Introduction

A. CALCULATION DATA (FORCES, VELOCITIES AND POWER REQUIREMENTS DURING METAL CUTTING)

The metal-cutting process is a result of two relative movements between the cutting tool and the material which has to be machined. The *cutting movement*, i.e. the relative movement between cutting edge and workpiece material, results in an amount of metal corresponding to the depth of cut being separated from the workpiece material in the form of chips; the *feed movement* brings new material in front of the cutting edge after a particular cut has been completed.

In some cases, e.g. planing or slotting, the operation is interrupted at the end of each cutting stroke and fresh material is brought in front of the cutting edge before the start of the succeeding one. In other processes, such as turning, drilling, cylindrical grinding and milling, the cutting and feed movements occur simultaneously and without interruption. In the case of broaching, no feed movement exists; each cutting edge, i.e. each tooth, executes only one working stroke, and new material is brought before the cutting edges by virtue of the fact that each tooth cuts deeper than the preceding one by an amount equal to the required feed rate (Fig. 1).

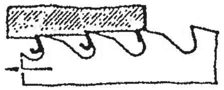


Fig. 1

The various machine tools produce the movements which are required by the different machining processes, and the cutting as well as the feed movements may be allocated either to the tool or the workpiece (Table 1).

A knowledge of the forces and velocities which occur during the various cutting processes is the essential basis for determining the size and material of the load transmitting elements together with the required driving power, in short, for the design of machine tools. Whilst, however, the interest of the research worker, and to a certain extent that of the cutting tool manufacturer, is concentrated on the phenomena which occur during cutting and on the relative influence of various parameters (tool, workpiece material, cutting conditions, etc.),* the machine tool designer need only know the order of influence of the various parameters and those results of research work which are essential for the actual design of the machines.

The resistance caused by deforming the workpiece material and by frictional forces acts in the form of a cutting force on the tool (action), and on the workpiece (reaction). The various parts of the machine tool (the structure, slides, workpiece and tool carriers, etc.), must be able to carry the resulting loads, and the driving elements must transmit the corresponding forces and torques at the required velocities.

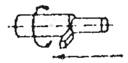
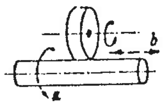
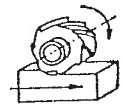
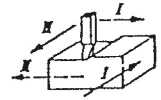
In general, a hyperbolic relationship exists between the *cutting speed* (v), which is the relative velocity of the cutting movement between cutting edge and workpiece, and the tool life (T) between re-grinds (Fig. 2). Taylor has expressed this relationship by the equation:

$$v \times T^{y} = C_T$$

where y is a parameter which depends on the materials of the workpiece and the tool,[2] and C_T, the cutting speed at which the tool life is one minute, is called the "Taylor Constant". It should

* The questions of metal cutting research, the problems of tool and workpiece materials, of chip formation and machinability have been dealt with in detail in many books, see ref. 1.

TABLE 1

Type of Machining Operation		Cutting Movement ⟶	Feed Movement ·······→
Turning		Workpiece	Tool
Drilling		Tool	Tool
Cylindrical Grinding		Tool	Workpiece (a) and Tool (b) or Workpiece ($a + b$)
Milling		Tool	Workpiece
Planing (*I*) *and Shaping* (*II*)		Workpiece (*I*) Tool (*II*)	Tool (*I*) Workpiece (*II*)

be mentioned that the influence of the chip section is not considered in the Taylor equation, and that it is necessary to allocate different values of C_T to different chip sections.[1] Permissible cutting speeds depend upon the material properties of tool and workpiece, the shape of the tool, the cutting conditions (e.g. with or without coolant) and the chip section. When a tool is used in normal production the tool life is the sum of all periods during which the tool actually cuts before a re-grind becomes necessary. In order to determine the economic cutting speed, it is necessary to consider the tool life and its relation to the time required for removing, re-grinding and re-setting the tool; the tool life which has to be selected for a particular operation (60, 240, or 480 min, are often recommended, see p. 5) depends, therefore, not only on technical but also on economic considerations.

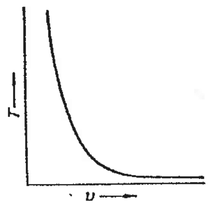


Fig. 2

The feed rate, which together with the depth of cut determines the chip section, affects—according to the cutting process in question—the cutting time, the surface finish, the tool life and the cutting force. Recommended values have been established for use by planning engineers and designers. As an example the German Committee for Economic Production (the AWF) have issued a leaflet, AWF 158 (1949), which recommends rake angles, cutting speeds, specific cutting pressures, etc., for cases of different tool and workpiece materials. With the ever-increasing application of carbide tools, especially for the machining of alloy steels, these recommendations today have only limited validity. The establishment of new values depends upon the results of long term researches, and such results are not as yet available.* Hence no concise picture exists to date. The values published in AWF 158 provide, however, a good idea of the order of magnitude of the various parameters

* See ref. 3.

The longitudinal force is:

$$P = 1{\cdot}15 \sum b(C_1.a^{0{\cdot}85} + C_2.k - C_3.\gamma - C_4.\alpha) \quad [\text{kg}]$$

The force at right angles to P which acts against the guiding faces is:

$$P_n = 1{\cdot}15 \sum b(C_5.a^{1{\cdot}2} - C_6.\gamma - C_7.\alpha) \quad [\text{kg}]$$

where b is the width of cut (mm)
k is the number of chip breaker grooves
γ is the rake angle
α is the clearance angle
a is the depth of cut per tooth (mm).

Table 8 shows some of the constants C_1 to C_7 given by Nalchan for Russian steels.

TABLE 8 (METRIC SYSTEM)

Material*	C_1	C_2	C_3	C_4	C_5	C_6	C_7
Steel 20 . .	115	0·060	0·20	0·12	55	0·018	0·045
Steel 35 . .	160	0·080	0·24	0·13	125	0·053	0·090
Steel 45 . .	220	0·108	0·32	0·14	215	0·081	0·117

* Russian Steels

B. GENERAL REQUIREMENTS OF THE MACHINE TOOL

The machine tool in the workshop has to satisfy the following requirements:

(1) Within permissible limits the specified accuracy of shape, dimensional accuracy and surface finish of the components produced on the machine must be obtainable consistently and, as far as possible, independently of the operator's skill.

(2) The operational speeds and rates of metal removal provided by the machine must be in accordance with the latest developments in materials and tools and, therefore, ensure the possibility of high productivity. In addition, the design must be such as to enable it to cope with future developments in these matters in order to prevent the machine from becoming obsolete in a short time.

(3) In order to be competitive in operation, the machine must show a high technical and economic efficiency.

The meeting of these requirements depends on a number of factors which can be considered in relation to each as follows:

Requirement 1. The machining quality depends not only upon the machine tool itself, but also on other factors. Examples of these factors are: the tool (shape and material, rake angles, quality of cutting faces), the tool carrier (stiffness of milling arbor or boring bar, etc., quality of tool clamping), the workpiece and its clamping (machinability of the material, stiffness of the workpiece and of the clamping fixture, accuracy of centres, etc.), the selected cutting conditions (cutting speed, depth of cut, feed rate), and changes of working conditions which may occur during the operations and may be caused by the machining processes (tool wear and crater formation, temperature changes, etc.).

These factors must not be overlooked by the machine tool designer, because he has to take the necessary steps which make it possible to obtain optimum working conditions. He cannot influence the shape, machinability and stiffness of the workpiece which may be put on the machine, but he can make certain that the clamping devices have the necessary stiffness; he cannot predetermine the dimensions of surfaces which have to be milled or the diameters and depths of bores

which have to be machined, but he can arrange for the size and stiffness of milling arbors and boring bars required for certain operations to be limited only by the dimensions of the tool and the workpiece and not by the design of the main spindle; he has no direct influence upon the selection and maintenance of the cutting tools which are to be used in his machine, but he can design the machine in such a manner that the best possible tools can be applied under conditions which satisfy the requirements of the latest developments in the field.

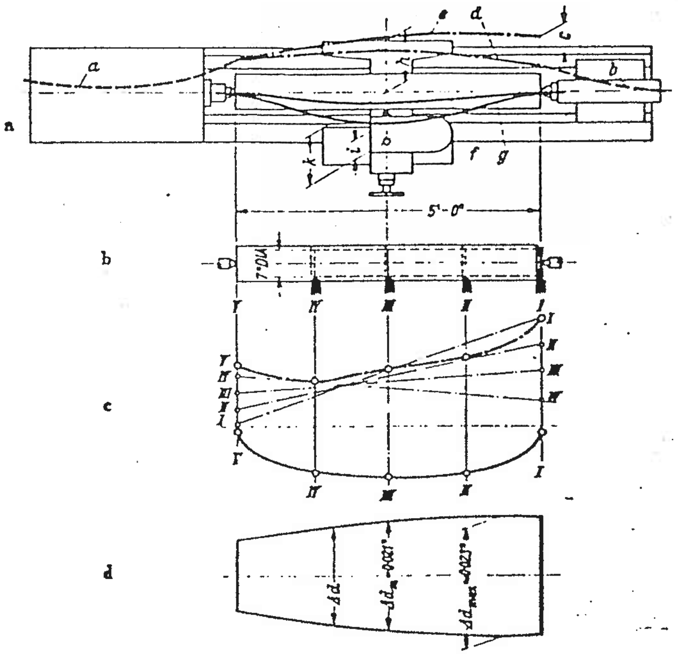


FIG. 33a–d

This book is concerned with the design of machine tools. As a basis for all discussions concerning layouts and considerations of power, efficiency, performance and working quality of the machines, it will be assumed that optimum working conditions are applicable, as far as the cutting tools, the tool carriers and the handling of workpieces are concerned.

(a) The *shape of the workpiece* depends on the instantaneous relative position of the machine parts which carry the tool and the workpiece. When the machine is idle, deviations from the geometrically required relative positions of the various parts are caused only by inaccuracies in the manufacture of the machine. Their magnitude depends, therefore, upon the quality of work produced in the workshops of the manufacturer. However, as soon as a machine tool begins its productive work, i.e. when it is running under load, the influence of the design becomes decisive.

Mechanical deformations, changes in the thickness of oil films in bearings and slideways, etc., can be caused by the operational forces and loads or by temperature changes of the various parts, and these together influence the relative position of tool and workpiece. The designer has to take these deformations and changes into consideration when deciding on the shape and size of the machine parts, when designing coolant and lubricating systems, etc. Figure 33 shows an analysis of typical measurements concerning the deformations of a lathe.[28] The deformations were measured under static loads which represented the following cutting force components: P_1 = 2640 lb,

P_3 = 945 lb (see Fig. 3). These force components could be expected on such a machine during roughing operations. Figure 33a shows the deformations and displacements of machine and workpiece at the moment when the tool is half way between the two centres.

The deflected centre lines of the spindle (a) and of the tailstock sleeve (b) are shown as dashed lines. The chain dotted line indicating the deflected centre line of workpiece (d) is further displaced by an amount (c) to position (e), by virtue of the fact that the workpiece "climbs" on the tailstock centre. As a result, that point of the workpiece centre line which lies opposite to the cutting tool is moved away from the tool edge by an amount (h) compared with its initial position. Under the action of cutting force component P_3 the bed shears are bent in accordance with line (f), and, as the front shear is further deformed, the total deflexion of the saddle slideway is indicated by line (g). Moreover, backlash in the cross traverse screw and in the slideways accounts for an additional displacement of the tool carrier by an amount (i), so that the tool is removed from its initial

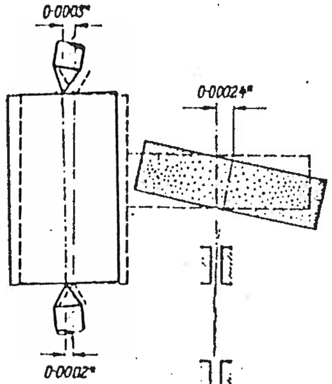


Fig. 34

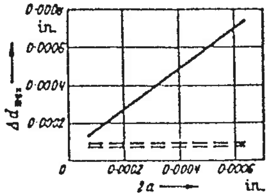


Fig. 35

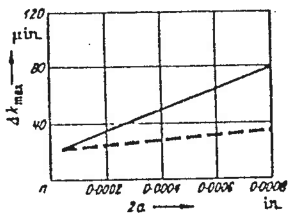


Fig. 36

position by a total amount (k). Under load, the distance between cutting edge and workpiece centre line increases, therefore, by an amount ($h + k$), and the resulting increase of the workpiece diameter becomes $\Delta d = 2(h + k)$. Corresponding displacements have been estimated for positions *I, II, III, IV, V* of the tool (Fig. 33b and c) and the deviation from the cylindrical form of the workpiece caused by the resulting changes in diameters Δd is shown in Fig. 33d. It may be pointed out that Fig. 33d is drawn at half the scale of Figs. 33a to 33c.

Although the forces encountered in grinding operations are relatively small, the resulting deformations of the centres, the workpiece, and the spindle carrier cannot always be neglected. K. E. Schwartz[29] found deformations shown in Fig. 34 under a radial cutting force component of 66 lb and H. Schuler[30] determined the maximum deviations from the cylindrical shape (Δd), (Fig. 35) and from the circular shape (Δk), (Fig. 36), as functions of the net reduction of the workpiece diameter (twice the depth setting = $2a$). With increasing depth setting the forces, deformations and deviations from the required dimensions of the workpiece grow. However, the cutting forces drop again during sparking out operations, and the errors are reduced although not fully eliminated (dashed lines in Figs. 35 and 36).

If a one-sided driver is used during turning and grinding operations, the transverse force (P_s), which acts on the headstock centre, pulsates and rotates (Fig. 37a).[28,31] The influence of this pulsation which can be eliminated by the application of a twin driver (balancing the transverse force component), is not considered in Figs. 35 and 36.

The influence of temperature rises in various parts of the machine has been investigated by Eisele, Kronenberg and others.[32] The latter measured the displacements of a vertical milling machine spindle caused by temperature rises. By suitable changes in the design of the bearings, and by careful selection of the lubricants, the temperature rise after seven hours' running was reduced from 17° to 3°C, and the resulting displacement from 0·1 to 0·015 mm. In a Russian investigation[33] it was found that the error in parallelism between two ground faces of rings, 100 mm dia., could be reduced from 0·012 to 0·005 mm by blowing hot air through the bed of the surface grinding machine, thus reducing the temperature gradients and the resulting deformations of various parts of the machine.

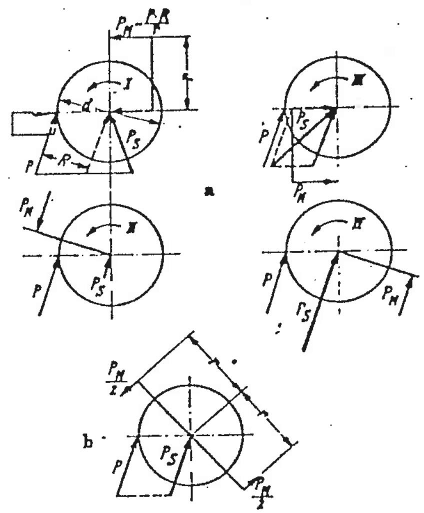


Fig. 37a and b.

When the designer considers these points, he must not forget the ultimate purpose of his machine. The tolerances which are laid down in different acceptance test specifications concerning the accuracy of shape, e.g. lathe turns true; cylindrical surfaces machined on a milling or planing machine are plane or parallel, etc., refer always to finishing operations, and the forces occurring during finishing operations are usually relatively small. On the other hand, even today many machine tools are used for both roughing and finishing. Table 9 shows the conditions in a first class medium size British machine tool factory.* The machine tool must, therefore, often be designed in such a

Table 9

Machine Tools		Employed for		
Type	Number off	Roughing only	Finishing only	Roughing and Finishing
Centre Lathes	34	2	—	32
Drilling Machines	12	—	12	—
Horizontal Borers	19	—	—	19
Cylindrical Grinding Machines	6	—	6	—
Surface Grinding Machines	6	—	6	—
Milling Machines	25	—	—	25
Planing Machines	8	8	—	—
Shaping Machines	3	—	—	3
Slotting Machines	2	—	—	2
Total	115	10	24	81
≈%	100	9	21	70

* H. W. Kearns & Co. Ltd., Broadheath, Manchester.

manner that the deviations and deformations during finishing operations are within permissible limits, although the deformations and deviations during roughing operations may exceed these limits. It is, however, important to ensure that roughing operations do not cause a reduction in oil film thickness, which may cause metal to metal contact thus inflicting permanent damage to the sliding surfaces, and that the highest stresses in the various parts of the machine lie well below the elastic limit of the material, so that no permanent deformations occur.

One limitation of these considerations must be mentioned. For certain finishing operations, especially those which serve for improving the surface quality, the tool may follow the shape of the workpiece as produced by a preceding operation. In this case, a complete elimination of inaccuracies of shape is, therefore, not possible, and the machine producing the required shape during a preceding operation must work within the final tolerances required from the workpiece.

(b) The *dimensional accuracy of the workpiece*, which may be influenced by the factors mentioned under (*a*), depends also upon the accuracy with which:

(1) the relative position of the moving parts of the machine and the dimensions of the finished surface can be measured, and

(2) the movement of various machine parts can be controlled.

The magnitude of a feed traverse is often determined by the number of revolutions of the driving screw. If this screw rotates under load, it may be subjected to wear and its value as a measuring device will deteriorate. The separation of measuring and driving elements, which has long been introduced into the design of jig boring machines, is today making more and more progress in other fields of machine tool design.

However, even if separate driving and measuring devices are employed, such as a traversing screw and a Vernier scale or a hydraulic cylinder and an optical scale, it is not usual to take measurements on the workpiece itself, but on the tool or workpiece carrier (table, cross slide, etc.). This means that workpiece deformations and tool wear are not covered by the measuring device. Moreover, even if the actual dimensions of the workpiece and, therefore, the magnitude of any necessary feed or depth setting can be determined accurately, errors in the slideways and in the driving elements for such movements will give rise to working inaccuracies.

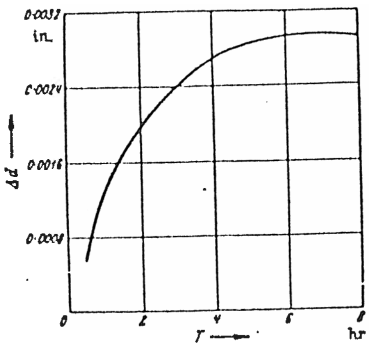


FIG. 38

Schuler[30] investigated the influence of the temperature rise in a cylindrical grinding machine upon the dimensional accuracy of the workpiece. The depth setting of the spindle head was determined by a rigid stop. Due to the temperature rise, the relative position between this stop and the workpiece changed over a working time T and a reduction in the diameter Δd as shown in Fig. 38 was measured.

(c) As already mentioned, the *accuracy and quality of a surface* produced on a machine tool depend very much upon the properties of the tool, the tool carrier, and the workpiece. In addition, the *surface quality* is affected by the following factors which the designer can control:

(i) The possibility of obtaining optimum cutting conditions (cutting speed, feed rate) by the provision of suitable speed change devices for spindle and feed drive.

(ii) The stiffness of the machine as a whole, of its components (bed, uprights, slides, tables, spindle) and of its operating elements (slideways and bearings, driving mechanisms, etc.).

The purely geometric influence of the cutting tool shape and of the feed rate upon the depth of the grooves in a turned surface is shown in Fig. 39. However, optimum surfaces can only be produced if the most suitable cutting speed, depending on tool and workpiece material, is

employed (Fig. 40)* and if bearings and slideways satisfy all requirements. For instance, the eccentricity (e) of a milling cutter can influence considerably the geometry of the machined surface (Fig. 41).[35]

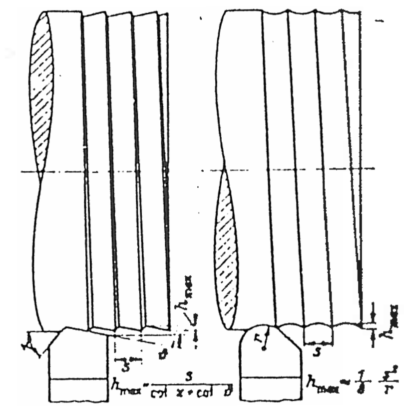


FIG. 39

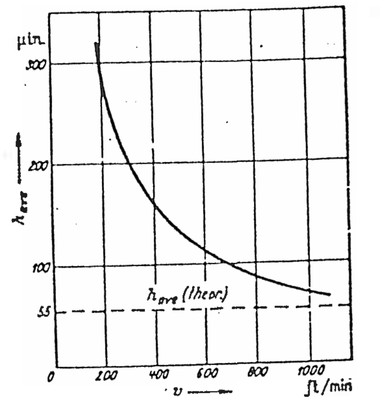


FIG. 40. Effect of cutting speed upon surface finish (from Kumar). Tool—tungsten carbide, negative rake (−3°); Material—mild steel (28 tons/in²); Depth of cut 0.002 in.; Feed—0.00875 in./rev.

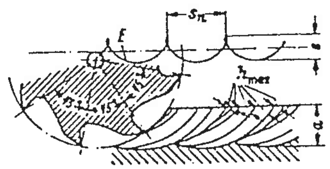


FIG. 41. Surface machined with eccentric milling cutter: a = depth of cut; E = path of cutter axis relative to workpiece; e = eccentricity of cutter; h_{max} = maximum chip thickness per tooth; s_n = feed per revolution of cutter. Number of cutter teeth: 8.

In addition to the conditions at the cutting edge, which affect the actual cutting process and its contribution to the generation of the machined surfaces, the dynamic factors which cause changes in the relative position between tool and workpiece must be considered. Vibrations of the various components, both absolutely and in relation to each other, vibrations of the driving elements (torsional vibrations of gears, torsional and transverse vibrations of shafts and traversing screws), and vibrations in slideways and bearings will combine to create a complex vibrational system which may produce various types of surface roughness and waviness.

The vibration conditions at the cutting edge, and their influence upon the surface finish have been referred to (see page 3). In addition, the periodic variation in the depth of cut caused by vibrations of the cutting tool will create pure geometrical irregularities on the surface (Fig. 42).[36]

FIG. 42. Influence of tool vibrations on the turned surface (from Kronenberg): ——*a*) normal position of turning tool; —·—·— *b*) highest position of turning tool during vibration; - - - - *c*) lowest position of turning tool during vibration; —) turned surface.

