface axes B and D no more than 0.05 mm; 2. A cover 12 to collect on a paint.

Technical characteristics can be specified on the assembly drawing, for example: The maximal frequency of rotation of the target shaft is 1000 R.P.M.

The nominal sizes and accuracy of manufacturing of fastened elements are specified on the drawing, for example, a target end of the shaft or distance between holes for fastening of the mechanism.

The design drawing is used for manufacturing a detail, therefore all necessary kinds and sections, all sizes and extreme allowable deviations of these sizes are specified on it. **The deviations** are defined in tables according with a base size and a tolerance zone. The tolerance zone of the size is nominated according to a fit on the assembly drawing. The deviations of the rough sizes (11 - 14 grade of tolerance) are supposed to be not specified, but in a right bottom corner of the drawing above a stamp the accuracy and position of a tolerance zone is shown.

Example: 1. H12, h12, $\pm$ IT12/2. It means, that all sizes concerning internal dimensions (for example, holes) are made on H12, if the deviations of the size on the drawing are not specified. All outside dimensions (for example, a diameter of shaft) are made on h12. If the size does not concern neither to outside nor to internal, it is made with a symmetric arrangement of a tolerance zone. The tolerance of corresponding grade of tolerance (in our example 12 grade of tolerance) is divided half-and-half and this is maximum (+) and minimum (-) limit deviations.

A roughness of all surfaces is shown on the worker drawing. For a plenty of surfaces having an identical roughness, the roughness is shown in a right top corner of the drawing. This mark shows, that if a roughness on a surface is not shown, it is equal to the roughness specified in a right top corner. Usually so the roughness is shown for the roughly processed surfaces (Ra ≥ 10 μm or Rz = 40 ... 60 μm). The mark of a roughness in brackets in a corner of the drawing reminds, that there are also surfaces with other roughness, which is shown on the drawing. The conformity between accuracy of manufacturing and possible allowable roughness is observed:

Ra = 6.3 ... 10 μm for 11 - 14 grades of tolerance, but can be Rz = 20 - 60 μm; Ra = 2.5 μm for 8-10 grades of tolerance; Ra = 1.25 μm for 7 grade of tolerance; Ra = 0.8 μm for 6 grade of tolerance. The roughness and the accuracy can not