Today's requirements of the finished products (high rotational speeds of engine shafts, high working pressures in cylinders, very high or very low working temperatures) and the problems of manufacture, especially with reference to interchangeability, make it essential for the designer to take fully into consideration all these conditions which may influence the quality of the finished workpiece.

* From work carried out in the Royce Laboratory of the Manchester College of Science and Technology.[34]

Requirement 2. The requirements of productivity and economy affect the question of working speeds and rates of metal removal. It may, however, be appropriate to quote here a statement made by Schlesinger[37] some thirty years ago:

"It is not essential for a machine tool to produce a maximum weight of chips per minute, but to produce as many workpieces as possible. In general, machine tools are not produced for use in chip factories, but for use in engineering workshops." In other words, the task of a machine tool is not to produce a maximum quantity of chips in a minimum time, but to produce economically workpieces of given shapes and dimensions.

However, it will often be found that the rate of metal removal, in other words, the quantity of material which has been cut from the workpiece in unit time will be decisive from the point of view of machining economy of certain workpieces, especially in the case of roughing operations. In such cases, it is necessary that:

(a) The cutting speeds and feed rates obtainable on the machine should satisfy the requirements and possibilities of the best available high performance tools
(b) The strength and stiffness of the machine elements and of the machine as a whole should be satisfactory under the operating forces which are likely to occur when the chip sections are very large
(c) The driving elements (motors, gears, etc.), the bearings and slideways shall be adequate to deal with the large cutting forces, and with the power required by large cutting forces and high operational speeds
(d) Provision be made for lubrication and cooling of the cutting tools
(e) Devices for removing the large amount of swarf produced be provided.

Whilst the points covered by (d) and (e) depend partly on the application and layout of auxiliary equipment, points (a) to (c) can be analysed quantitatively. Apart from the maximum dimensions of the workpieces which have to be machined, the capacity specification of a machine must cover the following:

(a) 1. The optimum cutting speeds and feed rates corresponding to the cutting tools to be used and the materials to be machined.
  2. The spindle speeds (or table speeds where applicable) and feed rates available on the machine.
(b) The permissible maximum values of cutting forces. These limit the maximum chip sections which can be machined on the various materials, and depend on the strength and stiffness of the workpiece and also of the machine.
(c) The forces, torques and powers which are to be transmitted by the driving motors and driving gear and carried by bearings and slideways.

The various relations between these factors can be shown graphically in what are termed performance nomograms. These not only help the planning engineer in determining the optimum conditions for employing his machines in the workshop, but also enable the designer to determine the basic parameters of his design.[38]

In order to develop such nomograms the following information is required:

(a) *Workpieces*
  1. Materials
  2. Maximum and minimum dimensions

(b) *Cutting tools*
  1. Materials
  2. Shape (rake angles, dimensions, etc.)
  3. Required life

(c) *Chip section*
  1. Depth of cut
  2. Feed rate

(d) *Speed change devices*
  1. Spindle speeds or, in the case of straight line movements, working speeds
  2. Feed rates

(e) *Drive*
  1. Power
  2. Efficiency (see page 37)

(f) *Design elements of the machine*
  Permissible maximum loads (forces, torques) on the structure, the tool and workpiece carriers.

Use may be made of this information in the following manner:

(a) From the information concerning the workpieces, it is possible to determine the specific cutting resistance (material), and the cutting forces which are permissible from the point of view of workpiece strength (workpiece material and dimensions). In addition, it is possible to determine the dimensions of those parts of the machine tool which carry or support the workpiece.
(b) The available tool materials and the workpiece materials which have to be machined determine the cutting speeds and feed rates. With knowledge of the dimensions of tool or workpiece respectively, the corresponding speeds can be calculated.
(c) The chip section (feed rate × depth of cut) is used for the calculation of the cutting force.
(d) After a range of spindle or working speeds has been established, it is possible to make a critical comparison of the required and the obtainable conditions.
(e) By using the values obtained under (a) to (c), the required net power at the cutting edge can be calculated. If, in addition, the efficiency of the drive between the motor and the cutting edge is known, the power required for the driving motors can be determined.
(f) Finally, the strength and stiffness of the various parts of the machine tool can be checked against the conditions previously calculated.

As examples for the layout and application of typical nomograms, in which the above-mentioned relations are shown graphically, two types of machines may be discussed which are of great importance in the average machine shop. For the first the centre lathe is selected. It is the basic machine for producing rotational components. The second type selected is the slab milling machine which produces plane surfaces.

It is not intended, however, to cover all possible cases, but to show only the basic principles which can be applied when developing such nomograms. It is advantageous to use the double logarithmic system, because in it most relationships appear as straight lines. For reasons of simplicity of presentation, preferred numbers have been used for the scales of the various coordinates of the nomograms.

1. CENTRE LATHE

The parameters are:

(a) *Workpieces*
  1. Materials
  2. Dimensions: Turning diameter (d), length between centres (l)
(b) *Tools*
  1. Materials
  2. Rake angles
  3. Life
(c) *Cutting conditions*
  1. Cutting speed (v)
  2. Feed rate (s)
  3. Depth of cut (a).

Recommended cutting speeds can be taken from Table 7. In the top left-hand corner of the nomogram (Fig. 44), the cutter diameter d (horizontal lines) and the cutting speed v (straight lines sloping at an angle of 45° downward from right to left) are plotted in such a manner that their intersections determine the spindle speed, $n = v/\pi d$ (vertical lines). These vertical lines are intersected by the less sloping lines for the number of teeth n_t and from these intersections are plotted the lines (sloping downward from left to right) which represent the product $n.n_t$. In the top right-hand corner the intersections between the vertical lines for the depth of cut a, and the horizontal lines for the cutter diameter d, represent the values of $\sqrt{(a/d)}$ (lines sloping downward at 45° from right to left). From the intersections of these sloping lines with the $(n \times n_t)$ lines in the middle of the nomogram are plotted the lines indicating values $(1/n.n_t)\sqrt{(a/d)}$, which slope downward at 45° from left to right. The intersections of these lines with the horizontal lines, which indicate the feed rate s, determine the position of the vertical lines for the middle chip thickness $h_M = (s/n.n_t)\sqrt{(a/d)}$, in the middle and at the bottom of the nomogram. Values of k_M for different materials are then plotted as functions of the middle chip thickness h_M (see Fig. 22a). At the right-hand side of the nomogram the vertical lines, which indicate the depth of cut a, and the horizontal lines, which indicate the width of cut b, intersect at points from which lines indicating the product $a.b$ slope downward from left to right. From the intersections of these sloping lines with the horizontal lines for the feed rate s originate the lines sloping downward from right to left, which indicate the rate of metal removal $V = a \times b \times s$. At the intersection of these lines with the horizontal lines for k_M vertical lines are plotted, and these indicate the net cutting power $a \times b \times s \times k_M$ in the right-hand bottom corner of the nomogram.

The use of this nomogram may also be shown through a simple example. A workpiece of mild steel, 3 in. wide, is to be reduced in thickness by a cut 0·16 in. deep, with a 4 in. dia. milling cutter having 8 teeth. The cutting speed is to be 100 ft/min (shown as 1200 in/min in the nomogram) and the feed rate 6 in/min. The spindle speed is obtained from the cutting speed and the cutter diameter ($n = 100$ rev/min) (Lines *I*). These lines (*I*) are continued from left to right and give, together with lines (*II*) starting at the top right-hand corner $\sqrt{(a/d)}$, the feed rate ($s = 6$ in/min) and the k_M line (*b*) (for mild steel) the value for k_M at the bottom. The vertical line representing $a = 0{\cdot}16$ in. is continued downwards and the values for $b = 3$ in. and $s = 6$ in/min show the rate of metal removal to be $V = 2{\cdot}9$ in^3/min, (Lines *III*), and this results in a net cutting power of 2·2 kW (Line *IV*).

The net cutting power is not directly proportional to the rate of metal removal. It depends upon the composition of volume V and is influenced in particular by the chip thickness (see page 12). If the same volume (2·9 in^3/min) is removed by doubling the depth of cut and halving the feed rate, the net power required would be 2·5 kW (Lines *V*, *VI* and *VII*), whilst it would drop to 2·0 kW if the depth of cut is halved and the feed rate doubled (Lines *VIII*, *IX* and *X*).

Requirement 3. A knowledge of the net power at the cutting edge as already discussed is not the only parameter which must be known for determining the required power of the driving motor for the machine tool. As the input power is equal to the net power divided by the efficiency, it is important to know the efficiency of the machine drive. Schlesinger[10] has established power balances from which the effect of the various driving elements upon the total efficiency can be deduced. However, this efficiency depends not only upon the loading of the machine but also upon the output speed, the influence of which may be considerable in view of the large speed ranges available in present day machines.

The net power for the feed drive is proportional to the product of feed rate and the cutting force component acting in the direction of the feed movement. It is, however, relatively small compared with the total power required. In the case of *turning operations*, the cutting force component in the direction of the feed movement P_2 is rarely greater than half the tangential force component P_1, i.e. $P_2 < 0{\cdot}5\, P_1$.

The ratio between feed rate v_s (in/min) and the cutting speed v (in/min) will rarely exceed 0·01:

$$\text{i.e. } v_s/v < 0{\cdot}01.$$

The ratio between the net power values for the feed drive N_s and that for the spindle N_v is then

$$\frac{N_s}{N_v} = \frac{P_2 . v_s}{P_1 . v} < 0{\cdot}5 \times 0{\cdot}01 \times 100 < 0{\cdot}5\%$$

In the case of *milling operations*, the tangential force component is approximately equal to the cutting force component acting in the direction of the feed movement, and the ratio between feed rate s and circumferential speed of the cutter v will not be more than 0·02, i.e. $s/v < 0{\cdot}02$. The ratio between the two net power values is, therefore, less than 2 per cent.

The ratio between the corresponding power input requirements would not be quite as low as that of the net power values because of the relatively low mechanical efficiencies of the usual feed drive gearboxes. Ratios up to 5 per cent in the case of a centre lathe and up to 20 per cent in the case of a milling machine have been found. Nevertheless, it can be safely said that as far as the total efficiency of the drive is concerned, the drive for the cutting movement is the decisive factor.

In the case of spindle drives with wide speed ranges and a large number of steps, the power required for idle running of the machine is often considerable. It can be up to 50 per cent of the total power requirements.[38] In the Machine Tool Laboratory of The Manchester College of Science and Technology, M. M. Sadek investigated the efficiency of a production milling machine (speed range 19–1700 rev/min, driving motor power 5 kW) and found the conditions shown in Figs. 45 and 46. H. Stute and E. V. D. Linde[39] found that the following relationship existed for main drives using ball and roller bearings:

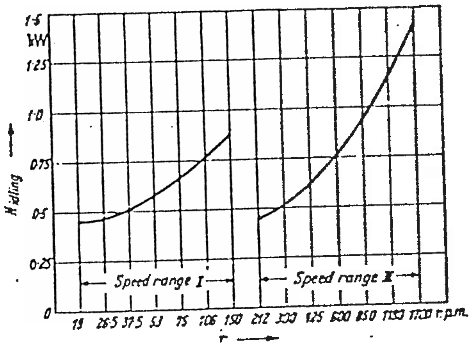


Fig. 45

$$N_{input} = N_{idling} + a . N_{net}$$

and

$$\eta = \frac{N_{net}}{N_{input}}$$

$$= \frac{1}{(N_{idling}/N_{net}) + a}$$

Hence

$$a = \frac{N_{input} - N_{idling}}{N_{net}}$$

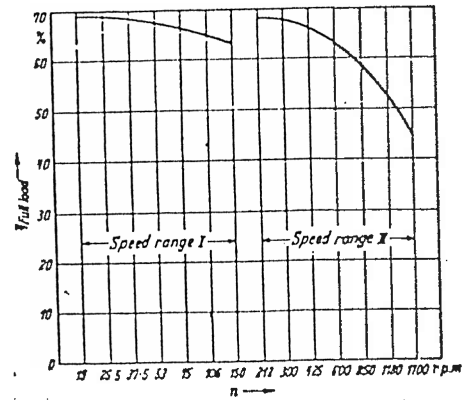


Fig. 46

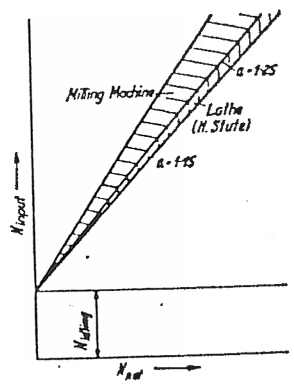


Fig. 47

They also found that the value of a appears to be generally between 1·15 and 1·25 (Fig. 47). However, this range may be wider in the case of very large speed ranges. In the case of the milling machine used by Sadek, the following conditions were found for the case of maximum load and maximum spindle speed (n = 1700 rev/min):

$$N_{input} = 5\text{ kW} \quad ; \quad N_{net} = 2{\cdot}3\text{ kW}$$

$$N_{idling} = 1{\cdot}4\text{ kW}; \quad a = \frac{5 - 1{\cdot}4}{2{\cdot}3} = 1{\cdot}56$$

and in the case of minimum spindle speed (n = 19 rev/min):

$$N_{input} = 4\text{ kW} \quad ; \quad N_{net} = 2{\cdot}8\text{ kW}$$

$$N_{idling} = 0{\cdot}5\text{ kW}; \quad a = \frac{4 - 0{\cdot}5}{2{\cdot}8} = 1{\cdot}25$$

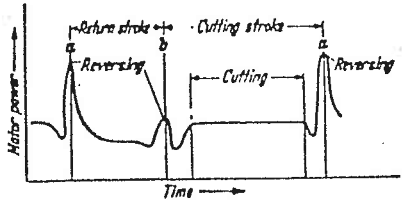


FIG. 48

In the case of machines with reciprocating movements, the power required for the feed drive is of secondary importance because the feed movement does not take place simultaneously with the cutting movement. However, the power required for the reversing and for the return movement of the table represents a considerable wastage. Figure 48 shows the power requirements of a planing machine drive. As the reversing power varies considerably, it is better to establish a balance for the planing machine on the basis of the work done. It is then possible to determine the efficiency of the machine as a function of the work done during a cutting operation over a given length of stroke (Fig. 49). The ratio between the work done during reversing and the total work decreases with increasing stroke and the total efficiency rises. In the case of a shaping machine, Schlesinger found a maximum efficiency of 60 per cent (Fig. 50).[40]

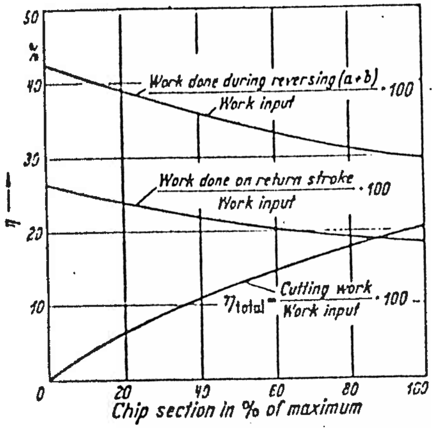


FIG. 49

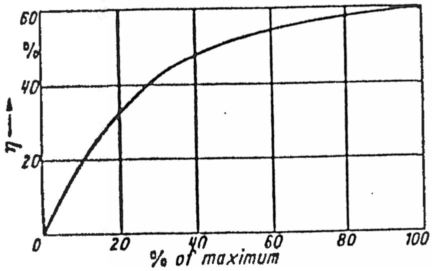


FIG. 50

The final assessment of the performance of a machine tool is determined by its economic efficiency and not only by the optimum use of the available driving power, the static and dynamic load carrying capacity and the performance of the cutting tools. In order to obtain the highest possible overall efficiency of a machine tool in production, the following points must not be overlooked:

(a) The operator must be able to set up and control the machine rapidly, safely and with the minimum amount of fatigue.

(b) It must be possible to carry out maintenance and any necessary repair work without difficulties and in the shortest possible time.

The factors and general considerations which control these specific points may be grouped in the following manner:

(a) Standards concerning the logical arrangement and direction of movement of operating handles, wheels, etc., have been introduced all over the world

1. STIFFNESS AND RIGIDITY OF THE SEPARATE CONSTRUCTIONAL ELEMENTS AND THEIR COMBINED BEHAVIOUR UNDER LOAD

The question of stiffness (a term sometimes confused with that of rigidity)* is often more important in the design of metal cutting machine tools than that of load carrying capacity, because the stresses which correspond to the permissible deformations are generally far less in value than those permissible for the various materials.

The idea of using stiffness or rigidity as a design or performance parameter was first proposed by Krug[1] who suggested that stiffness in this connexion should be measured by the ratio $\frac{\text{load in kg}}{\text{deformation in } \mu}$.

However, the problem is not only one of deformation under static load, such as the weight of the workpiece plus a cutting force which is assumed to act as a static load. The dynamic performance of the machine under the influence of pulsating cutting forces and inertia loads, which arise, for instance, during rapid control operations, is of extreme importance.

Moreover, it is necessary for the designer to examine and consider not only the stiffness of single elements, but also the cumulative stiffness of the groups and systems formed by these elements. The cumulative stiffness of the machine parts, the elements which join them (bolted connexions, oil films in bearings, slideways, etc.), of the driving elements (oil columns, feed screws, etc.), and of the resulting combinations of these must be such as to ensure that the resulting static and dynamic relative displacements between tool and workpiece are within permissible limits.

The term "stiffness" has to be considered, therefore, from the following points of view:

(a) Static stiffness against deformation under static loads;

(b) Dynamic rigidity, i.e. behaviour during vibrations under pulsating and inertia forces.

These will now be discussed:

(a) Amongst the *static deformations*, the more important are perhaps those which are caused by bending and torsional loads because these produce misalignments and displacements of the guiding elements and thus working inaccuracies of the machine. The forces which produce such loading conditions and deformations are:

1. The weight of moving parts of the machine.
2. The weight of the workpiece.
3. The cutting force.

It is not sufficient to determine a single value of stiffness. The variation of the deformation conditions depends not only upon the magnitude and direction of the forces, but also upon the instantaneous positions of their points of application and any change of these conditions is of great importance because it influences the type and magnitude of the various deformations which affect the working accuracy of the machine. However, the parts of the machine which are subjected to these forces must not be considered only as complete units (beds, uprights, slides, etc.). The deformations of their wall panels must also be studied because, for example, the deformation of a side

* The terms "rigidity" and "stiffness" are not always clearly distinguished in literature and speech. Whilst the author prefers the term "stiffness" for the ratio $\frac{\text{static load}}{\text{deformation}}$ and the term "rigidity" for the vibration behaviour, he has not changed the terminology of various research workers when quoting verbatim from their work.

wall in a headstock which carries a main spindle bearing may considerably affect the position of the spindle and, therefore, again the working accuracy of the machine.

In the case of bending the "spring constant", $\frac{\text{force}}{\text{deflexion}}$, which determines the stiffness value (c_b) is proportional to the product of Young's Modulus E and the second moment of area I. In the case of torsion the value for the stiffness is taken as $c_t = \frac{\text{torque}}{\text{angle of twist}}$. The stiffness is thus influenced by the material, the size and the shape of the section under load.

A comparison of the stiffnesses of four cross-sections of equal height h and of equal cross-sectional area (i.e. of equal weight per unit length of a beam) is given in Fig. 54.[2]

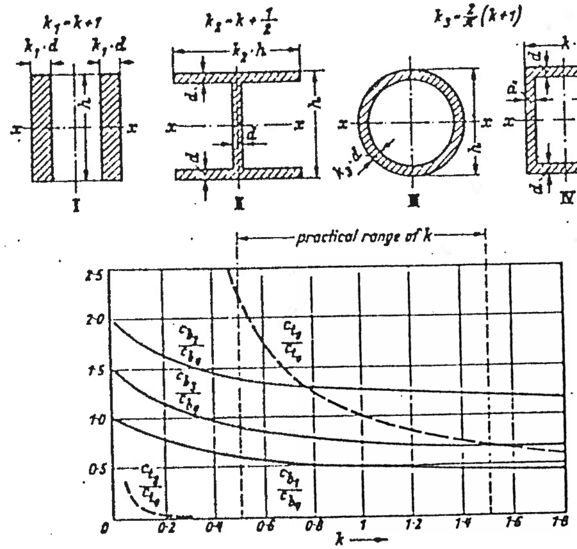


Fig. 54. Bending and torsional stiffness of different cross-sections. c_{b_1}—Bending stiffness of section I; c_{b_2}—Bending stiffness of section II; c_{t_2}—Torsional stiffness of section II; c_{b_3}—Bending stiffness of section III; c_{t_3}—Torsional stiffness of section III; c_{b_4}—Bending stiffness of section IV; c_{t_4}—Torsional stiffness of section IV.

Within practical limits of the ratio width-to-height (k), i.e. $k = 0{\cdot}5$ to $1{\cdot}5$, the closed box cross-section appears most favourable because, compared with the tubular section, the slightly lower torsional stiffness is more than compensated for by the increased bending stiffness. In addition, the ratio between free length and cross-sectional area is important and has to be given special consideration having regard to the properties of the material employed. This becomes particularly important when for certain reasons not only the stiffness but also the strength has to be considered and when, for instance, a choice has to be made between the use of cast iron or a welded steel fabrication for a machine structure.

Krug has pointed out the possibility of saving material by using rolled steel instead of cast iron. Young's Modulus E for steel is almost twice that of cast iron and the permissible tensile and bending stresses for cast iron may be 30 to 60 per cent of those permissible for ordinary mild steel. Krug has also shown that if both strength and stiffness of the material are to be fully exploited, considerable savings in material are possible if, for example, in the case of a rectangular base subjected to bending, an optimum ratio between depth and free length of the beam is chosen.

However, the simple ratio between the depth and free length of the beam suggested by Krug is not decisive. This can be shown if a calculation is made of the minimum volume of material required for a beam which is freely supported at the ends and loaded at the centre (Fig. 55) this representing a simplified form of a machine bed with two side walls.

Apart from the stiffness of single parts and their arrangement in a machine structure, the effect of the joining elements upon the total stiffness is important. Deflexion of flanges, elongation of fastening bolts, variations in the play between parts and deformations of the load carrying elements (balls, rollers, oil films) in slideways and bearings all play their part.

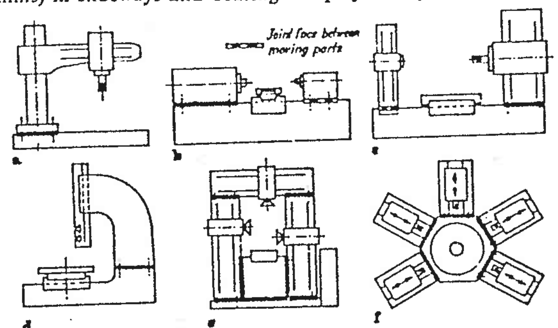


Fig. 62. Examples of bolted joints in machine tool structures. a—Radial drilling machine; b—Centre lathe; c—Horizontal boring machine; d—Vertical slotting machine; e—Plano milling machine; f—Multi-spindle drilling units.

Reduction in stiffness caused by joints in a structure (Fig. 62)[8] can be restored, at least partially, by a suitable arrangement of fastening bolts. For instance, an accumulation of bolts on the compression side of a flange joint subjected to bending is less favourable than a uniformly distributed arrangement, whereas an accumulation on the tensile side may have favourable effects. If a flanged joint is subjected to torsion a uniform distribution of the bolts along the circumference provides optimum conditions. If pre-loads are relatively small the bending stiffness of the flanged joint increases considerably with the pre-load, whilst the torsional stiffness is only slightly affected.

If the pre-load exceeds the minimum value which would be required to prevent opening of the joint under maximum load, a further increase in the pre-load has only little effect on the bending stiffness and no effect whatsoever on the torsional stiffness. The joint faces of two flanges should be as large as possible, plane and of a high surface finish. The stiffness of the flanges contributes, of course, considerably to the bending stiffness of the joint.

By changing the distribution of fastening bolts from the original arrangement of 12 to 10 which provided better load distribution, and by stiffening the flange (two, four or six stiffeners), the stiffness of an upright and its flange joint (Fig. 63) could be increased by almost 50 per cent.[9] Although the

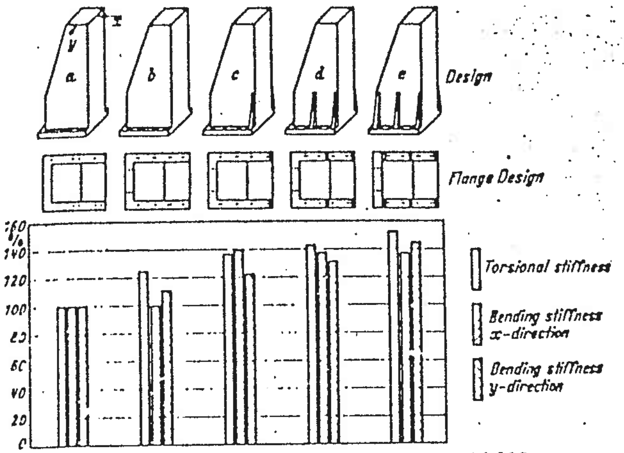


FIG. 63. Effect of flange design on stiffness (from Bielefeld).

increase in the flange thickness increases its second moment of area, the elongation of the necessarily longer bolts also becomes greater, and this results in an increased danger of the flange lifting off its supporting surface.

The designer will always endeavour to join the various parts of a machine structure as rigidly as possible so that they work together as a single unit, examples being found in the bed of a lathe with head and tail stock, the base plate of a radial drill with column, radial arm and spindle head. Different problems arise, however, when the operation of a machine requires its parts to move relative to each other, as is the case with spindles in their bearings, slides moving on their slideways, etc.

Many years ago, Kiekebusch* investigated the conditions of a main spindle in a lathe headstock. From a practical point of view the spindle is neither rigidly clamped, nor can it be assumed to be freely supported in its bearings. The deformation of a spindle depends, therefore, not only upon its own stiffness, but also upon the inclinations of its bearings under load and, therefore, upon the stiffness of the bearing carrying structure, in this case the headstock, the bearing itself (bush or ball bearing), and its location in the structure. In a recent investigation, K. Honrath[10] found that the main components in the displacement of a spindle are the deflexion of the spindle (50 to 70 per cent) and the deformation of the bearing (50 to 30 per cent). He confirmed Kiekebusch's observation to the effect that with growing load, the rate of deformation is at first high and decreases later (Fig. 64). This may be explained by the fact that with growing load and thereby increasing deformation of the

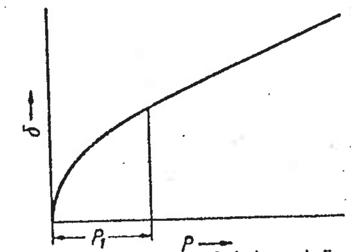


FIG. 64. Deflexion of lathe spindle. P—Transverse force acting on the spindle nose; δ—deflexion measured at the spindle nose; P_1—recommended preload.

* See ref. 28 Part I.

bearing elements, the load becomes more evenly distributed over the various parts of the bearing so that the specific pressures on the elements decrease, (Fig. 65).[10] Moreover, it is possible that the bearings exert more and more a "back bending effect" so that the conditions change from those

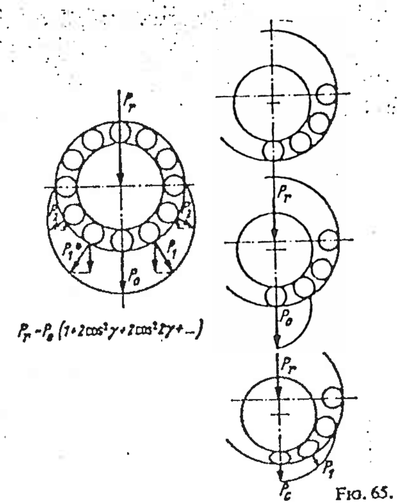


FIG. 65. Load distribution in a ball bearing (from Honrath).

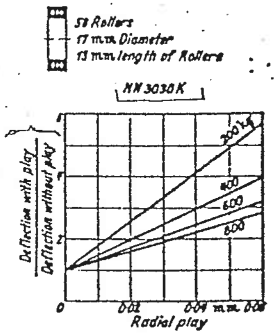


FIG. 66. Stiffness as a function of play in a roller bearing (from Honrath).

of a freely supported to those of a clamped beam. This may also be the reason for the stiffness to be influenced by the play in the bearing (Fig. 66)[10] especially if such play becomes negative when a pre-load is applied.

The results of measurements by Honrath reproduced in Fig. 67[10] showed that the share of the bearing in the total deformation could be reduced from 16 μ in the case of a bearing without pre-load to 5 μ when the bearing was pre-loaded to an interference of minus 15 μ. The clamping moment which the bearing also exerted on the spindle resulted in a reduction of the spindle deflexion from 14 μ to 11 μ so that the total displacement of the spindle nose was reduced from 30 μ to 16 μ. In the case of a spindle supported in plain bearings, it is often difficult to predict accurately the effective length between the bearings and the back bending and reinforcing effect of the bearing bushes. If a spindle is supported by ball or roller bearings, however, its bending deflexion can be determined with good approximation by means of Mohr's graphical method (Fig. 68)[10].

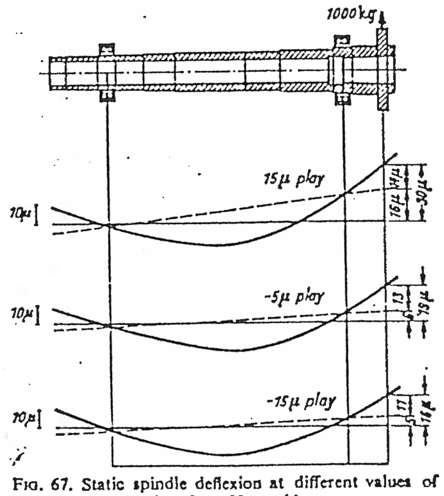


FIG. 67. Static spindle deflexion at different values of play (from Honrath).

The distance between two spindle bearings is, of course, an important design parameter. Figure 69[10] shows the spindle deflexion in μ/kg at the point of application of a force P as a function of the ratio $\frac{\text{bearing distance } b}{\text{overhang } a}$.

The straight line indicates the contribution of the spindle deflexion x_1, to the total deflexion and the hyperbola the contribution of the bearing deformation x_2. By adding together these two curves the total displacement is obtained, and the position of the minimum determines the optimum ratio, which for this example is $b/a = 3$. Honrath found that this ratio lies generally between 3 and 5, but it may drop to 2 when the spindle has long overhang.

The stiffness of balls and rollers in anti-friction bearings corresponds to the stiffness of oil films in plain bearings and slideways. Apart from the viscosity of the oil, the oil film stiffness increases generally with higher oil pressure and decreases with increased film thickness (see also page 263).

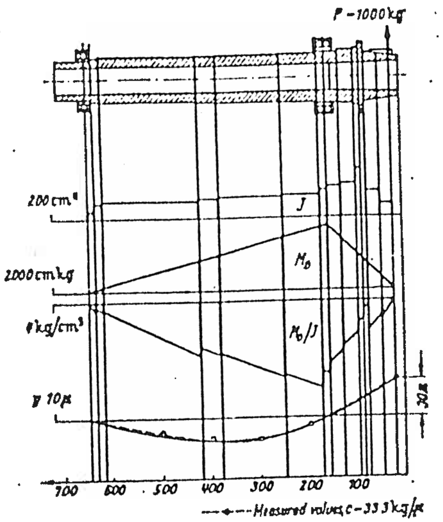


FIG. 68. Comparison of measured and graphically determined spindle deflexion (Mohr's graphical method) (from Honrath).

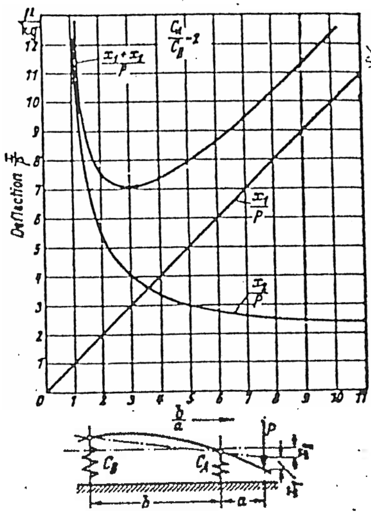


FIG. 69. Determination of optimum beaming spacing (from Honrath).

In the case of pressure lubricated bearings it is possible, therefore, to vary the stiffness of the oil film by changing the oil pressure. It must be remembered, however, that the high stiffness which can be obtained with a minimum oil film thickness has to be paid for by high manufacturing cost of the bearing surfaces and by increased cost for pump, filter, cooler, etc. because increased film thickness and increased pressure require more oil.

After the discussion which deals with super-imposing the stiffnesses of different parts of a machine, it is immediately evident that a further case is of interest. In this the point of load application varies and the relative effects of parts having different stiffnesses also change in such a manner that their influence upon the effective stiffness of the whole cannot be neglected. This can be shown for the simple example of a workpiece held between the centres of a centre lathe[11] with particular reference to the effect which the change in stiffness conditions has upon the working accuracy. The slideways for the saddle and the workpiece are assumed to be of infinite stiffness, so that only the displacements of the centres under the effect of a constant cutting force component, moving from right to left parallel to the axis of the workpiece, have to be considered (Fig. 70).

The forces supporting the workpiece at the centres are

$$P_1 = P.\left(1 - \frac{x}{l}\right)$$

and

$$P_2 = P.\frac{x}{l}$$

FIG. 70

(b) *Dynamic rigidity.* The ever-rising working speeds made possible by the development of tools and of machining processes, and the increasing requirements concerning the surface finish of machined workpieces demand machine tools which have high dynamic rigidity especially under transverse and torsional vibrations.

The properties which a machine and its elements must possess in order to reduce or prevent unfavourable vibrational effects, are determined by the types and modes of the vibrations which may occur. Impact loading, for instance, due to sudden entry of the tool into the workpiece or to the cutting edge meeting a hard spot in the material, can produce free vibrations, whilst forced vibrations may be generated by harmonic oscillating forces (e.g. unbalanced parts rotating at high speed)[12] or by non-harmonic forces (e.g. cutting forces during milling). The cutting process itself can generate self-excited vibrations[13] without additional energy being introduced from outside. The frequency of these lies near the natural frequency of the structure.

The vibration problems in a machine tool which concern the production engineer, are different from those facing the designer, because the production engineer must accept an existing machine and obtain optimum results. Tobias and Fishwick[14] have given a method for plotting stability diagrams. It is possible to determine for a particular machine with the aid of such diagrams the working conditions (cutting speed, spindle speeds, depth of cut, feed rates, etc.), which must be either applied or avoided so that no detrimental vibrations occur even if under certain conditions the machine is liable to chatter or to other vibration phenomena. The designer does not usually endeavour to determine the capacity or the weaknesses of an already existing machine. He develops new designs and must choose the design parameters in such a manner that the machine is free from any vibration phenomena over as wide as possible a working range. It is, of course, possible to experiment with existing machines and to determine the relative influences of different machine elements and of their combined effects within a dynamic vibration system which consists of tool, workpiece, machine and foundation.

The parameters which influence the vibration behaviour are:

1. The vibrating mass m
2. The static stiffness (see page 43), which can be expressed by the spring constant c
3. The damping factor ϱ
4. The natural frequency ω_0.

The deflexion δ_{stat} which occurs under a static load P is inversely proportional to the spring constant c.

$$\delta_{stat} = \frac{P}{c}$$

If an oscillating force P_{dyn} equivalent to the static load, is applied, the deflexion (the amplitude of the vibration) is increased by a magnifying factor Y.

$$\delta_{dyn} = Y.\delta_{stat}$$

$$= Y\frac{P_{dyn}}{c}$$

$$= \frac{P_{dyn}}{c/Y}$$

c/Y can be called the dynamic spring constant c_{dyn}.

Y is a function of the damping factor ϱ and the ratio $\eta = \omega/\omega_0$ between exciting frequency ω and natural frequency ω_0. When the amplitude of the exciting force is independent of the exciting frequency, then:*

$$Y_1 = \frac{1}{\sqrt{[(1-\eta^2)^2 + (2\varrho\eta)^2]}}$$

When the exciting force is generated by the unbalance of an element rotating at high speed, the amplitude depends upon the speed of rotation, i.e. the exciting frequency, and:

$$Y_2 = \frac{\eta^2}{\sqrt{[(1-\eta^2)^2 + (2\varrho\eta)^2]}}$$

The dynamic spring constant reaches a minimum when $\omega = \omega_0$ ($\eta = 1$, resonance condition).

$$c_{dyn\,min} = 2\varrho c$$

In Figs. 73a and 73b, the ratio c/c_{dyn} is plotted for different values of the damping factor ϱ and as a function of the frequency ratio $\eta = \omega/\omega_0$. Figure 73a shows the case of the exciting force amplitude being independent of the exciting frequency; Fig. 73b is applicable when the exciting frequency determines the amplitude of the exciting force (unbalance of rotating parts).

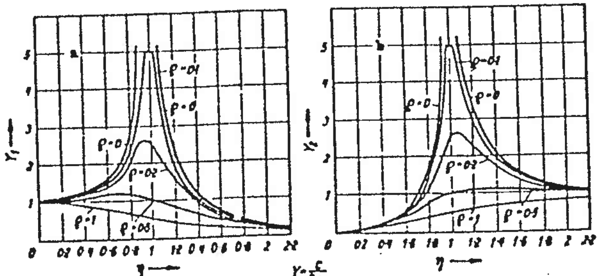


FIG. 73a and b.

It is possible to obtain high dynamic stiffness, i.e. a low value of c/c_{dyn}, by

(i) Arranging for the exciting frequency to be as far as possible either below or above the natural frequency;

(ii) Aiming at the highest possible damping.

As far as (i) is concerned, working speeds of modern machine tools vary in accordance with the required cutting conditions. As speed ranges have often to be very wide (see page 70) and top speeds rather high it would be difficult, if not impossible, to arrange for exciting frequencies to be so far above *and* below the natural frequency that ω never lies too close to ω_0. It is, therefore, safest to aim at a natural frequency which is so high that even the highest exciting frequency lies far below ω_0. The higher ω_0 in relation to ω the smaller will be $\eta = \omega/\omega_0$ and in the case of Fig. 73a, the value c/c_{dyn} approaches 1. In other words, the dynamic spring constant is not much smaller than the static stiffness. In the case of Fig. 73b, c/c_{dyn} approaches the value 0. The natural frequency is proportional to $\sqrt{(c/m)}$. This means that the natural frequency increases with growing static stiffness and with decreasing mass. High static stiffness is important for other reasons which have been discussed under (a) earlier, and if the mass is reduced, it is possible to further increase the natural frequency and with it the dynamic stiffness. This, in fact, was realized for the first time by Krug[1] and as a result he designed his grinding machines in the so-called "lightweight construction".

* For the case where the damping is proportional to the velocity. Further details and derivations of the equations mentioned will be found in text books on vibrations, and especially: S. A. TOBIAS, *Schwingungen an Werkzeugmaschinen*, Carl Hanser Verlag, Munich, 1961.

If a machine tool has a relatively small speed range and works exclusively at high speeds (for instance, a grinding machine), it is possible to work at the other side of the frequency ratio ($\eta > 1$) provided that ω is very much greater than ω_0. In this case it will be necessary to aim at a natural frequency which is as low as possible, because the exciting frequency is again determined by the working conditions and cannot be greatly influenced by the designer. It is, of course, necessary to make certain that the work speeds are not in resonance with natural frequencies which are higher than the lowest.

The permissible minimum value of static stiffness is limited by various other considerations, and in order to obtain a low natural frequency it is necessary to make the mass as large as possible. This condition is applied, for instance, in the design of "heavy" grinding machines. However, attention must be drawn to the fact that the concept of "heaviness" is often confused with those of "stiffness" and "rigidity". It is important to have a clear conception of these facts, because it is not necessary for a machine to be "heavy" in order to be "stiff" or "rigid".

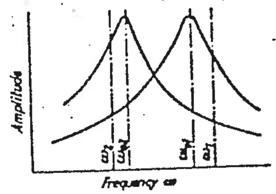


FIG. 74

Another example for relating the natural frequency to the possible exciting frequencies as a design criterion was shown by Pickenorink[15] in an investigation into the torsional vibrations of milling cutters. The natural frequency of a milling arbor can be reduced from $\omega_{0\,1}$ to $\omega_{0\,2}$ (Fig. 74) if a heavy mass is arranged on the arbor. If the exciting frequency ω_1 (rev/min of the milling cutter × number of teeth) is very high, any reduction in the natural frequency of the arbor will have a favourable effect, because the greater difference in frequency from resonance conditions will result in smaller vibration amplitudes. If, however, the exciting frequency ω_2 lies below the natural frequency, a reduction in the natural frequency will increase the danger of resonance and should be avoided.

Apart from the absolute values of forces and deformations, their relative phases are important. The exciting force precedes the deformation by an angle which depends upon the exciting frequency and the damping factor:

$$\tan\varphi = \frac{2\varrho\eta}{1-\eta^2} \quad \text{(Fig. 75)}$$

In the vector diagram (Fig. 76), the deformation δ, is drawn vertically upwards. The vector of the exciting force P precedes the deformation δ by an angle φ, and the spring force $c \times \delta$ acting against the deformation is directed downwards. The inertia force $m \times \delta$ precedes the damping force $\varrho.\delta$ by $\pi/2$ and the spring force $c \times \delta$ by π.

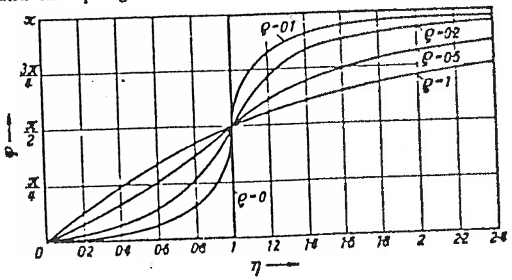


FIG. 75

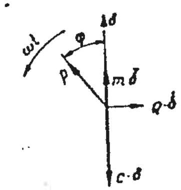


FIG. 76

For very small values of $\eta = \omega/\omega_0$, i.e. for small values of φ (see Fig. 75), inertia and damping forces are small. It depends on the value of φ (see Fig. 76), whether the exciting force is more or less in equilibrium with the spring force. A high spring force (static stiffness) in the range below the natural frequency ($\eta < 1$) is important. With increasing frequency (increasing η and increasing φ) the amplitude of the damping force grows, until in the case of resonance ($\eta = 1$), the amplitude of the damping force is equal to that of the exciting force and the amplitude of the inertia force is equal to the spring force. When resonance occurs ($\varphi = \pi/2$), the damping and exciting forces will be so to speak in equilibrium whilst at frequencies above the natural frequency, the inertia and exciting forces are approximately in equilibrium.

In order to obtain a picture of the exciting frequencies which arise in a machine, Kienzle has suggested a diagram (Fig. 77) which is similar to the spindle speed diagram (see page 82). The

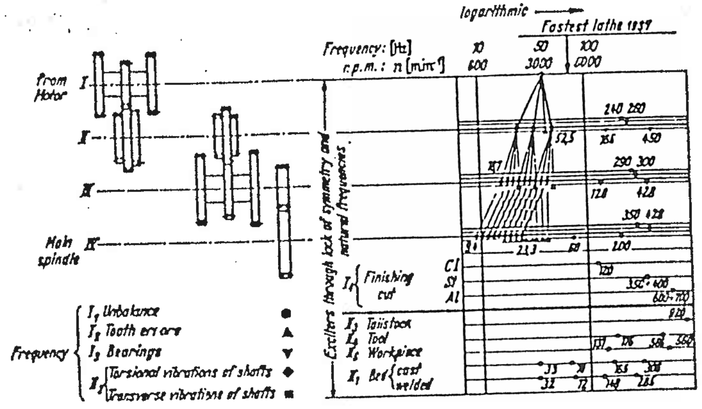


FIG. 77. Kienzle diagram.

diagram shows the exciting frequencies which are generated by the rotating parts in the drive of the machine. By careful consideration of the various natural frequencies, it is possible for the designer to avoid the danger of inadmissible vibration amplitudes and resonance conditions.