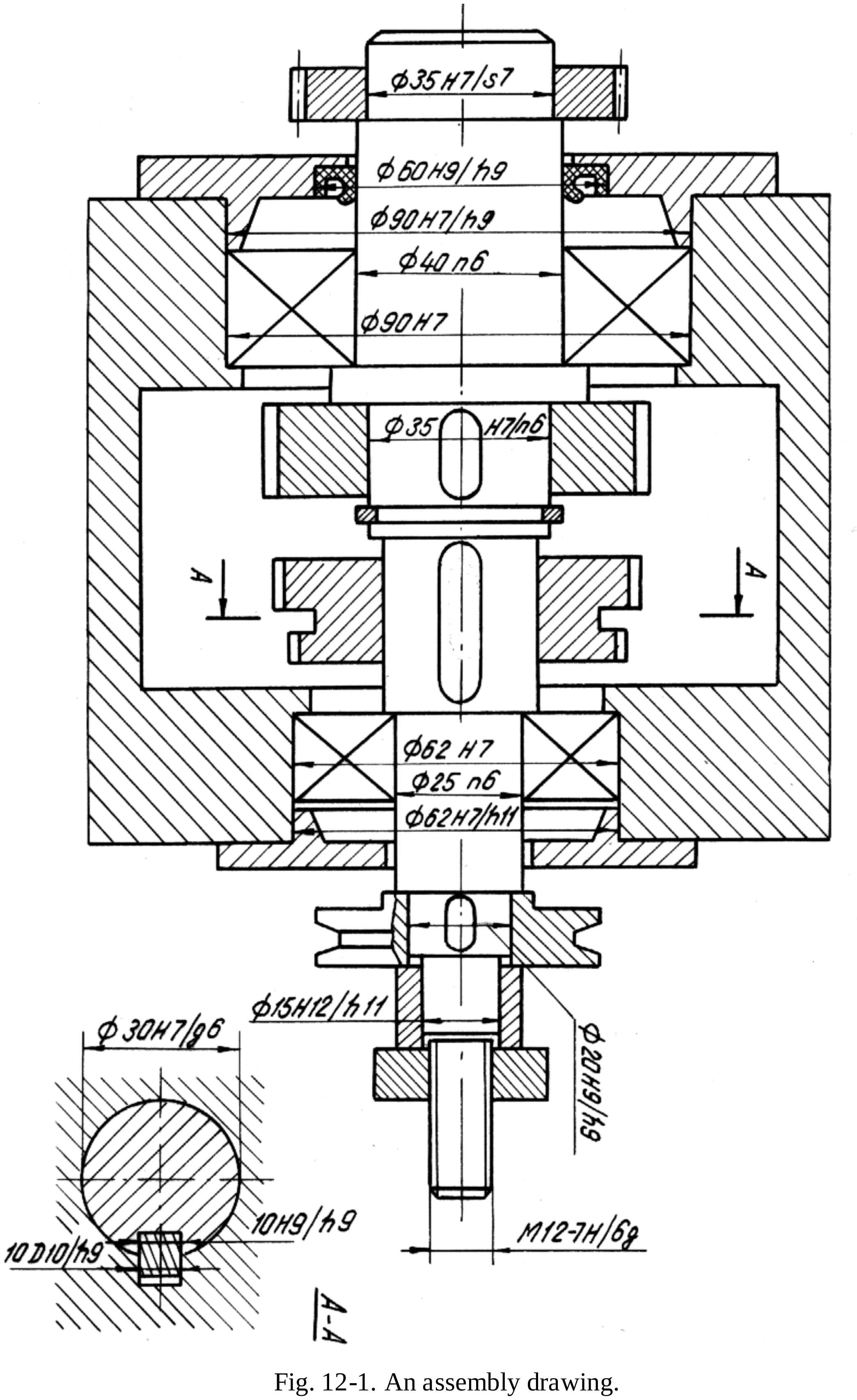


Fig. 12-1. An assembly drawing.

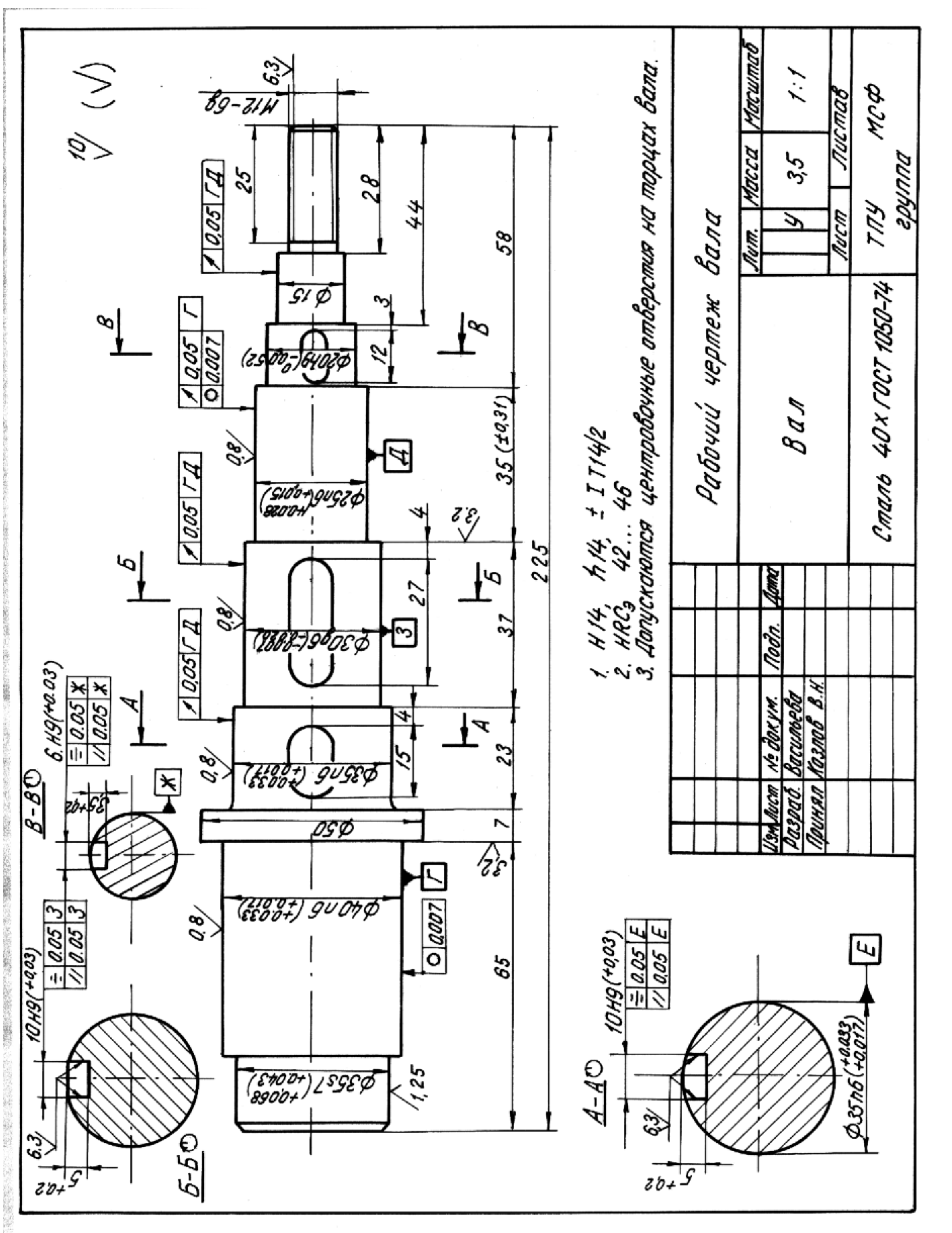


Fig. 12-2. A design drawing.

correspond. For example: the surface of the shaft is interfaced with rubber cuff. The high accuracy is not required here (9 grade of tolerance is required), but the roughness should be small (Ra ≤ 0.32 μm, polishing) for reliable condensation and increase of service life.