(ii) *High damping* not only influences the rapid decay of free and self-excited vibrations, but also increases the dynamic stiffness under forced vibrations (see Fig. 73). The inherent material damping in cast iron, which is usually considered to arise from the mechanical friction of fine needles of free graphite in the material[16] because the damping increases with the graphite content, is greater than that of steel. This deficiency in steel can be, however, easily more than compensated for by suitable design measures (see page 61).

From the theoretical viewpoint, the vibration problems which occur in machine tools hardly ever conform with the more easily understandable conditions of systems with one or two degrees of freedom. A purely theoretical calculation of natural frequency and damping for the case of the complex shapes of different machine tool elements is often not only difficult but also impossible.

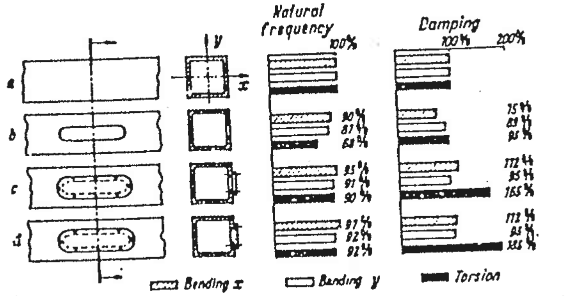


FIG. 78. Dynamic parameters of a box sectioned upright with apertures and cover plates (from Bielefeld).

No.	Sketch of beam	Weight kg	Static stiffness: Bending c_B (on axis X) kg/µ	Static stiffness: Bending c_B (on axis Y) kg/µ	Static stiffness: Torsion c_T 10^{-6} mkg/(µ/m)	Natural frequency: Bending n_B (on axis X) c.p.s.	Natural frequency: Bending n_B (on axis Y) c.p.s.	Natural frequency: Torsion n_T c.p.s.	Damping: Bending ρ_B (P = 200 kg) 10^{-4}	Damping: Bending ρ_B 10^{-4}	Damping: Torsion T = 1 mkg ρ_T 10^{-4}
Ia		42	3·2	1·6	1·0	195	135	50·5	1·12	0·58	1·38
Ib		43	3·65	1·6	1·6	209	135	54·5	0·74	0·31	0·56
Ic		47	3·65	1·6	1·6	190	128	53·5	0·73	0·47	1·07
II		38	3·0	—	1·0	196	—	50·5	0·81	—	—
III		46	3·6	1·75	1·75	194	132·5	58	0·86	0·345	0·595
IV		49	3·6	1·95	11·6	187	137·6	129·5	0·75	0·23	0·285
V		44	1·6	1·75	22·3	118	134	183	0·79	0·25	1·26
VI		44	3·1	1·85	2·9	181	134·5	70	0·63	0·24	0·89
VII		45	2·95	1·8	3·7	178	136	78·5	0·65	0·275	0·335
		10·5	0·85	0·01	0·25	200 calculated according to Rayleigh	11	41	—	—	3·0 T 0·25 mkg

FIG. 79

However, simplified considerations may help in understanding the problems which arise and in improving detailed design elements. Experiments carried out with existing machines may often help in providing the necessary information. As an example, Tlusty and Polacek[13] developed a method for recognizing the recuired design modifications in existing machines, the method being based on an evaluation of vibration tests carried out on these machines. More basic problems can be solved with the help of model experiments.[17] In both cases—model experiments or tests with existing machines—it is essential to determine, qualitatively and quantitatively, the influence of shape, size and layout upon the following.

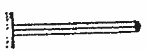
No.	Description	Sketch of beam and beam joints	Second moment of area compared with J_0	Natural frequency c.p.s.	Joint —	Increase in damping %
1	Single bar 10 mm thick		J_0	ω_0	—	—
2	Double bar not joined		$J_1 = 2J_0$	ω_0	free	0
3	Spot welded bars 1. 4 Spot welds		$J_2 = 5{\cdot}4J_0$	$1{\cdot}42\omega_0$	free	0
4	2. 2 Spot welds		$J_3 = 6{\cdot}1J_0$	$1{\cdot}6\omega_0$	touching	≈100
5	3. 8 Spot welds		$J_4 = 6{\cdot}7J_0$	$1{\cdot}75\omega_0$	touching	≈200
6	Fusion welded bars Fillet welded joint Bars only 9.5 mm thick!		$J_5 = 5{\cdot}6J_0$	$1{\cdot}7\omega_0$	free	0
7	Butt welded joint		$J_6 = 5{\cdot}2J_0$	$1{\cdot}66\omega_0$	touching	6400
8	Solid bar 20 mm thick		$J_7 = 8J_0$	$2\omega_0$	—	0

FIG. 80. Rubbing effect and second moment of area. Single and joined bars.

(1) *Static stiffness, which has an indirect influence upon the dynamic stiffness* $c_{dyn} = c_{stat}/Y$.
The influence of the design elements upon the static stiffness has already been discussed in the previous chapter (see page 43).

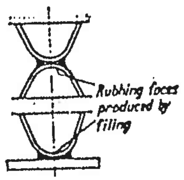


FIG. 81

(2) *Natural frequencies.*
The natural frequency is influenced by mass and stiffness. By suitable design shapes, optimum ratios between weight and stiffness can be obtained (see page 56). Bielefeld[3,9] has shown the effects which the shape and the arrangement of stiffeners have on lathe beds and the influence of apertures and cover plates in the case of box sectioned structures (Fig. 78; see also page 47).

(3) *Damping.*
Heiss[18] investigated the effect of design details, clamping and loading upon the damping of beams (Fig. 79). He has pointed out the importance of "rubbing faces"[19]

This is even more pronounced if the spindle speed range forms an arithmetic progression (Fig. 105). In this case, the lower limit v_l for the cutting speed must fall with increasing diameter if the upper limit v_u is constant. Arithmetic progression of spindle speeds, therefore, reduces the possibility of working at economic cutting speeds.

Kronenberg[29] suggested a logarithmic range of spindle speeds in which the ratio v_u/v_l is a function of the diameter. This is shown in Fig. 106 in which v_u is held constant.

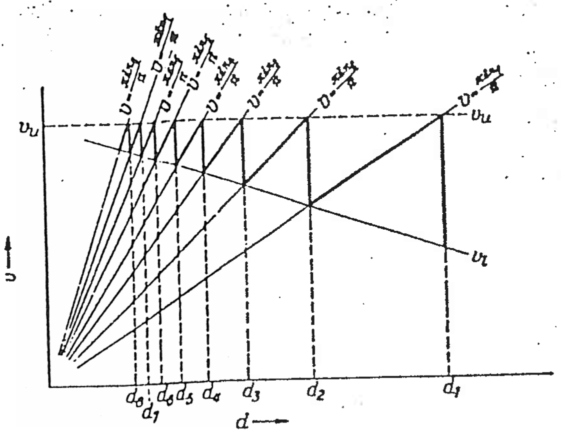


FIG. 105. Saw diagram (arithmetic progression).

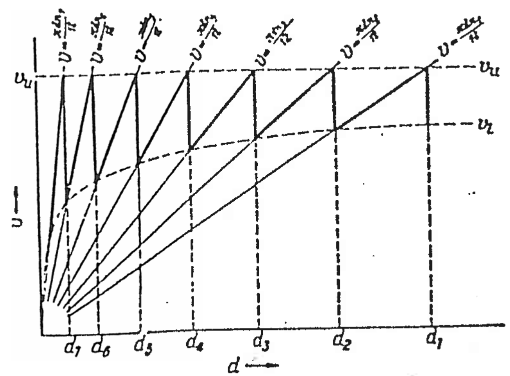


FIG. 106. Saw diagram (logarithmic progression).

The geometric progression, which is also used in the system of preferred numbers, is now generally accepted, because it is advantageous not only for reasons of easy calculation, but also from kinematic and design points of view. Some of its advantages are as follows:

(1) If from a geometric progression having the ratio φ members are removed in such a manner that only every x-th member remains, the remaining members form a new geometric progression having the ratio φ^x.

(2) By multiplying members of a geometric progression by a factor φ^y the whole series is shifted by y members.

(3) By multiplying the members of a geometric progression by a constant value *c*, a new geometric progression is obtained having the same ratio as the original one, but whose initial member is "*c*" times greater.
(4) If any two members of a geometric progression, which contains the value 1, are multiplied by each other, the products are again members of the same geometric progression.
(5) In the decimal geometrical progression, the same numerical values occur in each decimal group.

It is, therefore, possible to obtain the spindle speed of a geometric progression by arranging in series various driving elements and by providing suitable gear ratios, back gears, etc., thus always obtaining standard spindle speeds.

TABLE 11

STANDARD SPINDLE SPEEDS UNDER LOAD ACCORDING TO DIN 804

Basic Range R 20 $\varphi = 1{\cdot}12$	Range R 20/2 $\varphi = 1{\cdot}25$	Range R 20/3 $\varphi = 1{\cdot}4$			Range R 20/4 $\varphi = 1{\cdot}6$ (1400)	Range R 20/4 $\varphi = 1{\cdot}6$ (2800)	Range R 20/6 $\varphi = 2$		
100				1000					
112	112	11·2				112	11·2		
125			125						
140	140			1400	140				1400
160		16							
180	180		180			180		180	
200				2000					
224	224	22·4			224		22·4		
250			250						
280	280			2800		280			2800
315		31·5							
355	355		355		355			355	
400				4000					
450	450	45				450	45		
500			500						
560	560			5600	560				5600
630		63							
710	710		710			710		710	
800				8000					
900	900	90			900		90		
1000			1000						

The system of preferred numbers in general, and of standard spindle speeds in particular, has been described in numerous publications (a bibliography of these will be found in Kienzle).[30] The standard spindle speeds are established for full load conditions, so that in general, machine tool spindles and other shafts run slightly faster than the nominal values of the speed tables. This is important from the point of view of piece-work time calculations, because the operator cannot incur a loss due to a machine spindle running more slowly than originally estimated by the rate fixer. The motor shaft speeds of a.c. synchronous motors under full load are contained in the standard speed ranges, in order to make possible the direct drive through a rotor incorporating the spindle. In order to allow also for speed changes due to pole changes of motors, the ratio 2 must be available. As $\sqrt[3]{2} \simeq \sqrt[10]{10} \simeq 1{\cdot}25$, this condition is also satisfied in the normal range (*R20/2*, $\varphi = \sqrt[10]{10}$). This range has 10 steps in each decimal group and is considered the main spindle-speed range. Standard steps and spindle speeds are shown in Table 11. It will be noticed that for

the step $\varphi = 1{\cdot}6$ (*R20/4*), two ranges are provided, one of which contains a speed under load of 2800 rev/min, and the other a speed under load of 1400 rev/min, thus catering for a.c. motors with two or four poles.

The feed rate can be referred either to the spindle speed (in/rev), as in the case of turning, drilling or boring, or—independently of the spindle speed—to time (in/min), as in milling. In order to determine the time required for turning or drilling a certain length, it is necessary to know the feed rate per minute, which is equal to the product of spindle speed (n) and feed rate (s) per revolution of the spindle. If not only values for n but also values for s are standardized as in the preceding paragraphs, their products, i.e. the feed rates per minute, become standard values (Table 12). It is also possible to use standard speeds (rev/min) for the rotating parts in feed-drive mechanisms, as

TABLE 12

STANDARD FEEDS ACCORDING TO DIN 803 (MILLIMETRES)

$\varphi = 1{\cdot}12$	$\varphi = 1{\cdot}25$	$\varphi = 1{\cdot}4$			$\varphi = 1{\cdot}6$	$\varphi = 2$		
1	1		1		1		1	
1·12				11·2				
1·25	1·25	0·125				0·125		
1·4			1·4					
1·6	1·6			16	1·6			16
1·8		0·18						
2	2		2				2	
2·24				22·4				
2·5	2·5	0·25			2·5	0.25		
2·8			2·8					
3·15	3·15			31·5				31·5
3·55		0·355						
4	4		4		4		4	
4·5				45				
5	5	0·5				0·5		
5·6			5·6					
6·3	6·3			63	6·3			53
7·1		0·71						
8	8		8				8	
9				90				
10	10	1			10	1		

long as their dimensions (pitch of lead screws, diameter and number of teeth of feed drive pinions, etc.), are selected in compliance with these speeds.

Before the designer can lay out a gearbox which will provide the required spindle speeds and feed rates, it is necessary first of all to determine the speed range, the number of steps and the ratio between the steps. For this purpose, not only technical but also economic considerations must be borne in mind. If the variety of materials which have to be machined and the range of working diameters to be covered require a large speed range (e.g. in the case of radial drilling machines, see page 70), and if it is essential to obtain speeds which are as close as possible to the theoretically required values, very fine steps (close speed tolerances between v_u and v_l, see page 70) must be provided, and these result, of course, in a large number of steps, and considerable cost of the gearbox. It is not always necessary, however, to choose the limits v_u and v_l very close together, especially if the feed rate can be set independently of the spindle speed, and if the machining time is, therefore, independent of the spindle speed (milling machine). In such cases, a relatively coarse stepping of the spindle speeds may suffice. However, the feed drive must be more finely stepped.

On the other hand, it may be necessary to reduce the diameter range, or the variety of materials which are to be machined, if it is essential that optimum speeds be obtainable within fine limits. It is then possible to provide a finely stepped range without too large a number of steps. The relations between speed-range ratio B, step ratio φ and number of steps z for gear drives have been established once and for all by the speed standardization. They are:

$$B = \frac{n_{max}}{n_{min}}; \quad n_{max} = n_{min} \times \varphi^{(z-1)}; \quad \varphi^{(z-1)} = \frac{n_{max}}{n_{min}} = B$$

Through a graphic representation (Fig. 107), it is possible for the designer to view these relations quickly and clearly and to weigh up various solutions at his disposal.

With $(z-1).\log\varphi = \log B$ and $z = 1 + \log B/\log\varphi$, z becomes a linear function of B for each standard step ratio φ, if the speed-range ratio is plotted logarithmically as the abscissa and the number of steps linearly as the ordinate.

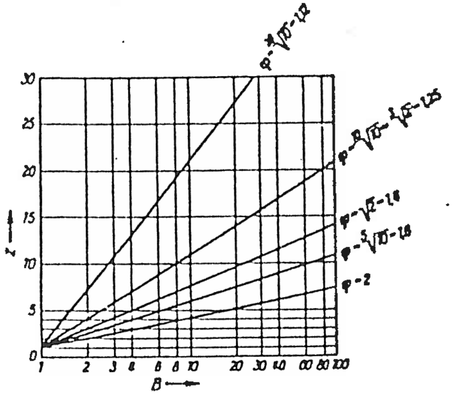


Fig. 107. Relation between speed-range ratio B, number of steps z at standard ratios φ.

After the speed range, number of steps and step ratio of a gear drive, have been established by the designer, it is still not possible to develop the actual layout, determine the load carrying capacities and the dimensions of the wheels, calculate the space required and design the control mechanism until consideration has been given to the following:*

(1) Establishment of gear ratios,

(2) Layout of the intermediate reduction gears,

(3) Calculation of transmission ratios and pulley diameters or numbers of teeth.

These will now be discussed in detail.

(1) *Establishment of gear ratios.* The basic unit of a speed change device, is the two-axes drive (Figs. 108–112). Between the input shaft (axis I) and the output shaft (axis II), different trans-

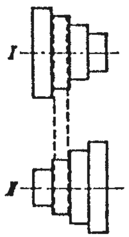


Fig. 108

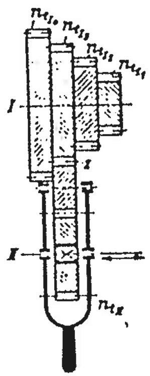


Fig. 109

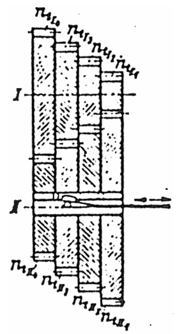


Fig. 110

mission ratios can be engaged so that at constant speed of the input shaft, specific required speeds of the output shaft are produced. The torque can be transmitted either by means of belts or gears.

* In this connexion, a Paper by H. Schöpke is interesting, entitled Stufengetriebe von Werkzeugmaschinen. *Industrie-Anz.*, 22. February, 1957.

When using belts, cone pulleys (Fig. 108) can be fitted. For gearing, the use of slip gears (i.e. interchangeable gears which can be fitted to each shaft by hand as required) is the simplest solution of the problem. This method is, however, time-consuming and is only applied if the speed of the output shaft can remain unaltered for a long time, so that the setting-up time does not represent a great loss. This is the case when, for example, special-purpose machines are working in quantity production.

For universal machines which are used for various operations during the machining of small quantities of workpieces, the spindle speeds and feed rates have to be changed rather frequently, and in this case for feed drives Norton type (Fig. 109), and draw-key type (Fig. 110) gearboxes are used whilst for both cutting and feed drives the clutch type (Fig. 111) or sliding gear boxes (Fig. 112), are in more general use.

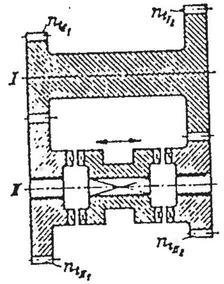


Fig. 111

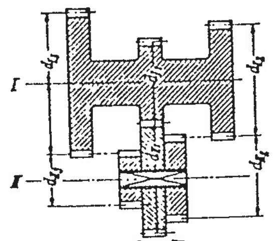


Fig. 112

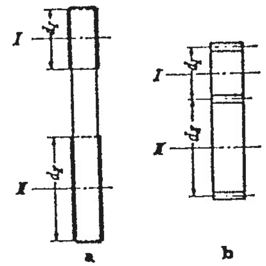


Fig. 113

In the case of all two-axes drives, the ratio between the speeds of the output and the input shaft is inversely proportional to the diameter of the corresponding driving elements (pulleys, chain sprockets, gears), so that:

$$\frac{n_{II}}{n_I} = \frac{d_I}{d_{II}} \quad \text{(Fig. 113)}$$

In the case of several pairs of elements, 1, 2, 3 (cone pulleys, gear blocks, etc., see Figs. 108–112), the transmission ratios are:

$$u_1 = \frac{n_{II_1}}{n_I} = \frac{d_{I_1}}{d_{II_1}}$$

$$u_2 = \frac{n_{II_2}}{n_I} = \frac{d_{I_2}}{d_{II_2}}$$

$$u_3 = \frac{n_{II_3}}{n_I} = \frac{d_{I_3}}{d_{II_3}}$$

$$u_{n-1} = \frac{n_{II_{n-1}}}{n_I} = \frac{d_{I_{n-1}}}{d_{II_{n-1}}}$$

$$u_n = \frac{n_{II_n}}{n_I} = \frac{d_{I_n}}{d_{II_n}}$$

When the speeds of both the driving and the driven shaft are parts of a standard range, then:

$$u_n = \frac{n_{II_n}}{n_I} = \varphi_N^x$$

Table 13

Gear Combinations which Produce Transmission Ratios and Therefore Output Speeds which Lie Within the Limits Allowed by DIN 804, and with which the Permissible Tolerances are Not Exceeded*

Transmission Ratio 1:	$1{\cdot}12^0 = 1$	$1{\cdot}12^1 = 1{\cdot}12$	$1{\cdot}12^2 = 1{\cdot}25$	$1{\cdot}12^3 = 1{\cdot}4$	$1{\cdot}12^4 = 1{\cdot}6$	$1{\cdot}12^5 = 1{\cdot}8$	$1{\cdot}12^6 = 2{\cdot}0$	$1{\cdot}12^7 = 2{\cdot}24$	$1{\cdot}12^8 = 2{\cdot}5$	$1{\cdot}12^9 = 2{\cdot}8$	$1{\cdot}12^{10} = 3{\cdot}15$	$1{\cdot}12^{11} = 3{\cdot}55$	$1{\cdot}12^{12} = 4{\cdot}0$
No. of Teeth of the Pinion	Number of Teeth of the Wheel												
16 :	16	18	20	23	25	28	32	36	40	45	50 51	56 57 58	63 64
17 :	17	19	21	24	27	30	34	38	42 43	48	53 54	60 61	67 68
18 :	18	20	23	25	29	32	36	40 41	45 46	50 31	56 57 58	63 64 65	71 72
19 :	19	21	24	27	30	34	38	42 43	47 48	53 54	59 60 61	67 68	75 76 77
20 :	20	22	25	28	32	36	40	44 45	50 51	56 57	62 63 64	70 71 72	79 80 81
21 :	21	24	26	30	33	37	42	47	52 53	58 59 60	66 67	74 75 76	83 84 85
22 :	22	25	28	31	35	39	44	49 50	55 56	61 62 63	69 70	77 78 79	88 89
23 :	23	26	29	32 33	36 37	41	46	51 52	57 58	64 65 66	72 73 74	81 82 83	91 92 93
24 :	24	27	30	34	38	42 43	48	53 54	59 60 61	67 68 69	75 76 77	84 85 86 87	94 95 96 97

* From E. Stephan: *Optimale Stufenrädergetriebe für Werkzeugmaschinen.* Berlin/Göttingen/Heidelberg: Springer 1958.

Table 14

Numbers of Teeth and Sums of Numbers of Teeth for Standard Transmission Ratios
(For the sake of this example, the table covers the sums of numbers of teeth from 100 to 109)*

Transmission Ratio 1:	1	1·12	1·25	1·4	1·6	1·8	2·0	2·24	2·5	2·8	3·15	3·55	4·0
Sum of Numbers of Teeth	Numbers of Teeth — Pinion : Wheel												
100	50 : 50	47 : 53	44 : 56		39 : 61	36 : 64		31 : 69		26 : 74	24 : 76	22 : 78	20 : 80
101	51 : 50		45 : 56	42 : 59	39 : 62	36 : 65	34 : 67	31 : 70	29 : 72		24 : 77	22 : 79	20 : 81
102	51 : 51	48 : 54	45 : 57	42 : 60		37 : 65	34 : 68		29 : 73	27 : 75			
103	52 : 51			43 : 60	40 : 63	37 : 66		32 : 71		27 : 76	25 : 78		21 : 82
104	52 : 52	49 : 55	46 : 58	43 : 61	40 : 64		35 : 69	32 : 72		27 : 77	25 : 79	23 : 81	21 : 83
105	53 : 52				41 : 64	38 : 67	35 : 70		30 : 75		25 : 80	23 : 82	21 : 84
106	53 : 53	50 : 56	47 : 59	44 : 62	41 : 65	38 : 68		33 : 73	30 : 76	28 : 78		23 : 83	21 : 85
107	54 : 53		47 : 60	44 : 63	41 : 66		36 : 71	33 : 74		28 : 79	26 : 81		
108	54 : 54	51 : 57	48 : 60	45 : 63	42 : 66	39 : 69	36 : 72	33 : 75	31 : 77	28 : 80	26 : 82	24 : 84	22 : 86
109	55 : 54	51 : 58	48 : 61	45 : 64	42 : 67	39 : 70		34 : 75	31 : 78		26 : 83	24 : 85	22 : 87

* From R. Germar: *Die Getriebe für Normdrehzahlen.* Berlin: Springer 1932.