The hardness of surfaces of a detail is shown above a stamp. If the shaft has exact surfaces (6 - 7 grades of tolerance), one is usual tempered and has hardness HRC = 42 - 46. Details of other types it is usual not tempered and have hardness HB = 180 - 260 depending on the mark of a material.

Other technical requirements also can be specified above a stamp. For example: 1. To mark «M12» on a surface A; 2. Radiuses are 2.5 mm, etc. **Tolerances of the form and disposition of surfaces** are shown by conventional signs on a contour of a detail, but can be specified in the technical requirements. For example: 1. The deviation from coaxiality of surfaces B and D no more than 0.05 mm.

Chapter 13. Measurements Errors and Measurements Planning Methods

13.1. The Distribution Laws of Measurements Errors

Because of the fundamental role of measurements it is necessary to consider in more detail what a measurement actually is. A true understanding of the nature of measurement would prevent many errors of interpretation which may be perpetrated at the conclusion level. The sources of perturbation of measurements cover such a wide range that it is possible to do no more than list a few typical headings. It is common to divide such perturbations into categories such as random, systematic, personal, etc. However, these terms are frequently too precise for the average experimental situation and their usefulness is consequently limited. They are commonly defined as follows:

Random (or accidental) error is said to be shown when repeated measurements of the same quantity give rise to differing values.

Systematic error refers to a perturbation which influences all measurements of a particular quantity equally. If the magnitude of this error does not change with time, it appears as a constant error in all of the measurements and also in the median and arithmetic mean values. If it changes in magnitude, it introduces some skew (asymmetry) in the observed histogram.

Examples of systematic errors are those caused by:

1. Incorrect (or an unjustifiably assumed) calibration of an instrument, friction or wear in moving parts of an instrument, failure to correct for the “zero” reading, etc.
2. Constructional faults in the apparatus, e.g., unaligned parts, screw errors, etc.
3. Inadequate regard to constancy of experimental conditions and imperfect measurement techniques, e.g. changes in dimensions owing to thermal

expansion, one-sided illumination of a scale, etc.

4. Failure to make necessary corrections, e.g., for the effect of atmospheric pressure.
5. Bias by the observer, e.g., more or less constant parallax, desire for the “right” result, etc.

Precision in a mean value is proportional to the reciprocal of the statistical error and is “high” if the statistical error is small; *accuracy* is “high” if the net systematic error is small. Precision and accuracy are not interchangeable terms. Statistical methods give specifically a quantitative measure of precision, not of accuracy.

However, these terms must be used with caution since a set of readings will show truly random error only if there are a large number of small perturbing influences. We shall use the terms systematic and random to indicate only clear-cut cases.

To detect a random error of measurement it is necessary to repeat the measurements many times. If measurements of the same parameter of an object give appreciably distinct results, then the random error may play an essential role, and it is important to find among the measurements the nearest to true value of the parameter.

For example, we have measured one diameter **n** times and have **n** values. All of them were done with the help of the same method and with the same degree of carefulness. Such measurements are called equal in accuracy. A set of observations or measurements is said to be an “ideal trial set” if they are all completely independent of each other and are carried out under strictly identical conditions. In practice, such measurements are never quite achieved, but, with meticulous care, actual measurements often come satisfactorily close to the ideal.

Methods of picturing ranges of numerical values can take many forms, but the most profitable device is the *"histogram."*

This diagram is drawn by dividing the original set of observations x_i into intervals of predetermined magnitude and counting the numbers of observations found within each interval. If one plots, on a suitable scale, this *frequency y* or *f* versus the readings themselves, a block diagram is obtained which is the required picture of the distribution of the readings along the scale of values. It actually contains no more information than the original set of readings but has, as required, the enormous advantage of visual presentation of the nature of the distribution of the readings.

Such a histogram is shown in Fig. 13-1. If the number of readings is very high, so that a fine subdivision of the scale of values can be made, the histogram approaches a continuous curve, and this is called a *distribution curve*. The *range* of the distribution curve is equal to the difference between the largest and the smallest measurements.