For the 12-speed gearbox, the following speeds ($\varphi_{12} = 1{\cdot}4$) may be chosen (see Table 11):

31·5, 45, 63, 90, 125, 180, 250, 355, 500, 710, 1000, 1400

and for the 18-speed gear box ($\varphi_{18} = 1{\cdot}25$):

35·5, 45, 56, 71, 90, 112, 140, 180, 224, 280, 355, 450, 560, 710, 900, 1120, 1400, 1800.

The design of the gearbox is shown in Fig. 124. The reversing drive between shafts *I*, *II* and *III* is obtained by means of the three-gear block on shaft *III*, in such a manner that the largest gear (34 teeth) drives the spindle forward and the smaller gear (30 teeth, difference in the number of

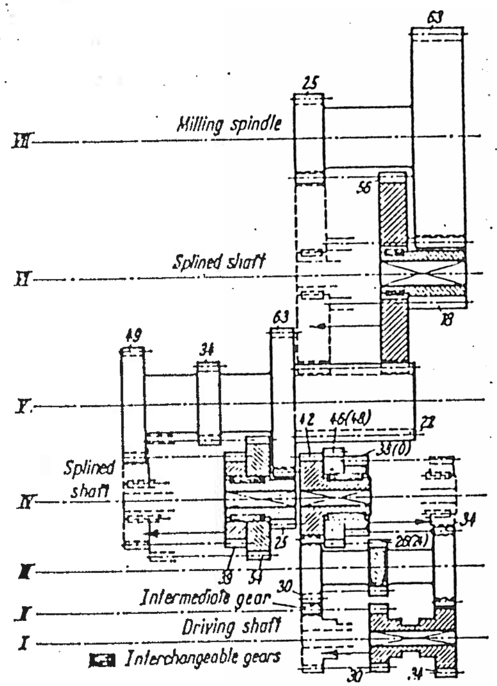


FIG. 124. Milling spindle drive for 12 or 18 spindle speeds.

teeth 4, see page 86), drives the spindle in reverse by means of the intermediate gear on shaft *II* (both transmission ratios 1 : 1). In order to reduce the number of relatively high-cost splined shafts to a minimum, the two sliding-gear blocks for the first (shafts *III* and *IV*) and second (shafts *IV* and *V*) part drive are arranged on shaft *IV*. The speed level of shaft *V* is relatively high so that for a given power smaller torques have to be transmitted between shafts *I* and *V*. A constant reduction between shafts *V* and *VI* lowers the speed level sufficiently for the required output speeds of the milling spindle (shaft *VII*) to be obtainable by means of a third part drive (shafts *VI* and *VII*) without the final reduction being greater than 1 : 4. The gear ratio for the higher speeds 2·24 : 1 is a little higher than the value 2 : 1 previously quoted (see page 77). However, the diameter of the larger gear is still smaller than that of the largest gear wheel on the milling spindle, so that the space required is readily available.

The minimum permissible centre distance between shafts IV and V is, therefore:

$$\frac{d_2 + d_4}{2} = \frac{\quad}{2} =$$

and is, therefore, less than the existing centre distance of

Finally, the exact values of the output speeds of the two gearboxes must be calculated and compared with the exact values of the standard speed ranges (Table 15). Most of the deviations of the actually obtainable speeds from the standard ones are negative. In contrast to the case of the lathe, the productivity of the milling machine is independent of the spindle speed (see page 74). It is, therefore, advantageous to avoid positive deviations, because if the cutting speed is slightly lower than the recommended one the tool life will be increased. In only three cases is the permissible error of two per cent very slightly exceeded.

When the Germar tables are used there is a danger that the actual output speed may be outside the permissible limits. Although the values given by Germar result in transmission ratios which are within $\pm 1{\cdot}5$ per cent, or in the case of very high ratios within ± 2 per cent of the standard ratios, the arrangement in series of several gears may result in excessive inaccuracies of the final output speeds. A set of tables developed by Jackel[32] is designed less for obtaining standard ratios within the part drives than for reducing the deviations of the output speeds to a minimum. They make it possible to compensate deviations in one part drive by corresponding deviations in the next one. Such problems are discussed in detail in the book by Stephan.[31]

3. ELECTRICAL, MECHANICAL AND HYDRAULIC DRIVES FOR THE OPERATIONAL MOVEMENTS

In general, the electric motor serves for driving the various operational movements of a machine tool. In the arrangement which was used frequently in the past where a relatively large motor drives a battery of machines via line shafts and belts, the advantage lies in the fact that the motor is subjected to a relatively even load that is near its maximum capacity, and therefore, works at a high efficiency. This advantage is, however, more than offset by the inevitable disadvantages, such as limitations in the possible arrangement of the machine tools, long distance energy transmission with loss in mechanical efficiency, etc. For this reason, the single motor drive has been developed to such an extent that in many applications not only does one motor drive one machine, but also separate motors, suitably interlocked and controlled, are employed for driving different operational movements of one machine. In this manner, the length of mechanical elements for transmitting power and control movements can be considerably reduced.

The energy provided by the electric motor must be transmitted to the driving elements and the tool and workpiece carriers in such a manner that at any given time the rotational movement of the motor shaft is transformed into operational movements of the required type, direction and speed. For this purpose, the machine tool designer has at his disposal drives and driving elements which may be:

I. electrical,

II. 1. mechanical, and

2. hydraulic.

If the operational movements are rotational, the length of the energy transmission can be reduced to a minimum by arranging for the energy to be transmitted directly from the motor on to the machine spindle. Frequently, however, mechanical or hydraulic elements are inserted between the motor and the tool or workpiece carrier. For I, the operational movements have to be controlled electrically by means of appropriate equipment, whilst for II, electrical, mechanical, hydraulic, hydromechanical, electromechanical or electrohydraulic control equipment is being used.

I. Electric Drive and Control Equipment

The performance specifications of the driving motors are influenced not only by the operational conditions but also by the requirements of controlling the machine tools. If a clutch is arranged between motor and main gearbox, the motor can start under practically zero load. If, on the other hand, no mechanical clutch separates the motor from the various machine elements which have to be accelerated, such as spindle, table, etc., and if the operational movements of the machine are controlled merely by switching the driving motor on and off, then the inertia forces caused by accelerating the moving parts and the idling resistances in the gearboxes must be overcome. Under these conditions the motor has to start under load and needs a high starting torque. For the type of operational movement which has to be produced, and depending upon the transmission which is arranged between motor and tool or workpiece carrier, the driving motor may have to run:

(i) With constant speed
  (a) in one direction
  (b) forward and reverse.

(ii) With two constant speeds, forward and reverse, i.e. slow speed forward and fast speed reverse, or vice versa.

(iii) With a stepped or infinitely variable speed which may be set either before the start or be adjustable during running
  (a) forward or reverse
  (b) forward and reverse.

Apart from these specifications concerning power, torque, speed and direction of movement, the design of the motor must satisfy the working conditions (open, protected, splash-proof, dust-proof, etc.), and the possibilities of connecting it to the machine tool (flange motors, built-in motors, etc.). For economic reasons the machine tool designer will endeavour to use standard motors as far as possible. It is often advisable, therefore, to provide suitable adaptor pieces which facilitate the use of standard motors under varying conditions of application.

As far as the designer is concerned, the motors which are generally used can be classified in accordance with the following:[33]

(1) Starting characteristic
(2) Behaviour during running
(3) Power and torque characteristics as a function of speed
(4) Speed adjustment
(5) Possibility of braking (rapid braking, inching, etc.)
(6) Efficiency (electrical or mechanical) and power factor ($\cos \varphi$)
(7) Freedom from vibrations.

The average workshop is usually equipped with three-phase a.c. supply for reasons of simplicity and reliability of the motors. Three-phase a.c. motors are, therefore, installed either for driving the machine tools directly, or for driving Ward-Leonard sets (see page 98), and in this latter case the machine tools are driven by d.c. motors.

Amongst the three-phase a.c. motors available the squirrel-cage motor is widely used. The starting torque is about 60 to 100 per cent higher than the nominal torque, and when running the torque increases with speed until a maximum is reached, after which it drops rapidly. The nominal torque is usually reached at about 94 to 97 per cent of the synchronous speed. If the load exceeds the nominal torque the speed will drop, and if the maximum torque is exceeded the motor stalls. If such motors are switched on directly the current will rise to above five to seven times the nominal value. In many cases, this may be permissible. If the starting current is inadmissibly high, star-delta starters are used, which are controlled either manually or automatically. By inserting a resistance into one phase during the starting period and automatically short-circuiting this by means of a relay as soon as the nominal speed is reached, the starting torque of the motor can be reduced and smooth starting obtained.

In the case of slip-ring motors, the terminals of the rotor winding are connected over slip rings with an external adjustable resistance. This is made operative during the starting of the motor, in order to increase the rotor resistance, and is short-circuited when the nominal speed is reached, thus keeping the starting current and the starting torque within permissible limits. The use of eddy-current rotors results in higher starting torques and lower starting currents, and motors thus

equipped are particularly suitable for starting under load. Motors using a rotor resistance, due to their higher starting torque and low starting current, are especially suitable for frequent starting and stopping conditions.

The direction of rotation can be reversed by exchanging two connexions. The synchronous speed n_0 depends upon the supply frequency f_0 and the number of poles p:

$$n_0 = \frac{60 . f_0}{p/2} \qquad (n_0 \text{ in rev/min}; f_0 \text{ in c/sec}).$$

By changing the number of active poles (2, 4, 6, 8 or 12), through suitable switching devices, it is possible to obtain, with a supply frequency $f_0 = 50$ c/sec, synchronous speeds of 3000, 1500, 1000, 750 or 500 rev/min. The corresponding standard speeds under load are then 2800, 1400, 900, 710 and 450 rev/min.

If the driving motor has to be directly connected with the main spindle of the machine tool, i.e. without an intermediate mechanical or hydraulic transmission, and if spindle speeds of more than 3000 rev/min are required, high frequency motors are used. Their speed can be varied by means of frequency variation.[34] A standard (50 c/sec) a.c. motor is rigidly coupled with a 3-phase a.c. generator, the stator winding of which is supplied with the normal mains frequency of 50 c/sec. The speed of the motor (n_m) and the generator depends, therefore, upon the number of poles (p_m) of the motor:

$$n_m = \frac{50}{p_m/2} = \frac{100}{p_m} \qquad [\text{rev/sec}]$$

The frequency f produced by the generator depends upon the generator speed, the number of poles of the generator (p_g) and the direction of the rotating field in the stator winding ($f_0 = 50$ c/sec), relative to the direction of rotation of the motor. (If it is in the opposite direction, it is positive, and if it is in the same direction it is negative).

$$f = n_m . \frac{p_g}{2} \pm f_0 = 50\frac{p_g}{p_m} \pm 50 = 50\left(\frac{p_g}{p_m} \pm 1\right)$$

The speed n of any motor (number of poles p) supplied from the generator is

$$n = \frac{f}{p/2} = \frac{2f}{p} = \frac{100}{p}\left(\frac{p_g}{p_m} \pm 1\right) \qquad [\text{rev/sec}]$$

With the exception of the speeds in the lower range, the same speed range can be obtained when generator rotor and rotating field are rotating in the same direction or in opposite directions, so that rotation in the same direction need not be considered.[30] Therefore:

$$n = \frac{6000}{p}\left(\frac{p_g}{p_m} + 1\right) \qquad [\text{rev/min}] \qquad \text{(Table 16).}$$

The safety of the operator and the requirements of the working conditions make it necessary in many applications to provide for the motor to be stopped rapidly and safely. When mechanical friction is used, for instance, by solenoid-operated brakes, the brake materials are subject to wear. This can be avoided by applying an opposing current, e.g. by reversing two connexions or by appropriate switching through which the stator winding is suitably excited. In the case of reverse current braking, it is necessary to provide a circuit which prevents the motor from starting to run in the opposite direction.

Efficiency and power factor of 3-phase a.c. motors increase with the ratio $\frac{\text{actual load}}{\text{nominal load}}$. Whilst, however, the efficiency changes only slightly, even if the actual load drops to about 40 per cent of the nominal value, the power factor drops relatively fast, when the load is only slightly less than the nominal one.

Speed adjustment within very fine limits is possible with d.c. shunt wound motors. If these motors are started directly, a peak current occurs which is not permissible except in the case of very small motors. For this reason, a starting resistance in series with the rotor winding is used for larger motors, and this limits the peak value of the starting current. The size and type of the starter depends upon the requirements of the motor (starting under load, short or long starting period, etc.).

With increasing torque, the set speed of a shunt-wound motor drops slightly, the drop being far greater in the case of smaller motors than for larger ones.

TABLE 16

Number of Poles: Driving Motor	Number of Poles: Generator	High Frequency	Motor Driving the Machine Tool: Number of Poles	Motor Driving the Machine Tool: Synchronous Speed	Motor Driving the Machine Tool: Speed under Load
p_m	p_g	f	p	n	n_L*
		Hz		rev/min	rev/min
2	2	100	2	6000	5300
	4	150		9000	8000
	6	200		12000	10600
	8	250		15000	13200
	10	300		18000	16000
	12	350		21000	18000
	14	400		24000	21200
	16	450		27000	23600
4	2	75	2	4500	4000
	4	100		6000	5300
	6	125		7500	6700
	8	150		9000	8000
	10	175		10500	9500
	12	200		12000	10600
	14	225		13500	11800
	16	250		15000	13200

* At double slip; see O. KIENZLE: Normungszahlen.

The speed can be adjusted by varying either the rotor current or the field current. Insertion of a resistance in the rotor circuit results in a lower rotor current and reduced speed. This adjustment is, however, obtained at the expense of the efficiency, because part of the energy supplied to the motor is transformed into heat in the inserted resistance. By changing the rotor voltage (see Ward-Leonard set), it is possible to vary the speed at constant torque, i.e. the power increases or decreases with the speed (Fig. 127). By weakening the field (shunt adjustment), it is possible to increase the speed practically without losses, as the power remains constant (Fig. 128). Such an arrangement enables a speed range of about 3 : 1 to be obtained economically.

A wider speed range can be obtained with a Ward-Leonard set (Fig. 129). A motor A drives a d.c. generator B and an exciter C, which provide the supply for the motor D driving the machine tool. By varying the resistance E, the terminal voltage of the generator B and, therefore, the voltage of rotor D can be adjusted between zero and a maximum value, and the speed of the motor can, therefore, be varied continuously within wide limits and at constant torque.

If, in addition, the field is weakened (resistance F), it is possible to increase the speed of motor D even further, at constant output power. This results in a speed range of up to 20 : 1 with continuous control over the working range (Fig. 130). However, the field weakening has the disadvantage of giving low torque at high speeds. Moreover, the maximum permissible speed may also be limited by the centrifugal force acting upon the rotor winding.

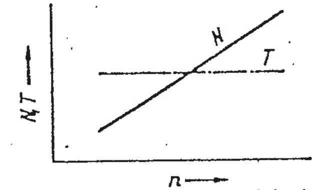


FIG. 127. Speed adjustment of the d.c. shunt motor by changing the rotor voltage.

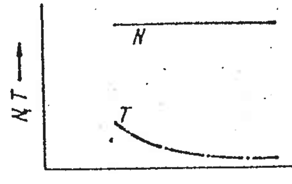


FIG. 128. Speed adjustment of the d.c. shunt motor by changing the field current.

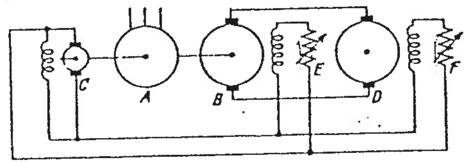


FIG. 129. Circuit of a Ward-Leonard set.

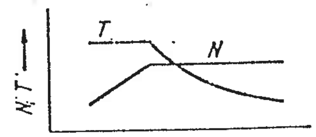


FIG. 130. Speed adjustment of the Ward-Leonard set.

The minimum speed may be limited by the cooling requirements of the motor, since below a certain motor speed the cooling effect of the fan may be insufficient. In the case of very low speeds, stick-slip phenomena may also cause irregularity of rotation, and it may be necessary to provide control equipment which serves not only for varying the motor speed, but also for guaranteeing that the speed, once set, remains constant.[35]

Reversing of shunt-wound motors is usually obtained by reversing the current in the rotor. This can be achieved in the Ward-Leonard set by reversing the current in the generator field. Apart from solenoid-operated mechanical brakes (see page 97), it is also possible to brake the motor by making it act as a generator through suitable switching. The motor will then run until its momentum is absorbed. Another means of braking consists in switching off and short-circuiting the rotor through a resistance.

The efficiency of d.c. shunt-wound motors is reasonably good even at low loads (down to 30 per cent of the nominal load).

The design of manually or relay operated control equipment for starting, stopping, speed adjustment and reversing of motors will not be discussed here as these problems do not really concern the machine tool designer. However, the designer must consider the arrangement and the interrelation of such equipment, especially in cases where electrical or mechanical devices work together as parts of a large control system in a machine tool.[36] Problems of this nature become of paramount importance in fully automatic controls, which are to be discussed in a separate chapter (see page 161). However, some of the problems are encountered in manually or semi-automatically operated machines, and the following examples may be cited.

One of the advantages of electrical controls lies in the fact that levers, gears and shafts which transmit movements and forces and in which relatively large mechanical losses are often unavoidable, are replaced by current conductors which have no mass and can easily bridge long distances.

The centralized arrangement with a single driving motor for various elements necessitating the engagement and disengagement of clutches for starting and stopping, is replaced by a separate

The power transmission from an electric motor to the machine tool can be coaxial, i.e. by rigid or elastic coupling or by means of a clutch, or two-axial, by means of a belt, chain or gear drive. The choice will often depend upon the various possible ways of arranging the motor and upon the most economic motor speed. In the case of high spindle speeds, the coaxial direct drive simplifies the design considerably. In the case of relatively low spindle speeds, a speed reduction from a high-speed motor to the driving shaft is preferable, as low-speed motors are generally expensive. If the motor cannot provide the torque necessary for starting the machine, a clutch has to be inserted between the driving shaft in the machine and the reduction gearbox. Otherwise, a direct drive using an elastic (especially if alignment between motor and gearbox input shaft is difficult or if impact loads are likely to occur) or a rigid coupling can be provided. In special cases, the rotor of the electric motor can be mounted directly on the driven shaft (see).

The elastic transmission of impact loads is possible not only with elastic couplings, but also with a belt drive. If the centre distance between motor and input shaft of the machine is small, and especially if high torques have to be transmitted without slip, a positive chain drive may be the best solution. Although this is more expensive than a belt drive, it has a higher efficiency and can work at lower circumferential speeds. Between the flat belt and the chain drive lies the Vee-belt drive with which larger power can be transmitted and which shows a certain elasticity in transmitting impact loads, although its slip is negligibly small. Smooth driving conditions without vibrations can be obtained with Vee-belts, endless silk or nylon belts. The latter are often used in grinding machine drives. Moreover, whilst chain drives require suitable provision for lubrication, the Vee-belt drive does not need this, and it is, therefore, today perhaps the most frequently used energy transmitting element between motor and machine tool.

II. Mechanical and Hydraulic Drives

Mechanical and hydraulic drives can be divided into two groups. These are: (1) Drives for producing rotating movements, and (2) Drives for producing rectilinear movements. The first group includes devices which transform the rotation of an input shaft driven by an electric motor into rotation of an output shaft (main spindle, cam shaft, etc.), at the required speed and in the desired direction. The devices of the second group transform a rotational input movement, usually produced by a drive of the previous group, into a straight line reciprocating movement of a table, ram, etc.

(a) *Drives for Producing Rotational Movements*

Stepped drives. The stepped speed range (see page 71) of the output shaft can be produced by various types of mechanical devices.

Slip gears can be fitted to shafts with fixed centre distances if the total number of teeth of all meshing gears is constant. Such slip gears are used in machines which necessitate few speed changes and only a limited number of different rotational speeds or in cases where the accuracy of the output speed is not critical. This applies, for instance, in the case of many single purpose and special machines.

If a large number of very finely stepped rotational speeds is required and if the actual speed values must be accurate within fine limits (for instance, in drives for lead screws and dividing heads), slip gears (Fig. 143) are arranged in such a manner that between the fixed axes of the driving (*I*) and the driven (*III*) shaft, an adjustable intermediate shaft (*II*) is provided. The position of the latter on a carrier (quadrant *a*) can be varied by either linear displacement or by swivelling the carrier round axis *III*. Shaft *II* serves not only for carrying intermediate gears *2*, *3* and thus for obtaining large reduction ratios, but also for compensating the differences of centre distances between wheels of different sizes. Sets of slip gears are usually so chosen that practically all required

transmission ratios can be obtained. The quadrant *a* has to cover the full range of slip gear combinations. In other words, the radial slot *b* must be of a length which corresponds to the required maximum and minimum distances between axes *II* and *III* of wheels *3* and *4*, and the quadrant *a* must be able to swivel through an angle which corresponds to the required maximum and minimum distances between axes *I* and *II* of wheels *1* and *2*. This angle is limited by the length of the circular slot *c*. The axle peg *d* for wheels *2* and *3* is clamped in the radial slot *b* in a position which is determined by the centre distance between wheels *3* and *4*, and the centre distance between wheels *1* and *2* is then adjusted by swivelling the quadrant around the axis *III* of the wheel *4*. After adjustment, the quadrant *a* is clamped in position by stud *e*, which is fixed to the bed of the machine, the circular slot *c* riding over this peg.