Clearly a histogram or a distribution curve contains the information about the spread of the experimenter's readings which he wishes to communicate to other people, and does so in a clearer and more informative manner than does the original table of values. The histogram is an excellent way of presenting experimental results and is commonly used in reports on work which shows a high level of statis-

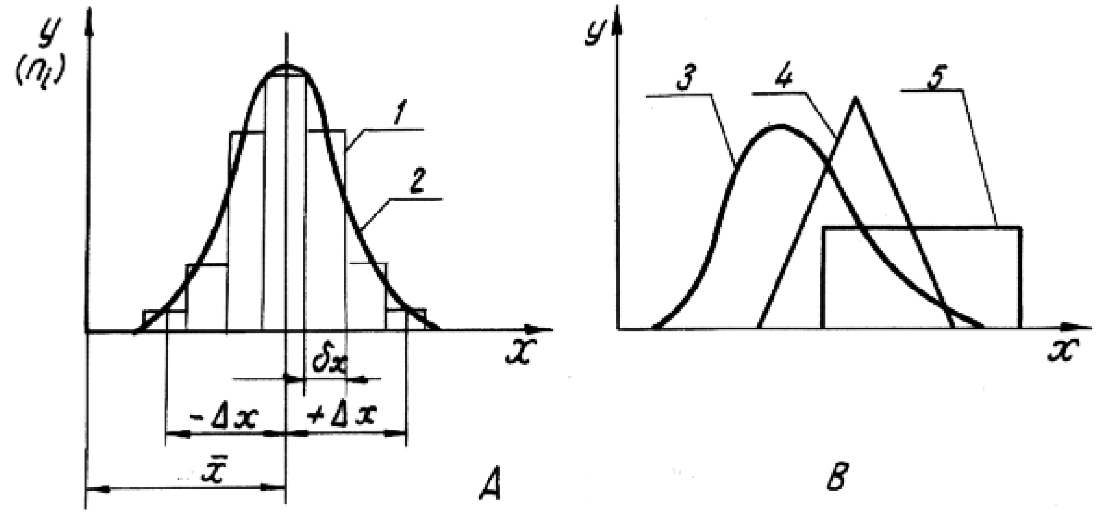


Fig. 13-1. The histogram 1 and the curve 2 of density probability distribution *y* of occurrence of parameter *x* (A); curves of density probability distribution *y*: normal distribution (Gauss's law) (2); Markskvell's law (3); Simpson's law (triangle) (4); law of equal probability (5).

tical fluctuation. It has the further advantage of presenting the observations themselves, free from manipulation on the part of the experimenter, and thus permits the reader to form his own judgement regarding the value of the work.

However, such a full-scale representation of the results may be undesirable, perhaps on grounds of difficulty of presentation, or more significantly, the results may be required for further work. In either case, one wants to find numbers which will represent the distribution of the values so as to answer the questions, "Which single number shall I take as the answer, and how reliable is it?" It is necessary, therefore, to define quantities which will serve as a "best" value and an "uncertainty."

The uncertainty is obviously associated in some way with the spread of the results, and the best value with any tendency of the results to cluster in the middle of the distribution.

In treating random trial data, it is often possible to invoke a mathematical model of the variations among the trials. If this model is not too complex, it enables the experimenter to make quickly very significant computations about such properties of the set as (1) the best value and its reliability, (2) the frequency with which a particular result or measurement may be expected to occur when a certain number of trials are made, (3) the number of trials that need be made for a specified precision in the best value, etc.

The branch of mathematics that applies mathematical models for random trial data is called *probability theory*.

The simplest mathematical model is the *binomial distribution*. The two mathematical models of outstanding importance in an experimental science are the *normal* (or *Gauss*) *distribution* and the *Poisson distribution*.

For the most probable value of a measured variable it is common to use its *mean* (or *arithmetic average*) *value m* ($\bar{x}$), calculated on the basis of measured values x_1, x_2, x_3, …, x_i

$$m \equiv \bar{x} = \frac{1}{n}\sum_{i=1}^{n} x_i \,. \qquad (13.1)$$

The mean does have the characteristics of a "best" value and is commonly quoted as such.

However, if the distribution is not symmetrical, the mean value is not the most commonly found value, and the value at which the peak of the distribution occurs is called the *mode*. For many purposes it is the most frequently found value which is the significant feature and so, for many purposes, the mode is quoted as the single number which best characterizes the distribution. Note that the mode and the mean can have differing values and this shows again the need for caution, since asymmetry of the distribution curve would probably be revealed only by actual plotting. In such a case, careful thought should be given to the question of which number best typifies the set of values.

Yet another quantity is commonly defined at the center of the set of readings so that as many readings fall below it in value as above it. This is called the *median* and a line drawn vertically from the median on the distribution curve will divide it into two equal areas. For a symmetric distribution it coincides with the mean and the mode. These three quantities are illustrated in Fig. 13-2.

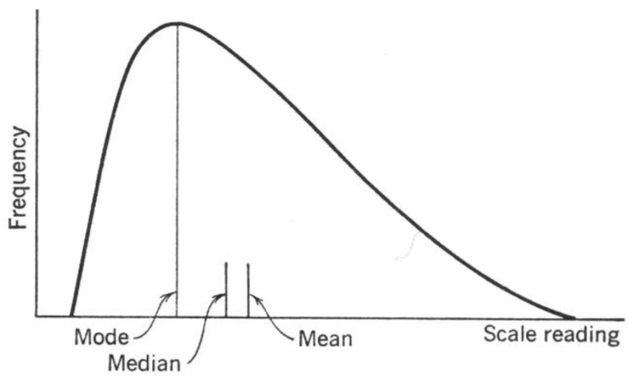


Fig. 13-2. The constants of a skew distribution.