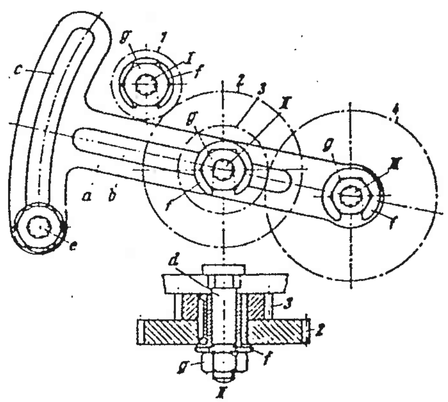


FIG. 143. Slip gear quadrant.

If the transmission ratio produced by the slip gears

$$i = \frac{n_{t_1}}{n_{t_2}} \cdot \frac{n_{t_3}}{n_{t_4}}$$

serves for cutting an inch screw thread with a metric lead screw or vice versa, a so-called conversion gear with 127 teeth (5 in. = 127 mm) is used. If, for instance, a metric thread with 1 mm pitch is to be cut with a lead screw of ½ in. (12.7 mm) pitch, the transmission ratio between the main spindle (*I*, Fig. 143) and the lead screw (*III*, Fig. 143), must be

$$i = \frac{n_{t_1}}{n_{t_2}} \cdot \frac{n_{t_3}}{n_{t_4}} = \frac{1}{12{\cdot}7} = \frac{10}{127} = \frac{20}{50} \cdot \frac{25}{127}$$

In other words,

$$n_{t_1} = 20$$

$$n_{t_2} = 50$$

$$n_{t_3} = 25 \quad \text{and} \quad n_{t_4} = 127 \quad \text{(the conversion gear).}$$

The time required for changing these gears can be considerably reduced if the diameters of the screwed ends of peg *II* and shafts *I* and *III*, which serve for clamping the gears by means of washers and nuts, are chosen in such a manner that the dimension across corners of the clamping nut is smaller than the bore of the slip gears, and if C-shaped washers (*f*) are used instead of standard washers (Fig. 143). In this case, there is no need to remove the small clamping nut *g* completely.

The change gears can be withdrawn over the clamping nuts after these have been slightly loosened and the C-shaped washers laterally removed.

Advantages of the *Norton-type gearbox* (Fig. 144) are the compact arrangement of the gear

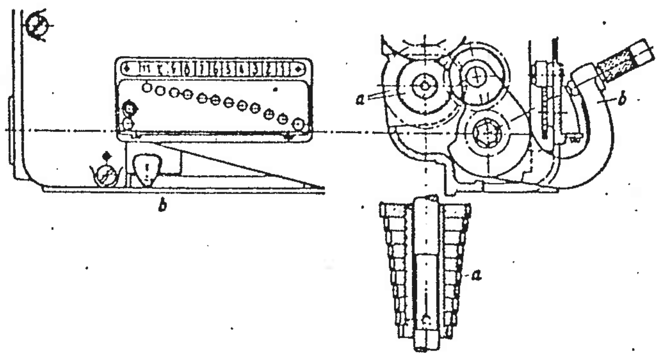


Fig. 144. Norton-type gearbox.

block *a*, the possibility of very fine stepping (the stepping of the numbers of the gear teeth is equal to the stepping of the output speeds, see page 84) and the fact that only those gears are in mesh which are actually required to transmit the torque. However, the unavoidable weakness of these gearboxes is the lever *b* carrying the intermediate gear, and for this reason, Norton-type gearboxes are used only for low power transmission (for instance, feed drives in centre lathes).

An interesting development is shown in Fig. 145,* in which special gear profiles and a special arrangement of spring loaded clutches in gear block *3* enable the transmission from a splined shaft *1* over a sliding gear *2* directly on to a stepped gear block *3* on shaft *4* and from there, via bevel gears *5* and *6*, a clutch or a back gear *7* on to the output shaft *6*.

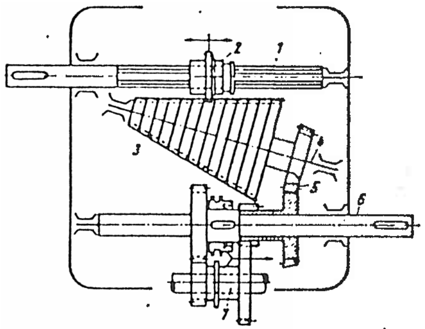


Fig. 145. Sliding gearbox (from *Stanki I Instrument*, December 1958).

In the Norton-type gearbox, the transmission ratios between driving and driven shaft are determined by a single pair of gears via the movable intermediate gear. The maximum transmission ratio is, therefore, determined by the space available for the largest gear, the minimum ratio by the permissible minimum number of teeth of the smallest gear.

The speed range of a Norton-type gearbox can be increased considerably if instead of a single pair of gears for producing different transmission ratios, several pairs can be arranged in series. If such a train of gears is arranged on two axes, and if a Norton carrier can be used for "tapping" one of the axes at different positions, the so-called "Meander" drive (Fig. 146) is obtained. The "Meander" drive differs from the Norton-type gearbox in that all gears remain engaged and, therefore, rotate continuously, although they are either heavily or lightly loaded according to the "tapping point". This fact has an unfavourable effect on the working and running accuracy of the drive. The transmission ratios between shafts *I* and *IV* in the "Meander" drive (Fig. 146) are:

$$u_1 = \frac{n_{tI_1}}{n_{tIII_x}} \cdot \frac{n_{tIII_x}}{n_{tIV_x}} = \frac{n_{tI_1}}{n_{tIV_x}}$$

* From *Stanki I Instrument*, December 1958.

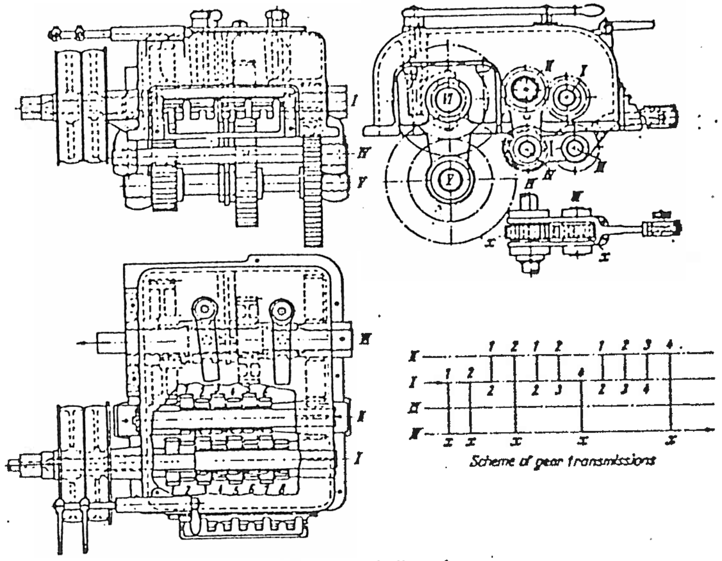


Fig. 146. "Meander" gearbox.

(III_x is an intermediate gear only, and its number of teeth drops out of the equation).

$$u_2 = \frac{n_{tI_2}}{n_{tIV_x}}$$

$$u_3 = \frac{n_{tI_2}}{n_{tII_1}} \cdot \frac{n_{tII_2}}{n_{tI_3}} \cdot \frac{n_{tI_3}}{n_{tIV_x}} = \frac{n_{tI_2}}{n_{tII_1}} \cdot \frac{n_{tII_2}}{n_{tIV_x}}$$

$$u_4 = \frac{n_{tI_2}}{n_{tII_1}} \cdot \frac{n_{tII_2}}{n_{tI_3}} \cdot \frac{n_{tI_4}}{n_{tIV_x}}$$

$$u_5 = \frac{n_{tI_2}}{n_{tII_1}} \cdot \frac{n_{tII_2}}{n_{tI_3}} \cdot \frac{n_{tI_4}}{n_{tII_3}} \cdot \frac{n_{tII_4}}{n_{tI_5}} \cdot \frac{n_{tI_5}}{n_{tIV_x}}$$

$$= \frac{n_{tI_2}}{n_{tII_1}} \cdot \frac{n_{tII_2}}{n_{tI_3}} \cdot \frac{n_{tI_4}}{n_{tII_3}} \cdot \frac{n_{tII_4}}{n_{tIV_x}} \text{ etc.}$$

$$\frac{u_1}{u_2} = \frac{n_{tI_1}}{n_{tI_2}}; \frac{u_2}{u_3} = \frac{n_{tII_1}}{n_{tII_2}}; \frac{u_3}{u_4} = \frac{n_{tI_3}}{n_{tI_4}}; \frac{u_4}{u_5} = \frac{n_{tII_3}}{n_{tII_4}}; \text{ etc.}$$

It has been shown earlier (see page 77) that in a driving mechanism which produces a geometric progression of output speeds, the ratios

$$\frac{u_1}{u_2} = \frac{u_2}{u_3} = \frac{u_3}{u_4}$$

etc., must be equal and constant, and that for this reason the ratios of the numbers of teeth must also be equal and constant.

$$\frac{n_{tI_1}}{n_{tI_2}} = \frac{n_{tII_1}}{n_{tII_2}} = \frac{n_{tI_3}}{n_{tI_4}} = \frac{n_{tII_3}}{n_{tII_4}} \text{ etc.} = \text{const.}$$

Moreover, it is necessary that:

$$n_{II_1} + n_{III_1} = n_{II_3} + n_{III_3} = n_{II_4} + n_{III_5} = \dots = \text{const.}$$

Furthermore

$$n_{II_2} = n_{II_4} = n_{II_6} \text{ etc.}$$

and

$$n_{II_1} = n_{II_3} = n_{II_5} \text{ etc.}$$

Hence

$$n_{II_1} = n_{III_1} = n_{II_3} = n_{III_3} \text{ etc.}$$

and

$$n_{II_2} = n_{III_2} = n_{II_4} = n_{III_4} \text{ etc.}$$

The twin gear blocks I_1–I_2, I_3–I_4, I_5–I_6, etc. and II_1–II_2, II_3–II_4, etc. are all equal and the gearbox is relatively easy to produce.

In the drives which have been described an intermediate gear on a pivoted carrier provides the compensation for the different centre distances between gears with different numbers of teeth. Such an intermediate gear is not required in clutch-type drives, where all gears are continuously in mesh,

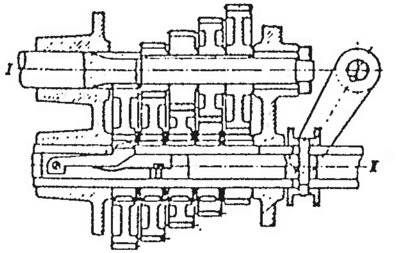


FIG. 147. Draw-key type gearbox.

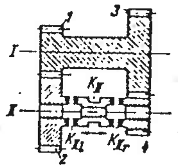


FIG. 148

the set of gears required for producing a particular transmission ratio being connected with the output shaft by means of a suitable clutch. An application of this idea, which makes possible a design of small length, even in the case of a large number of steps, is the draw-key drive (Fig. 147, see also Fig. 110). In this device, a number of gears run idly on a shaft. Any one of these can be connected with the shaft by a key (the draw-key), which can be axially moved to engage one of the gears through a radial spring operated movement. The required transmission ratio between driving and driven shaft can thus be obtained. However, a certain play between the movable draw-key on the one hand and the keyway in the shaft as well as the slot in the gears is necessary and unavoidable. Moreover, the load transmitting surfaces of draw-keys cannot be very large for obvious reasons, and as a result draw-key drives can transmit only relatively small torques and loads which fluctuate little. They are occasionally used for the feed drives of small drilling machines.

However, greater power can be transmitted by clutch-type gears in which axially controlled positive, e.g. dog clutches, or friction clutches are employed (Fig. 148). The gear block (gears *1* and *3*) is again keyed to the driving shaft *I*. The meshing gears *2* and *4* are idling on the driven shaft *II*, and can be connected with it by clutch K_{II} on the left (K_{II_l}) or on the right (K_{II_r}), so that shaft *II* is driven either by gears *1–2*–clutch K_{II_l} or gears *3–4*–clutch K_{II_r}. If at all possible it is advantageous to arrange the clutches on the driven shaft, because otherwise the idling gear would be driven by the fixed gear block at an excessive relative rotational speed on its shaft.

Clutch-type drives are particularly suitable for preselector gearboxes, because the clutches required to be operative for a particular output speed can be set ready for engagement, while the drive is still working at a previous output speed. At the moment of speed change only the mechanism engaging the clutch is put into operation and the gear sets producing the desired transmission ratio are engaged.

[illegible] However, the assumption of constant power transmission over a whole speed range results frequently in excessive safety factors because, e.g., the spindle drive of a milling machine will rarely have to transmit the full maximum power when the spindle revolves at its lowest speed.

If the torque transmitted by the output shaft must remain constant over the full range of output speeds, the power increases with the output speed (see Fig. 127). In this case, the driving motor must be able to provide the power required when the output shaft runs at its top speed:

$$P_{max} = \frac{M[Nm] \times n_{max}}{9550} \quad [kW]$$

This means that a limitation of the motor power by means of an overload relay or similar device, or a limitation of the input torque in the case of a constant input speed, would not protect the mechanism against an overload which might occur at the lowest output speed. A protective device which limits the output torque is then necessary on a shaft whose speed ratio related to the output shaft is constant (see Fig. 137, *f* and *h*), i.e. a slipping clutch on a shaft, which lies behind the last speed change device in the gearbox.

It may also be possible to combine the two conditions in such a manner that both the maximum power of the motor, as well as the maximum torque of the output shaft, are limited by protective devices. The application of such an arrangement will make the full power of the motor available down to a certain output speed below which the torque of the output shaft will be limited. Such an arrangement is, for instance, used in the Ward–Leonard sets (see page 99, Fig. 130).

Stepless Drives.[42] Electrical, mechanical or hydraulic devices may be used for producing speed ranges with infinitely fine steps. The electrical drives have been discussed earlier (see page 97). A discussion of the mechanical and hydraulic drives will follow.

Mechanical drives. The most elementary type of gear is the friction drive (Fig. 163), in which a friction roller (diameter *d*) drives a large disc. By axial displacement of the friction roller, the effective diameter *D* of the disc is changed, so that the ratio d/D can be varied in infinitely small steps. If the power, contact pressure, friction force and efficiency are constant, the output torque is inversely proportional to the speed of the output shaft, in other words, the torque decreases with increasing speed.

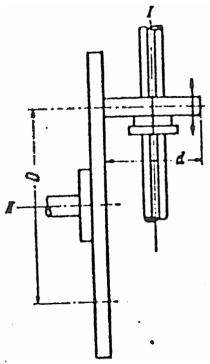


FIG. 163

The friction material of the driving roller should be softer than that of the driven disc, in order to ensure that the former remains round, even if the driven disc is stalled by an overload. The driving roller is, therefore, often covered by a leather or fibre ring, whilst the disc is made of steel.

In view of the relatively small area over which the friction force between the roller and the disc is transmitted, and because of the finite width of the driving roller, a certain amount of slip cannot be avoided. For this reason such drives are only suitable for transmitting relatively small torques, and are limited to reduction ratios of not more than 1 : 4.

Greater reliability and safety, longer life and higher efficiency can be obtained with more elaborate friction drives, some of which will now be described.[43]

In the driving mechanism of William Prym (Fig. 164), a ground conical casting *1* drives a ring of synthetic material *2*, which is held in a metal carrier. The latter is connected to the shaft *3* and via the gears *4* and *5* drives the output shaft *6*. The transmission ratio depends upon the axial position of cone *1* relative to the housing and this can be swivelled around shaft *6*. Its position determines the diameter *d*, against which ring *2* (diameter d_2) is pressed on to cone *1* under the effect of the torque acting on gear *5*. An important feature of this device is the fact that the pressure between the friction elements is thus proportional to the output torque; this keeps the slip and possible wear low.

Whilst the largest speed-range ratio of this drive is about 5, a speed-range ratio of up to 10 can be obtained in the more elaborate Prym drive (Fig. 165). Here, input and output shafts are coaxial, the torque being transmitted via two driving cones (*1* and *3*) having variable effective diameters d_1 and d_3, and two driven friction rings (*2* and *4*) with constant effective diameters d_2 and d_4. The transmission ratio can be changed by simultaneous variation of d_1 and d_3, because the shaft *5* with the ring *2* and the cone *3* is carried in a drum which is eccentrically supported relative to the axis of the drive. This drum is rotated by means of the hand-wheel *6*, the worm *7* and the worm wheel segment *8*. Under the effect of the transmitted torque, the right- and left-handed threads (*9* and *10*) axially displace ring *2* and cone *3* towards the outsides of the drive, and in this manner adjust again the pressure as a function of the load. These drives can transmit up to about 6 kW.

Intermediate members between the driving and driven elements are used in the friction drives (Figs. 166 to 169). In the "Heynau" gear (Fig. 166) the hardened and ground ring *3* made of high alloy steel is in contact with the tapered surfaces of two twin cones, *1a*/*1b* and *2a*/*2b*, respectively. By simultaneous axial displacement of the cones *1a* and *2b*, it is possible to vary the transmission ratio between input shaft *I* and output shaft *II* from an initial speed reduction ratio (Fig. 167a), to a 1 : 1 transmission (Fig. 167b) and to a speed increase (Fig. 167c). The maximum ratio between the effective diameters of the two cones being 3 : 1, it is possible to cover a transmission range of from 1 : 3 to 3 : 1, i.e. a speed-range ratio of 9.

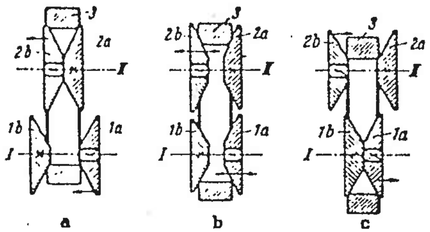


FIG. 167. Working principle of the Heynau drive. *I*—Driving shaft; *II*—Driven shaft.

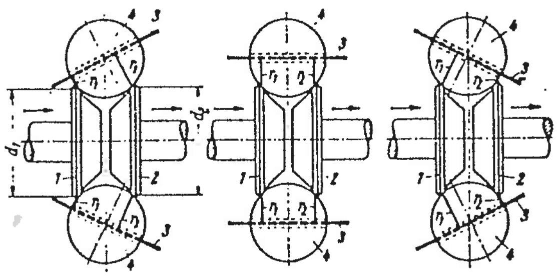


FIG. 168. Working principle of the Wülfel-Kopp Tourator.

In the Wülfel-Kopp Tourator (Fig. 168)[43] whose speed-range ratio is about 9, the effective diameters d_1 and d_2 of the discs *1* and *2* on the driving and driven shafts are constant, the steel spheres *4* supported on the shafts *3* acting as intermediate members. By changing the angular position of the shafts *3*, the effective driving radii *r* of the spheres are varied. The transmission ratio between driving and driven shaft is then:

$$u = \frac{d_1}{2r_1} \times \frac{2r_2}{d_2}$$

and with $d_1 = d_2$

$$u = \frac{r_2}{r_1}$$

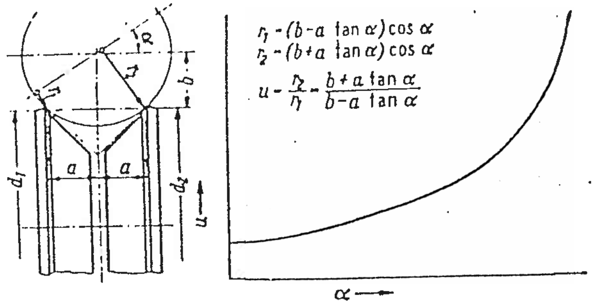


FIG. 169

The transmission ratio is, therefore, independent of the effective disc diameter and depends entirely upon the angular position (α) of the shafts *3* which carry the spheres *4* (Fig. 169). With

$$r_1 = (b - a \tan \alpha) \times \cos \alpha$$

and

$$r_2 = (b + a \tan \alpha) \times \cos \alpha$$

$$u = \frac{r_2}{r_1} = \frac{b + a \tan \alpha}{b - a \tan \alpha}$$

The transmission ratio is, therefore, not directly proportional to the angle α.

In the five drives previously described the torque transmission is not positive. This means that slip may occur, a fact which in certain cases is not permissible. An infinitely variable speed drive

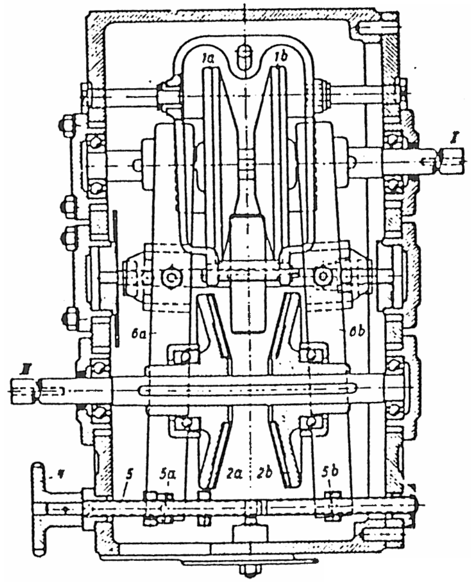


FIG. 170. P.I.V. drive: *I*—Driving shaft (input); *II*—Driven shaft (output).

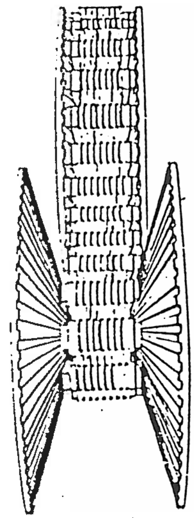


FIG. 171

with positive torque transmission is the P.I.V. drive (Positive Infinitely Variable). In this drive an endless chain transmits the torque between two chain wheels with variable pitch circle diameters (Fig. 170). Each chain wheel consists of a pair of cones which can be axially displaced (*1a/1b* and *2a/2b*, respectively). The teeth of the chain wheels are produced on the conical surfaces by the machining of radial grooves (Fig. 171). The two cones facing each other on each shaft are arranged in such a manner that the teeth are displaced by half a pitch relative to each other, so that a tooth on one cone faces a gap on the other. Each link of the torque transmitting chain consists of a frame which holds a certain number of laterally displaceable steel lamellae. These are pushed by the teeth of one cone into the gaps facing them on the other, and they adjust themselves, therefore, to suit the width of the teeth in action at the wheel diameter which is effective for any given setting (Fig. 172). The effective diameters, and with them the transmission ratios between shafts *I* and *II*, are changed by axial displacement of the two chain-wheel halves relative to each other. A rotation of the hand-wheel *4* and with it the screw *5* carrying a right- and a left-handed thread (*5a* and *5b*), moves the levers *6a* and *6b*

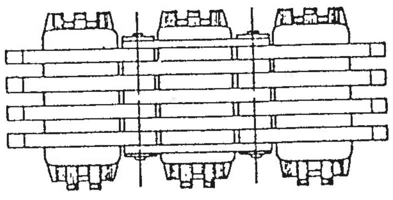


Fig. 172

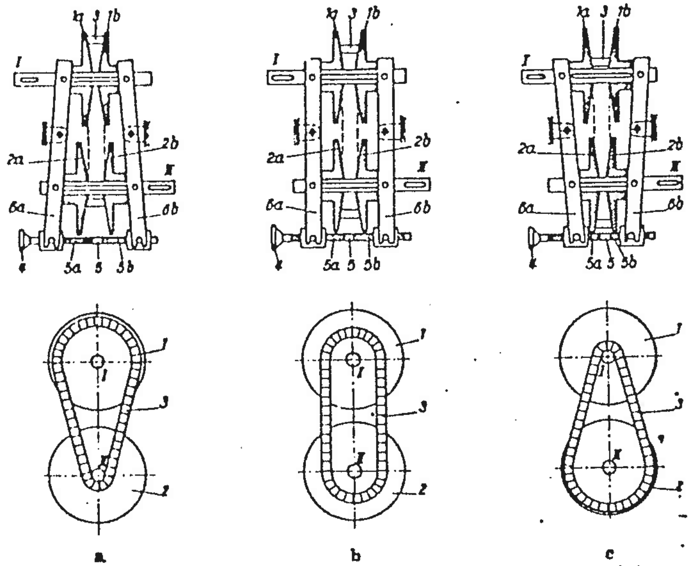


Fig. 173. Working principle of the P.I.V. drive. a—Speed increase; b—1:1 transmission; c—Speed reduction.

and through them the chain wheel cones *1a/1b* and *2a/2b* (Fig. 173). The chain tension is maintained by spring loaded jockey pulleys.

The efficiency of a P.I.V. drive is high, Fig. 174 showing typical efficiency curves, and speed range ratios of up to 6 are obtainable. As both pairs of chain wheel cones are designed to have equal effective maximum and minimum diameters, the speed range lies symmetrically around a mean transmission ratio of 1 : 1. At constant input speed n_I, the variable output speed is

$$n_{II} = \frac{d_1}{d_2} \times n_I$$

such arrangements would not require additional energy for the reversing action. Illustrations can be found in the crank drives discussed earlier, in some hydraulic reversing devices and in some electrical drives (Ward-Leonard control and shunt adjustment of d.c. motors).

The values given above represent, of course, only the theoretical energy requirements during the reversing action. In addition, friction losses in drives and mechanisms and heat losses in electrical supply lines must be considered. They result in a lower overall efficiency, but the general result remains unchanged.

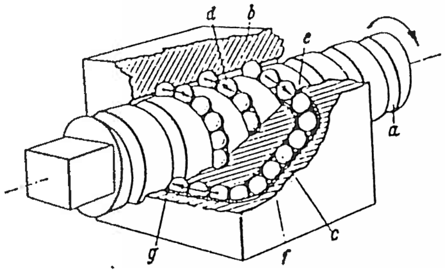


Fig. 203. Recirculating ball nut.

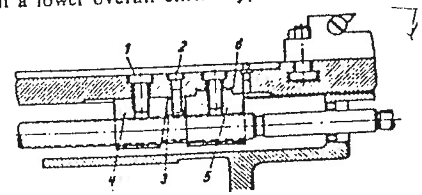


Fig. 204. Backlash elimination in the transverse feed of the Schaerer lathe (Industriewerke, Karlsruhe, Germany). After loosening screw *1*, the wedge *3* is drawn upwards by tightening screw *2*. This results in the movable portion of the nut *4* being pushed to the left. When the backlash is thus reduced to the permissible minimum, screw *1* is tightened again. Part *5* of the nut is held in its axial position by spigot *6*.

The efficiency of *screw drives* is generally rather low, especially if they are irreversible, i.e. if their helix angle (α) is small. If the friction angle is ϱ (friction coefficient $\mu = \tan \varrho$) the efficiency is

$$\eta = \frac{\tan \alpha}{\tan(\alpha + \varrho)}$$

In certain cases irreversibility is desirable, for instance, when the axial load on the screw should not be transmitted to the driving elements (see Fig. 211). In general, however, low friction and the resulting increase in mechanical efficiency are preferable. Even in cases where, for instance, the screw drive is to be reversed and the friction assists the decelerating action before reversing, it opposes afterwards the acceleration in the opposite direction. The efficiency of a normal single start screw and nut is of the order of 30 to 50 per cent.

Frequently, a reversing action of screw drives is unnecessary, for instance, after screwcutting with a lead screw on a lathe the lead screw nut can be opened and the quick return is obtained by means of a rack and pinion.

An interesting design with considerably higher efficiency incorporates the recirculating ball nut, (Fig. 203), in which the load between the flanks of the screw thread *a* and the nut *b* is not transmitted by direct contact, but through the intermediary of balls *c*. The balls roll between the flanks of the threads (arrows *d*) in a similar manner as they do between the outer and inner races of ball bearings. When the balls leave the nut at one end (side *e*) they are returned through a channel *f* to the other side *g*, where they again re-enter the nut and continue the operation. Such screw drives have an efficiency of about 93 per cent. It is, of course, necessary to consider that their stiffness is slightly less than that of ordinary screw drives of equal size, because of the insertion of an intermediate elastic member in the form of the load transmitting balls.

The unavoidable play between the flanks of screw and nut drives results in a certain amount of backlash which, in many cases, is not only undesirable but also not permissible. Should such backlash become excessive due to inaccurate manufacture or wear, it can be reduced from time to time either by axial displacement or by relative rotation of two nuts acting on one screw. After such adjustment, the two nuts can then be locked in position (Fig. 204). However, such manual adjustment can only cover the minimum play which exists in any part of the screw, and not the most heavily worn part of a long screw drive, because otherwise the lesser worn parts of the screw would not be able to rotate in the nut which had been adjusted to eliminate larger play. If, therefore, such manual adjustment is insufficient, an automatic backlash eliminating device may have to be provided. The simplest design would consist of two nuts which are axially or rotationally pre-loaded against each other, by means of an elastic member, e.g. a spring. The spring pre-load must be greater than the maximum axial load or torque respectively, as otherwise undesirable vibrations could occur. In the case of recirculating ball nuts (see Fig. 203), the loss in stiffness, which could be caused by the use of springs, has been reduced by axially clamping two nuts together, the pre-load being produced by the insertion of shims between the nuts. The disadvantage of such arrangements lies in the fact that the pre-load produces frictional forces which have to be overcome even if the drive is not heavily loaded. In the case of recirculating ball nuts such friction losses are, however, very small.

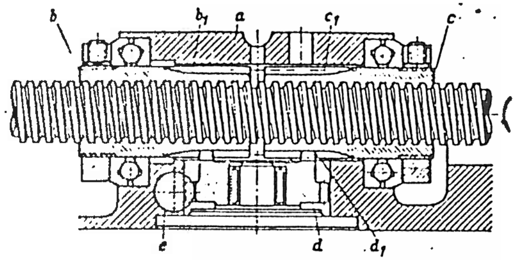


FIG. 205. Backlash eliminator in the feed drive nut of a milling machine table. (The Cincinnati Milling Machine Company, Cincinnati, Ohio, U.S.A.).

Figure 205 shows a device which permits manual as well as automatic backlash elimination. The two nuts *b* and *c* can rotate in the housing *a* and carry pinions b_1 and c_1. A torque which acts on one nut is transmitted by the pinion via the crown wheel d_1 on to the other nut, which is thus rotated against the first one, the resulting pre-load being proportional to the load transmitted by the screw drive. Moreover, crown wheel d_1 is part of pinion *d*. This can be rotated by rack *e* which is pre-loaded by a compression spring. The pre-load of the spring is manually adjustable and helps in reducing the load on the screw thread under small loads.

When using a nut which can be displaced by the axial component of the tooth pressure acting on a helical gear, the pre-load can be kept proportional to the torque transmitted. Such an arrangement is shown in Fig. 206. Worm wheel *1* drives via clutch *2* two pinions (spur gear *3* and helical gear *4*), which are rigidly located in the axial direction and keyed to the driving shaft. The spur gear *3* drives via spur gear *7* one nut which is rigidly held in the axial direction between pre-loaded ball thrust bearings *5* and *6*. The helical gear *4* drives the helical gear *8* on the second nut, which is axially unrestrained and can, therefore, be displaced relative to nut *7*. Nuts *7* and *8* are both on the lead screw *9*, which is fixed and secured against rotation in the moving part of the machine (in this case, the table of a milling machine). The axial component of the tooth pressure between gears *4* and *8*, which is proportional to the torque transmitted between these two gears, displaces nut *8* axially as soon as play appears between its flanks and those of the lead screw. This play is thus automatically reduced. If the play decreases over another part of the screw, an axial displacement in the opposite direction is produced, as the gear teeth slide accordingly.

The length of stroke of a screw and nut drive is limited, because with increasing length the sag of the screw increases and its stiffness decreases. If long strokes and drives with high degrees of stiffness are required, singly or in combination, the ordinary screw and nut drive may be replaced by a worm and a nut-type rack (Fig. 207), or by a worm and a rack with inclined teeth (Fig. 208). The latter arrangement allows the axis of the worm to be inclined relative to the rack. This enables the driving mechanism for the worm to be arranged outside the rack, so that the length of the driving

shaft can be kept relatively short and independent of the length of the working stroke of the drive. The efficiency of these drives is similar to that of a screw and nut.

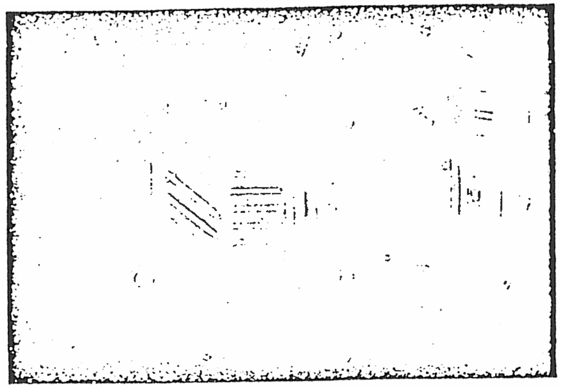


FIG. 206. Backlash eliminator in the feed drive nut for the table of the milling machine (Fig. 133).

Higher efficiency values can be obtained with a rack and pinion drive (Fig. 209, see also Fig. 195). If large forces have to be transmitted, it is often advisable to use large diameter gears, so that several teeth are in mesh simultaneously. Large diameter gears also ensure quieter and smoother running than would be possible with small pinions. Backlash in these types of drives again can be detrimental to their working efficiency. Figure 210[46] shows an interesting design of a pinion and rack

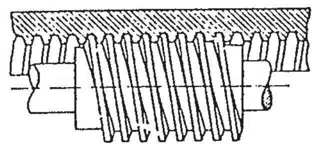


FIG. 207

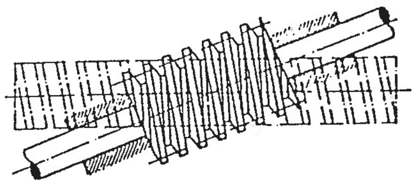


FIG. 208

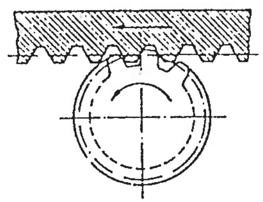


FIG. 209

drive with backlash elimination. The two pinions *1* and *2* are driven by two gear trains in which the helical pinions *4* and *5* are displaced axially by bellows *3* (which provide an axial pre-load under air pressure), and rotate the two gear wheels which are driven in opposite directions by the helical pinions. This results in the teeth of the pinion *1* bearing on one side, those of the other pinion *2* bearing on the other side of the rack rooth flanks and this in turn eliminates any possible backlash.

(b) The advantages of *hydraulic drives* (see page 125)*, especially the possibility of smooth reversing and quiet and vibration-free working, are particularly useful when rectilinear reciprocating movements have to be produced. The hydraulic motor is usually a cylinder and piston, the cylinder being fixed and the piston moving, or vice versa. The mechanical connexion between the hydraulically moved element (piston or cylinder), and the actual machine part which is to be moved (table, slide, saddle, etc.), can be rigid and direct or through intermediate members (for

* For a detailed study of these drives see reference 44.

that the resulting relative movements between the different slides satisfy the accuracy requirements of the operation. This ensures that at any moment the slides will be within the required limits of their positions which are determined by the required relative position between tool and workpiece. In order to obtain completely satisfactory conditions, the misalignments of each slide must be corrected in three dimensions. For the purpose of simplicity, however, the idea may be explained with the help of a two-dimensional example. Figure 293 shows in schematic form a milling machine

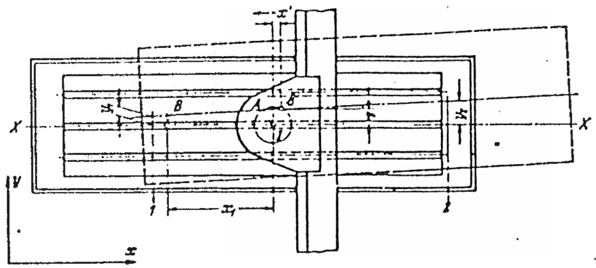


FIG. 293

table and spindle head. It is required to move the table by an amount x_1 in the direction of the x-axis, whilst the spindle slide, which can be moved in the direction of the y-axis should remain in its position (milling of a straight line groove or edge). In other words, the axis of the milling cutter is to move from point A to point B along the centre line of the table (distance $AB = x_1$). It may now be assumed that due to inaccuracies of slideway manufacture or for any other reason, the table which ought to move in a direction parallel to the line X–X is laterally displaced by an amount measured at the two points 1 and 2 and equal to the y-coordinates y_1 and y_2. This means that after traversing the distance x_1 point B arrives in a position B'. In order to maintain the required relative position between the milling spindle 1 and the table (point B), the table must make an additional movement $-x'$ and the spindle head a correcting movement $+y'$. The magnitude of these movements can be determined by the computer on the basis of the measurement signals y_1 and y_2 and in accordance with the theoretical position of the table relative to the cutting edge. Corresponding signals must now be transmitted to the devices which control the feed drives in the direction of the x- and y-axes and superimposed on to the existing signals.

In order to make full use of this idea, the position of each slide ought to be measured at three points in the direction of the two axes (in the case of the table the y- and z-axes, in the case of the spindle head, the x- and z-axes). This will make it possible to cover not only lateral displacements in the direction of the two axes, but also rotational displacements around the three axes. It would also, if necessary, be possible to cover the position of the spindle in its bearings in a similar manner. How far one has to go in each case is a question of practical requirements and these have to be investigated from case to case. For a portal milling machine produced in accordance with normal commercial acceptance tolerances, which has to work within an accuracy of 0·0001 in., D. L. Leete[75] has determined the required measuring accuracies by means of statistical methods (Table 20).

The practical execution of this idea necessitates the solution of optical, electronic and mechanical problems.

DESIGN OF CONSTRUCTIONAL ELEMENTS

1. MACHINE TOOL STRUCTURES

The beds, columns or frames form the backbones of machine tools. They have to transmit the weights of various parts (headstocks, slides, etc.), on to the supports (foundations, supporting

wedges), and they have to close the flow of the operational forces which are exerted between workpiece and tool carrier during cutting operations.

The power capacity, the required working accuracy and the ability to produce a machined surface of the quality specified by the designer of the workpiece determine the necessary static and dynamic stiffness (see page 43); the operating and loading conditions and the arrangement of the various parts of the machine tool (tool and workpiece carriers, gearboxes, control equipment, motors) affect the shapes and layouts of the design. The basic principles which have to be considered in order to obtain the required static and dynamic stiffness, have already been discussed (see page 43). They have to be correlated with the required layout of the machine as a whole, the ease of its manufacture, assembly, maintenance and operation, the requirements of the working conditions (lighting, inspection, chip and swarf removal, etc.), in such a manner that the finished design is not only technically acceptable, but also aesthetically satisfactory.

In order to satisfy all these requirements, it is necessary not only to consider basic principles which are determined by the type and operation of particular machines (lathe, drilling machine, milling machine, planing machine, etc.), but also to investigate and specify the following:

(A) Installation
(B) Power requirements and loading conditions (forces and velocities)
(C) Points of application and direction of the forces which are transmitted by various parts of the machine on to the structure
(D) Stresses and deformations
(E) Materials of the structural components
(F) Shapes and quantity of the chips.

These will now be discussed in detail.

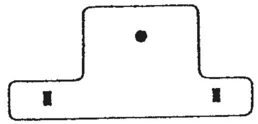


FIG. 294

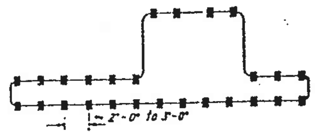


FIG. 295

(A) Machines which have to satisfy requirements of high precision are usually freely supported at three points without restraint. The vertical supporting forces (example of a grinding machine, Fig. 294), are the reactions to the weights of the machine bed and the machine parts carried on the bed (headstock, slide, grinding wheel, workpiece, etc.). The supports cannot and do not transmit any other forces exerted on the machine bed, such as centrifugal or cutting forces. As these latter are, therefore, not transmitted to the foundation, it is not permissible to consider in any way a stiffening effect which the foundation may have on the bed. The bed itself must be capable of transmitting these forces satisfactorily, i.e. in such a manner that it would perform its duties even if it were suspended from a crane.

If three-point support was to be applied to very long beds, it would be necessary to provide for very deep and stiff cross-sections in order to obtain the necessary stiffness. For this reason, long beds of precision machines usually rest on more than three points. In order to facilitate the levelling and aligning of such machine beds, they are often supported on wedges placed about 2 ft to 3 ft apart (Fig. 295). As soon as they are satisfactorily levelled, they are grouted in, so that not only the weights but also the deforming working forces are transmitted to the foundation. If, in addition, the bed is tightened to the foundation by means of anchor bolts, not only compressing but also tensile forces can be transmitted, and the stiffness of the bed is thus increased. As an example, the deformations of a base plate for a radial drilling machine were found to be reduced by 30 per cent when the base plate was grouted and bolted to a suitable foundation.

Instead of using simple wedges, it is possible to provide at each supporting point two holes, one plain and one tapped, and these can serve for one tension and one compression screw respectively. Instead of driving or withdrawing a wedge, it is then possible to lift the particular point of the bed by tightening the compression screw, or to draw it down against the foundation by tightening the tension screw which is anchored in the foundation. In the grinding machine bed (see Fig. 331), twelve such supporting points are provided, and at each of these two such holes are arranged at about $3\frac{1}{2}$ in. centre distance, a tapped hole ($\frac{3}{4}$ in. dia.) for the compression screw and a plain hole ($\frac{11}{16}$ in. dia.) for the tension screw.

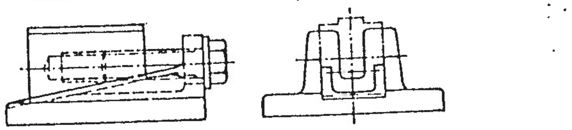


FIG. 296

Some deformations may occur with time even if a bed is grouted to the foundation. They may reach excessive values, especially in cases of precision machines, such as precision planers. The beds of such machines are usually supported on adjustable wedge units (Fig. 296), so that the beds can be tested from time to time (every one to two months), and their levelling re-adjusted if necessary.

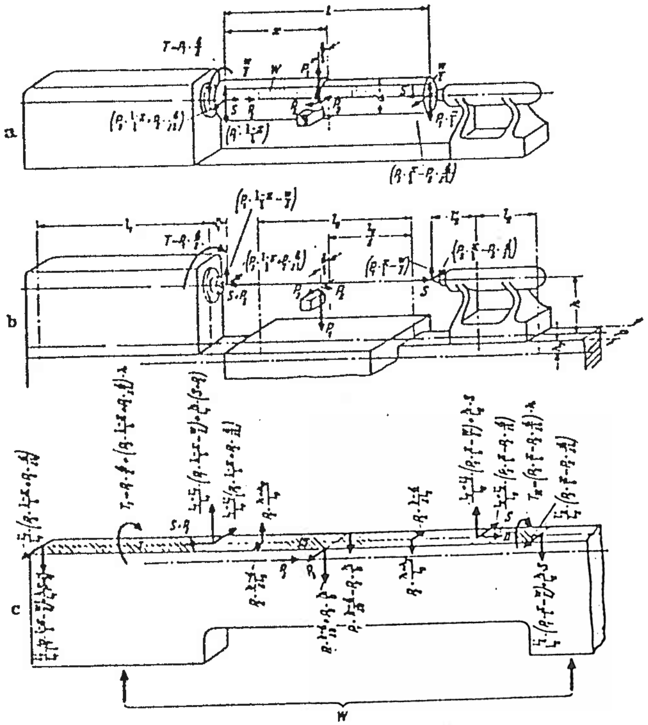


FIG. 297

The wedge units, in which a supporting block is displaced along a sloping surface by means of a screw, thus varying the height of the supporting face, are usually grouted to the foundation. After re-alignment, the bed can be tightened against these supporting faces by means of anchor bolts.

(B), (C), (D) The cutting and operational forces have to be determined in accordance with the working conditions (see page 1 *et seq.*). Their reactions and the resultant forces transmitted upon the

structure must be analysed. If the masses of certain parts are to move at relatively high speeds, it is also necessary to consider the effect of inertia forces, not only upon stresses and deformations, but also upon the vibration conditions (see page 55). The magnitude of stress is, however, usually less important, because the requirements of stiffness necessitate cross-sections and layouts which result in low stress levels.