In any case, for a symmetric distribution these three values coincide and so it is now necessary to define a quantity connected with the actual width of the curve.

A *deviation* or *statistical fluctuation* is the difference between a single measured value and the “best” value of a set of measurements whose variation is apparently random. The “best” value is defined for this purpose as the arithmetic mean of all the actual trial measurements. If a set of n readings x_1, x_2, etc. have a mean $\bar{x}$, the *mean deviation* r_n is defined as

$$r_n = \frac{1}{n}\sum_{i=1}^{n} \left|\bar{x} - x_i\right|. \qquad (13.2)$$

However, a more commonly used term is the *variance* or *mean squared deviation* σ^2 and it is defined as

$$\sigma^2 = \frac{\sum_{i=1}^{n} (\bar{x} - x_i)^2}{n}. \tag{13.3}$$

The square root of the variance is called the *standard deviation* or *root-mean-square*, denoted by S, so that

$$S = \sqrt{\frac{\sum_{i=1}^{n} (\bar{x} - x_i)^2}{n}}. \tag{13.4}$$

If the number of observations is n > 30 we can use the next equation

$$S = \sqrt{\frac{\sum_{i=1}^{n} (\bar{x} - x_i)^2}{n-1}}. \tag{13.5}$$

If the number of observations is very large, the value S, which is subjected to random fluctuations, converges to some constant σ, which can be named as a *statistical limit* of S

$$\sigma = \lim_{n \to \infty} S. \tag{13.6}$$

Actually, not value σ, but its approached value S is always calculated. σ is also called the “*universe*” or “*parent*” standard deviation. We shall use latin letters, e.g., S for standard deviation, for quantities related to finite sets of actual observations, and greek letters, e.g., σ, when referring to the normal distribution itself, or a "universe" of readings.

The relative value of the universe standard deviation σ, expressed in percentage, is called the *factor of variance* or *fractional variance*

$$\omega = \frac{\sigma}{\bar{x}} \cdot 100\%. \tag{13.7}$$

With a limited number n of measurements the *fractional standard deviation* is defined as

$$\omega_n = \frac{S}{\bar{x}} \cdot 100\%. \tag{13.8}$$

By α we shall designate probability that the difference between the true value of x (or *arithmetic average* $\bar{x}$) and sample mean does not exceed Δx:

$$P(-\Delta x < x - \bar{x} < \Delta x) = \alpha \tag{13.9}$$

The probability α is called the *confidence level*, or a *reliability coefficient*. The interval of values from x-Δx to x+Δx is called a *confidence interval* or a *range of uncertainty* ($\pm\Delta x$).

The expression (13.9) shows, that with the probability equal to α the result of measurements lies inside the confidence interval. It is often to use the confidence interval Δx in shares of σ or S

$$\varepsilon = \Delta x/\sigma. \tag{13.10}$$

With a limited number n of measurements

$$\varepsilon_n = \Delta x/S. \tag{13.11}$$

The problem is thus a matter of reducing a set of observations to such a condensed form as will permit further work or calculation. The nature of this condensed form depends on the nature of the results. If the distribution curve is symmetric, the obvious quantities to quote are the central value (i.e., the mean) and some measure of the width of the curve, such as the standard deviation of the set. If the distribution curve is markedly asymmetric, it may be necessary, in order to provide an accurate impression of the results, to quote the mode and/or median, in addition to the mean, and, in extreme cases, there is no alternative to quoting the whole distribution curve.

Many theoretical distribution curves have been defined and their properties evaluated, but the one which has most significance in the theory of measurement is the Gaussian distribution.

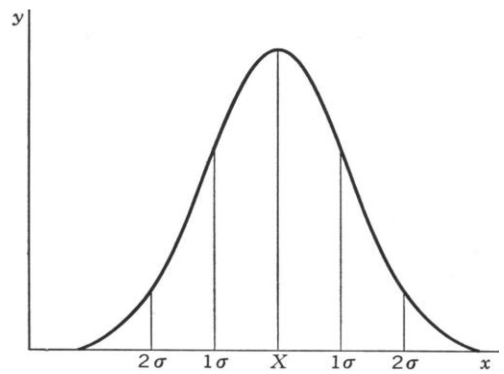


Fig. 13-3. The relationship of 1σ and 2σ limits to the Gaussian distribution.

The Gaussian error curve can be defined on the assumption that the total deviation of a measured quantity, x, from a central value, X, is the consequence of a large number of small fluctuations. If there are m such contributions to the total deviation, each of equal magnitude, a , and either positive or negative, the total set of observations may range from $X+ma$, if all fluctuations happen to be positive simultaneously, to $X-ma$ if the same happens in the negative direction. It can be shown, in such a random summation of positive and negative quantities (as in the "random walk”), that the most probable sum is zero, meaning that the most common values of x are in the vicinity of X. The distribution curve is thus peaked in the middle, is symmetric, and declines smoothly to zero at $x=X+ma$ and $x=X-ma$. If this concept is taken to the limiting case of an infinite number of infinitesimal contributions to the total deviation, the curve has the form shown in Fig. 13-3. Treating the curve solely from the mathematical point of view for the moment, its equation can be written

$$y = \frac{1}{\sigma\sqrt{2\pi}} \cdot e^{-\frac{(\Delta x)^2}{2\sigma^2}}, \tag{13.12}$$

where σ is a variance of measurements, e is the base of the natural logarithms e=2,71… .

The confidence level α for the Gaussian distribution is calculated by the Laplas function

$$\alpha = \frac{2}{\sqrt{2\pi}} \int_0^{\varepsilon} e^{\frac{\varepsilon^2}{2}} \cdot d\varepsilon . \qquad (13.13)$$

To characterize a random error it is necessary to set two numbers: value of the error (or confidence limits) and value of confidence level. The knowledge of confidence level allows to estimate a degree of reliability of the result.

The degree of reliability is defined by the character of made measurements: for example, requirements are more rigid to details of the plane engine than to the boat motor. Usually it is possible to limit the confidence level by 0.9 or 0.95.

Table 13-1

The confidence level α for the confidence interval Δx expressed in shares of variance of measurements σ ε=Δx/σ

ε	α	ε	α	ε	α
0	0	1.0	0.68	2.2	0.97200
0.05	0.04	1.1	0.73	2.3	0.978
0.1	0.08	1.2	0.77	2.4	0.984
0.15	0.12	1.3	0.8	2.5	0.988
0.2	0.16	1.4	0.84	2.6	0.990
0.3	0.24	1.5	0.87	2.7	0.993
0.4	0.31	1.6	0.89	2.8	0.995
0.5	0.38	1.7	0.91	3.0	0.997
0.6	0.45	1.8	0.93	3.3	0.999
0.7	0.51	1.9	0.94	3.6	0.9997
0.8	0.57	2.0	0.95	3.9	0.9999
0.9	0.63	2.1	0.96	4.0	0.99993

Examples of usage of Table 13-1:

a) The arithmetic average value (mean), calculated on the basis of some measured values, is $\bar{x}$=1.27 mm. The standard deviation is S=0.032 mm. What is the probability that the result of separate measurement will be into limits:

1.26 mm < x_i < 1.28 mm?

Confidence limit (±Δx) is ±0.01 mm (1.26 mm < x_i < 1.28 mm). We define ε = Δx/ S = 0.01:0.032=0.31.

From Table 13-1 we find, that for ε≈0.3 confidence level α=0.24, or 24 %, i.e. approximately quarter of all measurements will be into confidence interval ±0.01 mm.

b) For the previous example we shall define, what the confidence level will be for limits 1.2 mm < x_i < 1.34 mm.

We determine confidence limit (from the mean $\bar{x}$ =1,27 mm) Δx =±0,07 mm. Then we define $\varepsilon = \Delta x / S$ = 0.07:0.032≈2.2.

From Table 13-1 we find α for ε=2.2, it is equal to 0.972. i.e. the results approximately 97 % of all measurements will be into limits 1.2 mm < x_i < 1.34 mm.

c) What confidence interval should be chosen for the same measurements, that the confidence interval to cover approximately 98% of measurements?

From Table 13-1 we find, that to α=0.98 corresponds ε≈2.4, hence $\Delta x = S \cdot \varepsilon$=0.032·2.4≈0.077. i.e. confidence interval ±0.077 mm corresponds to confidence level α=98% (98% confidence interval are ±0.077 mm) from $\bar{x}$=1.27 mm, so for 1.193 мм<x_i<1.347 mm.

From Table 13-1 it follows, that for the universe standard deviation σ the corresponding confidence level is 0.68, for the double universe standard deviation (2σ) - confidence level is 0.95, for the trebled (3σ) – 0.997. For practical measurements application of the trebled universe standard deviation 3σ is more often in connection with rather high confidence level 99.7 %.

If the law of distribution of errors is unknown or it obviously differs from the normal, we can use an inequality of Tchebyshev.

If the variable N is the sum (or difference) of two and more variables, the universe standard deviation of this sum (or difference) will be equal to a root square of the sum of variances

$$\sigma_N = \left| \sqrt{\sigma_1^2 + \sigma_2^2 + \ldots + \sigma_i^2} \right|. \quad (13.14)$$

From this property two important conclusions follow:

1. If one of two errors is twice less another, the total error will increase at the expense of this smaller error all by 10 % . Therefore it is always necessary first of all to reduce an error having the greatest value.
2. The standard deviation S_n is equal to the standard deviation of separate result, divided on a root square from the number of measurements

$$S_y = S_n = \frac{S}{\sqrt{n}} . \quad (13.15)$$

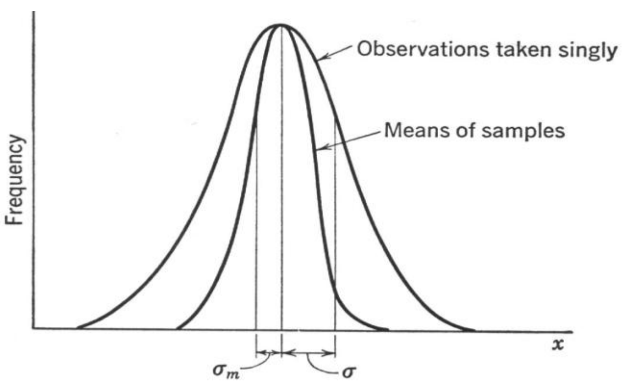


Fig. 13-4. Distribution curve of single observations and sample means.

By this rule: if we need to increase the accuracy of measurements in 2 times, we should do instead of one - four measurements; to increase the accuracy in 3 times it is necessary to increase the number of measurements in 9 times etc.

Suppose that we obtain several groups of measurements, calculate the mean of each group of measurements and plot the distribution curve of these means. A derivation based on the statistics of sampling shows that new distribution curve of means will also be Gaussian (even if, in fact, the distribution of single observations was not Gaussian) and will be centered at the same point X as the first curve. However, its most striking feature is that, as illustrated in Fig. 13-4, it is narrower than the distribution curve of the readings taken singly and it can be shown that the standard deviation of this set of means, which we shall denote σ_m, is given by

$$\sigma_m = \frac{\sigma}{\sqrt{n}} \ . \tag{13.16}$$

S_m or σ_m is often called the *mean-root-square error*, or *standard error in the mean*, or simply *standard error*.

If the number of measurements is bounded, it is necessary to take into account that the new confidence interval will change as

$$\Delta x = \frac{t_{\alpha n} \cdot S_n}{\sqrt{n}} \ . \tag{13.17}$$

Whence

$$t_{\alpha n} = \frac{\Delta x \cdot \sqrt{n}}{S_n} . \tag{13.18}$$

The value $t_{\alpha n}$, named as the Student factor, is similar to ε, but only in the case, when the number of measurements is bounded.

Table 13-2

Student factor $t_{\alpha n}$ depending on number of measurements n and confidence level α

	α										
n	0.1	0.2	0.5	0.6	0.7	0.8	0.9	0.95	0.98	0.99	0.999
2	0.16	0.33	1.00	1.38	2.0	3.1	6.3	12.7	31.8	63.7	636.6
3	0.14	0.29	0.82	1.06	1.3	1.9	2.9	4.3	7.0	9.9	31.6
4	0.14	0.28	0.77	0.98	1.3	1.6	2.4	3.2	4.5	5.8	12.9
5	0.13	0.27	0.74	0.94	1.2	1.5	2.1	2.8	3.7	4.6	8.6
6	0.13	0.27	0.73	0.92	1.2	1.5	2.0	2.6	3.4	4.0	6.9
7	0.13	0.27	0.72	0.90	1.1	1.4	1.9	2.4	3.1	3.7	6.0
8	0.13	0.26	0.71	0.90	1.1	1.4	1.9	2.4	3.0	3.5	5.4
9	0.13	0.26	0.71	0.90	1.1	1.4	1.9	2.3	2.9	3.4	5.0
10	0.13	0.26	0.70	0.88	1.1	1.4	1.8	2.3	2.8	3.3	4.8
...											
20	0.13	0.26	0.68	0.85	1.0	1.3	1.7	2.1	2.5	2.9	3.9
...											
60	0.13	0.25	0.68	0.85	1.0	1.3	1.7	2.0	2.4	2.7	3.5
...											
120	0.13	0.25	0.68	0.85	1.0	1.3	1.7	2.0	2.4	2.6	3.4
...											
∞	0.13	0.25	0.67	0.84	1.0	1.3	1.6	2.0	2.3	2.6	3.3

Examples of usage of Table 13-2:

a) We have received values 24.8; 24.6; 25.0; 24.8 mm (four measurements of a shaft diameter). It is required to determine confidence level that the mean differs from the true value no more than on 0.1 mm.

1. We find the mean

$$\bar{x} = \frac{\sum_{i=1}^{n} x_i}{n} = \frac{\sum_{i=1}^{4} x_i}{4} = \frac{24{,}8 + 24{,}6 + 25{,}0 + 24{,}8}{4} = 24.8$$,

i.e. diameter in the mean is d_m = 24.8 mm.

2. We calculate a standard deviation with the help of formula (13.4):

$$S_n = \sqrt{\frac{\sum_{1}^{n} (\bar{x} - x_i)^2}{n}} = \sqrt{\frac{\sum_{1}^{4} (\bar{x} - x_i)^2}{4}} =$$

$$= \sqrt{\frac{(24.8 - 24.8)^2 + (24.8 - 24.6)^2 + (24.8 - 25)^2 + (24.8 - 24.8)^2}{4}} =$$

$$= \sqrt{\frac{0 + 0.04 + 0.04 + 0}{4}} = \sqrt{0.02} = 0.14.$$

3. When the diameter in the mean is d_m = 24.8 mm, the confidence interval is Δx = 0.1 mm, i.e. $24{,}7 < d_i < 24{,}9$. We calculate Student's factor $t_{\alpha n}$ by the formula (13.18)

$$t_{\alpha n} = \frac{\Delta x \cdot \sqrt{n}}{S_n} = \frac{0.1 \cdot \sqrt{4}}{0.14} = 1.43.$$

4. We find from Table 13.2 for n=4 and $t_{\alpha n} = t_{\alpha;4} = 1.43 \approx 1.4$ that the confidence level $\alpha \approx 0.7$, or 70 %.

Student's factor in this case is $t_{0.7;4} \approx 1.4$, i.e. the probability that the diameter in the mean d_m = 24.8 mm differs from the true no more than 0.1 mm is approximately 70 %.

13.2. Methods of Measurements Planning

Whatever the nature of the experiment, it will be constructed out of measurements, and we must consider this first. The conduct of the experiment will be determined very largely by the precision which is required. The instruments used to measure individual quantities, and the whole method of measurement will depend on it. This quantity should be chosen realistically, since too optimistic a value will very quickly lead to too great complexity. It is necessary to remember the important distinction between readings whose precision is limited by statistical fluctuation and those whose precision is limited by the measuring scale. If we knew a tolerance of the measuring parameter, then a scale division must be 5…10 times less the tolerance and uncertainty of measurements will depend on it.

If the uncertainty of the reading proves to be statistical in nature, obtain an estimate of the standard deviation using, say, 10 readings. The precision can be improved by taking more readings. The standard deviation of the mean involves $\sqrt{n}$. Consequently, if 10 readings suggest that the precision must be improved by a factor of 10 (the fluctuations may be 10 per cent, when the experimenter wants 1 per cent), the number of readings must be increased by a factor of 100. This is an undesirable method of improving precision and some improved measurement procedures must be used.

Frequently the statistician makes previously some measurements, obviously less than it is required. On the basis of the results he solves the problem of necessary number of measurements. He sets necessary confidence level α and confidence interval Δx. The least number of tests n is defined by the formula

$$n = \left(\frac{S_m \cdot t_{\alpha m}}{\Delta x}\right)^2 \cdot \left(1 + \frac{1}{2m}\right), \qquad (13.19)$$

where S_m is the standard deviation; $t_{\alpha m}$ is the Student factor, determined from the table 13.2 by the known number of preliminary tests m and required confidence level α; Δx is the confidence interval; m is the number of preliminary tests, obviously smaller, than required.

For example, we have made 4 measurements of the shaft diameter and calculated the diameter in the mean d_m =24.8 mm and the standard deviation $S_{m=4}$=0.14 mm (see previous example). It is required to calculate the least number of measurements n, for which the deviations from the diameter in the mean d_m would not exceed ±0.1 mm with probability not less than 98 %.

1. We determine the Student factor from the table 13.2 at m=4 and α=0,98: $t_{0,98;4}$=4.5.
2. We calculate the least number of measurements n

$$n = \left(\frac{0.16 \cdot 4.5}{0.1}\right)^2 \left(1 + \frac{1}{2 \cdot 4}\right) = 58.32 \approx 58,$$

i.e. 58 measurements are necessary.

If in preliminary tests S_m =0.05 mm (i.e. less 3 times)

$$n = \left(\frac{0.05 \cdot 4.5}{0.1}\right)^2 \left(1 + \frac{1}{2 \cdot 4}\right) = 5.69 \approx 6,$$

i.e. only 6 measurements would be required.

In order to extract all the information from the observations it is necessary to plot not only the measured values but their confidence intervals Δx_i and Δy_i with their confidence levels α_x and α_y. The confidence intervals can be put on the diagram as symmetric segments of a direct line which pass through points $(\bar{x}_i; \bar{y}_i)$ parallel to axes of ordinates and abscissa.

If the experimental points are put on a coordinate plane, and it is necessary to construct a curve by these points, we choose a type of a curve and equation, describing it. If y is directly proportional to x, it is a direct line specified by the equation

$$y = ax + b\ . \tag{13.20}$$

Many physical processes can be described as

$$y = ax^m. \tag{13.21}$$

This equation can be transformed to the linear form by taking logs to the base e or *10*

$$\log_e y = \log_e a + m \times \log_e x. \tag{13.22}$$

Other more complicated functions are commonly found but, in all cases, some kind of linear analysis can be achieved, provided one is prepared to accept compound variables.

The kind of interpolation function should be chosen by some physical reasons, and after that the best interpolation function is found by the appropriate choice of the parameters. A commonly used interpolation function is described as

$$y = a + b_1 x_1 + b_2 x_2^2 + \ldots + b_n x_n^n\ . \tag{13.23}$$

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