The magnitude of the permissible deformations is determined by the required accuracy and surface quality (see page 23). An accurate calculation of the deformations is often difficult or even impossible, because the shapes of beds, columns and frames are usually relatively complex and it is not easy to determine with any degree of accuracy, the exact type of load application (concentration, distribution over a certain length, etc.), between the various parts of the machine structure. For the theoretical analysis, certain assumptions must, therefore, be made and although these may not produce accurate results, they provide important indications which the designer can use during his work. The conditions of force application may be studied for some typical examples.

(i) Centre lathe. In Fig. 297a, the forces acting on the workpiece are shown. The cutting force is resolved into three components (P_1, P_2, P_3, see Fig. 3). These components are exerted by the tool edge on the workpiece (length l) at a varying distance x from the headstock centre and at a diameter d. They are kept in equilibrium by the supporting forces which act at the headstock $P_1 \times (l - x)/l$; $P_3 \times (l - x)/l + P_2 \times d/2l$ and P_2 and tailstock centres $P_1 \times x/l$; $P_3 \times x/l - P_2 d/2l$ and by a torque which is exerted by the driver on the spindle nose $T = P_1 \times d/2$.

As the difference in diameter of the machined and unmachined lengths is relatively small, it is assumed that the weight of the workpiece (W) is evenly distributed over its length and held in equilibrium by two equal supporting forces $W/2$, one acting at each centre. The axial pre-load S is exerted by the centres onto the workpiece. The forces which act on the spindle nose, the tailstock and the tool rest on the saddle are equal and opposite to those exerted on the workpiece (Fig. 297b), and thus determine the forces which are exerted by the headstock, the tailstock and the saddle on the bed (Fig. 297c). The part I of the bed surface is covered by the headstock. The tailstock is usually held down at the front by one or more clamping bolts, and the area II is that between the centre lines of these bolts and the rear edge of the tailstock. The area III is that part covered by the saddle. The feed force component P_2 of the cutting force acting on the saddle at the height of centre (h) is held in equilibrium by an equal and opposite force which acts on the feed pinion at the pitch line of the rack (distance h_3 below the bed surface). This results in a tilting moment $P_2 . (h + h_3)$ which has to be counteracted by the saddle slideways. In Fig. 297c are shown the forces and moments which are exerted on the bed by the headstock, the tailstock and the saddle. They form an equilibrium system excepting that the weight W of the workpiece is transmitted directly to the legs and the foundations, together with the weight of the machine. With this exception of W, the flow of forces is, therefore, closed within the bed which is thus stressed in tension (very slightly by P_2 and S), vertical bending (upwards at the front ends of headstock and tailstock, downwards under the saddle), horizontal bending (similar to vertical bending) and torsion. An analytical or graphical determination of the deformations under the assumption that the cross-section of the bed and the magnitude, direction and points of application of all forces are known, is relatively simple and need not be discussed in detail. It is, however, necessary to consider the relative importance of the various deformations.

In Fig. 298, let

H = height of workpiece axis above axis of bed
d = machined diameter of workpiece
δ = displacement of cutting edge arising from deformations of the bed
δ_1 = this displacement in the vertical plane (Fig. 298a)
δ_2 = this displacement in the horizontal plane (Fig. 298b)
δ_3 = this displacement due to torsion (Fig. 298c)

If it may now be assumed that $\delta_1 = \delta_2 = \delta_3$, then it will be clear that the deformation in the vertical plane has less effect upon the diametral error Δd_1 than the deformation in the horizontal plane

The design of slideways for tables, saddles, cross-slides, etc., will be discussed under the following aspects:

(A) Shapes of the guiding elements and arrangements of their combinations
(B) Effect of material and working conditions upon the guiding accuracy (wear)
(C) Friction conditions and load carrying capacity (roller bearings, lubrication, etc.).

(A) Slideways have to satisfy the following requirements:

(1) To give exact alignment of the guided parts in all positions and under the effect of the operational forces
(2) There are to be means of compensating for possible wear

Fig. 343

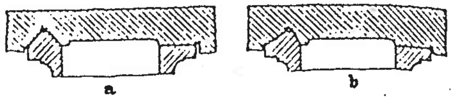


Fig. 344

(3) There must be ease of assembly and economy in manufacture, i.e. possibility of adjusting the alignment in order to allow for manufacturing tolerances
(4) To allow freedom from restraint
(5) There must be prevention of chip accumulation and ease of removal of any chips
(6) Effective lubrication must be possible.

The design of slideways is usually based on one or several of the following elements, and these can be arranged in different positions and combinations:

(a) The Vee, Fig. 343a
(b) The flat surface, Fig. 343b
(c) The dovetail, Fig. 343c
(d) The cylinder, Fig. 343d.

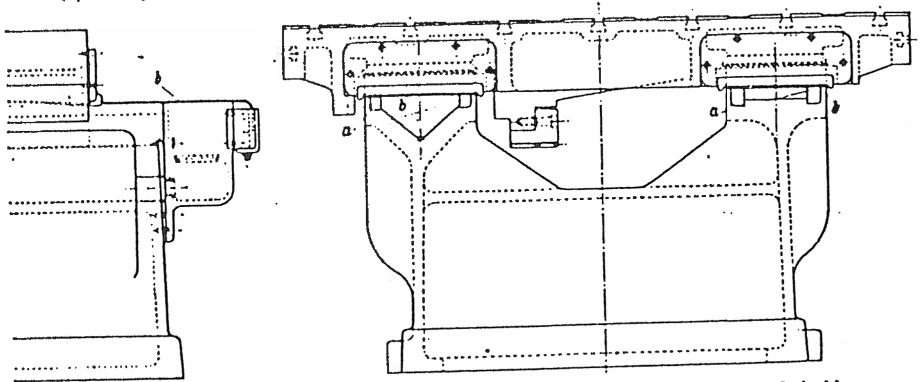


Fig. 345. Protected slideway for a grinding machine table (The Churchill Machine Tool Co. Ltd., Manchester). a—Oil channels; b—Protective cover.

The Vee (Fig. 343a), may have its apex upwards (see Fig. 344), or downwards (see Fig. 345), and with it the positions of the guided part are determined in two directions, in the example of Fig. 343a, vertically and horizontally in the plane of the picture. In order to satisfy the requirement of unrestrained guidance, it is usual to combine one Vee, either symmetrical (Fig. 344a) or unsymmetrical (Fig. 344b), with a flat slideway (Fig. 343b). Figures 344 and 345 show such combinations. However,

even today some centre lathes are equipped with two Vee slides for the saddle (see Fig. 321). Although in this case it is theoretically possible yet practically improbable that all four faces of the Vee are in perfect contact and carry the forces acting on them, some designers prefer this arrangement because of the reduced wear effects (see page 250), upon the working accuracy.[82]

One advantage of the Vee lies in the fact that it is self-adjusting under the weight of the guided part, so that even after wear or other conditional changes, play cannot develop. The Vee which points upwards also prevents accumulation of chips on the sliding surfaces. The Vee which points downwards and is usually found in planing and grinding machines, can contain the lubricating oil.

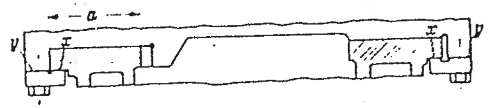


Fig. 346

It is, however, necessary to give it careful protection (Fig. 345), in order to prevent chip accumulation, unless the design of the bed is such that the slideways are outside the range of falling chips or covered by other parts of the bed (see Fig. 321).

The combination of several flat slideways is often used for transmitting high supporting forces on long slideways (Fig. 346). The locations in the horizontal and vertical directions are then independent and the alignment and fitting of slides thus facilitated, because an adjustment in one direction does not result in a displacement in the other, as in the case of Vee slides. Guiding surfaces for location in the secondary direction (vertical faces in Fig. 346) are suitably arranged as close together as possible (distance a), in order to prevent skewing or jamming. Play adjustment or wear compensation is not automatic as in Vee slides, and it is therefore necessary to provide an adjustable strip.

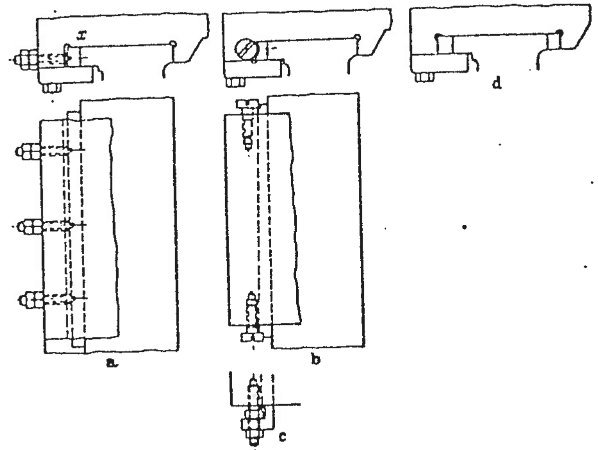


Fig. 347

This can either have parallel faces, being adjustable by means of laterally arranged screws (Fig. 347a), or it can be wedge shaped and adjustable by longitudinal displacement (Fig. 347b). In the case of Fig. 347a, where the adjustment screws have to be tightened separately, the tightening force depends upon the operator's touch and will hardly be uniform. In addition, such strips are deflected at the points of screw application and do not, therefore, carry the loads uniformly. Moreover, as the tightening screws must keep the strips in position (counter bore x, Fig. 347a), it is possible that, during the longitudinal movement, the screws work loose unless they are secured by lock nuts. In order to avoid excessive stressing of the adjusting screws, such strips are usually arranged on that side of the slideway which is not exposed to heavy loads.

The slightly more expensive taper strip (Fig. 347b) bears on its whole length and therefore provides better conditions. The bearing area is independent of the positional adjustment and with the usual tapers of 1 in. in 5 ft or 1 in. in 8 ft, fine adjustments are possible. Care must be taken that the heavy mechanical advantage provided by the wedge effect does not create considerable lateral stressing. The tightening of the taper strips must, therefore, be carried out with great caution. In addition, taper strips must be prevented from undesirable longitudinal displacement, for instance, under the effect of friction forces, which may either loosen or tighten them. The provision of two adjusting screws (one at each end, Fig. 347b), or a stud with a nut and lock nut (Fig. 347c), may serve this purpose. If very long taper strips are necessary, the required minimum thickness at the thinner end may result in a weakening of the guided part at the thicker end of the strip. This weakening may be excessive, e.g. in the case of a taper of 1 in. in 5 ft and a 22 in. long slide, the thickness difference at the two ends of the taper strip is almost ⅜ in.! If the transverse position of a slide relative to its direction of movement must not be affected by play adjustment—for instance, in the case of turret head slides where the turret head axis must be aligned with the spindle axis—two adjusting strips are usually provided (Fig. 347d).

a

b

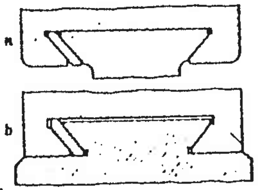
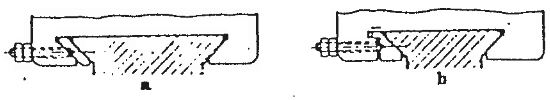


Fig. 348



a b

Fig. 349

Both the Vee (Figs. 321 and 344) and the flat slide (Fig. 346), secure the vertical location only in the downward direction, i.e. under the effect of the weight of the guided part. If these guided parts are heavy, as, for instance, the tables of planing or grinding machines, this should be sufficient. If, however, forces or couples occur which may tend to lift or tilt the moving parts (see page 215), holding strips have to be provided (Fig. 346). These must be carefully adjusted in order to ensure that the play in the vertical direction is not excessive. For this reason, it is advisable to separate the sliding face (x) from the fitting face (y) by a groove, so that the fitter files or scrapes only one or the other and knows exactly how far he has to go on each face.

The shape of dove-tail slideways locates the guided parts horizontally and vertically, the latter both upwards and downwards. Either the inner (Fig. 348a) or the outer (Fig. 348b) faces serve for carrying the vertical load. Play adjustment in two directions (vertical and horizontal) can be carried out by the use of only one strip (see Fig. 348). Adjustment by means of set screws (Fig. 349a) has the disadvantages which have already been mentioned. The strip can be held in place vertically by using a "hooked" profile (Fig. 349b).

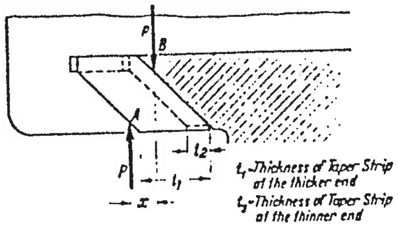


Fig. 350

It has already been mentioned that taper strips for long flat surfaces must be so designed that the wall thicknesses of either the guiding or guided parts are not excessively weakened at the thicker end of the strip. A further difficulty arises when taper strips are used on long dove-tail slideways. Here the thickness of the strip may reach a value at which the stability of the guiding action is reduced (Fig. 350). If point A lies too far outside the vertical line of action through point B, a tilting couple ($P \times x$) creates instability. Long slideways are, therefore, often equipped with two taper strips (Fig. 351). Another solution which is even more favourable in the case of dove-tail slideways makes use of a wedged (Fig. 352) and not parallel (Fig. 348) cross-section for the strip, because in this case the wedge effect of the cross-section counteracts the tilting couple ($P \times x$).

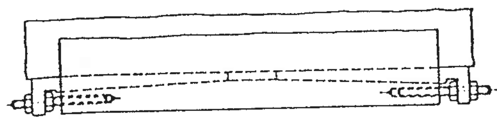


Fig. 351

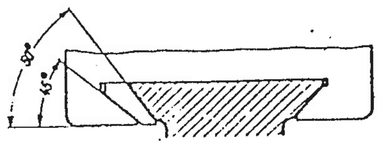


Fig. 352

The detailed parts which make up the foregoing designs can be assembled either by sliding them together in the direction of their intended working movement, or by tilting the guided part into position (Fig. 353). The former method of assembly is possible only when sufficient space in the longitudinal direction is available and the sliding parts are relatively light. The second method is permissible only if the lower edge of the taper strip need not be too far away from the guiding surface (Fig. 350).

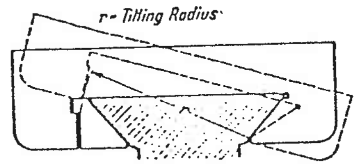


Fig. 353

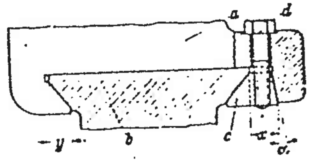


Fig. 354

In the case of very heavy parts (such as milling machine knees), neither of the two methods is really suitable, and a wedged strip (Fig. 354) is often applied. The moving part such as the knee a can be fitted to the slideway b at any point of its traverse and the strip c can be inserted afterwards. In order to make such an assembly possible, the angle α and the dimension x must be so chosen that $x > y$. By tightening the screws d the play can be adjusted, and by providing two clamping screws, it is possible to clamp the moving part, if required. As the steeper side of the wedged strip is sloping outwards ($\alpha = 5$ to $10°$), the strip will be moved from right to left, when it is tightened.

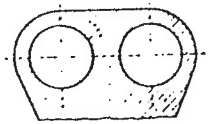


Fig. 355

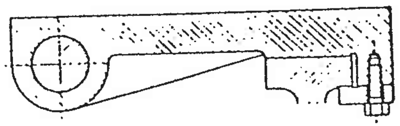


Fig. 356

It is therefore necessary to provide corresponding play in the clearance holes for the tightening screws. Fitting and scraping must be very accurate, so that the strip bears well on the sliding surfaces, even after adjustment for wear.

The use of cylindrical guiding elements enables the designer to apply kinematically determinate and restraint-free slideway arrangements. However, the manufacture of the various components must be very accurate because cylindrical slideways are difficult to scrape or fit. They can be extremely stiff and are, for instance, used as overarms for milling machines. Other examples are the column for the radial drilling machine, the drilling spindle sleeve, the tailstock sleeve for a lathe, etc. The guiding device using two cylinders (Fig. 355) is not free from restraint and must be manufactured with great care. A combination of a cylindrical and a flat slideway which is often used in optical length L_3) by an amount l equal to the total movement (Fig. 374). This results in the slideway surfaces always being covered by the moving part. If it is impossible to design slide and slideway in this manner, the length of the moving part can be extended by the application of cover plates, which do not do any operational work but form a continuation of the moving part over the slideway. This solution is, for instance, applied to the table slideways of the milling machine (Figs. 133 and 134). However, such a cover is fully effective only if the covering surface is in close contact with the guiding surface, as otherwise dirt may get underneath the cover and from there, between the working surfaces.

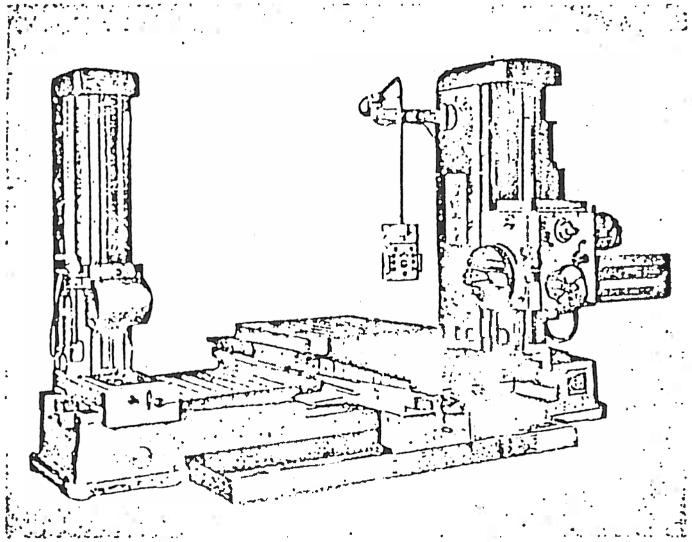


Fig. 377. Protection of slideways by means of telescopic cover plates on a Plauert-Wetzel horizontal boring machine (Vereinigte Werkzeugmaschinenfabriken, Frankfurt a.M., Germany).

Another design solution is the provision of covering devices which surround the otherwise open guiding elements and seal them off hermetically. Such devices have to act either telescopically, by becoming lengthened and shortened, as required, during the movement (see Fig. 377), or they can be of the concertina-type (Fig. 375). Covering belts may also be arranged above the guiding surfaces and these are either kept in tension by spring-loaded rollers (Fig. 376), or they cover the whole length of the fixed guiding surface and are lifted off over the length of the moving part (see Fig. 345). Figure 377 shows an application of telescopically arranged cover plates. In these designs, hermetic sealing is, of course, not possible.

Fig. 378

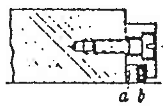


Fig. 379

The provision of simple felt seals (Fig. 378) is not advisable, as these are subject to wear and lose their effectiveness. Under such circumstances, it is better to combine the felt seal a with a rubber seal b (Fig. 379).[82] In order to ensure the required pressure between the seal and the guiding surface, a leaf spring b can be arranged between the cover strip a and the seal itself (Fig. 380).[82] An even better protection of the seal is provided by a spring loaded brass strip (a, Fig. 381).[82]

Instead of exposing the accurately ground or scraped slideway surface to wear, an intermediate elastic member can be inserted between the guiding and the guided surface, for instance, a thin steel tape held tight by a tensile preload. This hardened tape attaches itself tightly to the shape of both

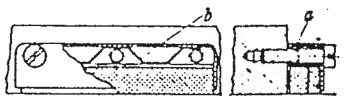


Fig. 380

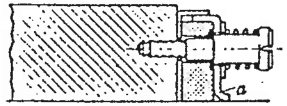


Fig. 381

surfaces and, being of constant thickness within very fine limits, keeps the distance between the guiding and the guided surface constant. In the design (Fig. 336), such a steel tape can be seen under the roller bearings which carry the spindle head of a radial drilling machine. Apart from the protection of the cast iron sliding surface against dirt, the insertion of the hardened steel tape has a double purpose:

(i) The wear is less than that of an unhardened cast iron surface, and the surface pressure acting on the cast iron surface is evenly distributed over a greater length and therefore reduced.

(ii) If the steel tape should become damaged, it can be more easily replaced than a cast iron surface.

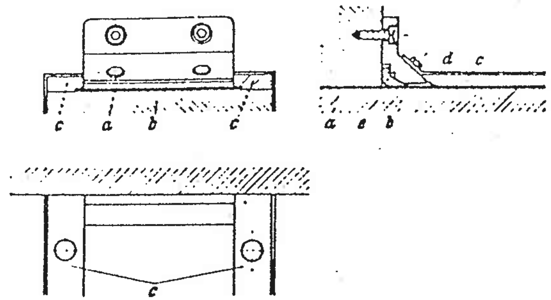


Fig. 382. Arrangement of scrapers on flat slideways (Scharmann). a—hardened steel tape; b—slideway casting; c—lateral steel strips; d—scraper fitted without play between c; e—"hydrofit" seal scraper.

In the case of the example shown in Fig. 382,[89] the upper surface of the protective steel tape (a), which covers the cast iron slideway (b) and lies between the two steel strips (c), is protected against dirt by a spring steel scraper (d) and a seal (e).

(C) The friction conditions in slideways are important not only from the point of view of wear. The forces and powers required for moving the various parts and the accuracy of their control are very much affected by the kind and magnitude of friction resistances. Of particular importance is the so-called "stick-slip" effect, which is caused by the fact that in many cases the coefficient of static friction (friction coefficient at speed $v = 0$) is higher than that encountered at a definite low speed $v < v_e$. The friction coefficient μ increases, of course, again with further increasing velocity $v > v_e$ (Fig. 383).[90] Over the range $v < v_e$, the friction has therefore a negative damping effect: If at the beginning of a setting movement the driving elements have to be strained until the displacement force necessary to overcome the static

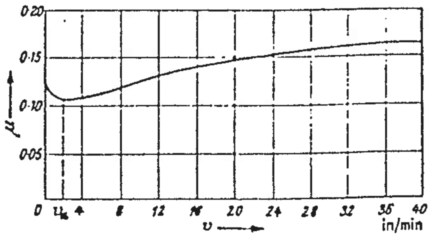


Fig. 383. Friction coefficient between the slideway and the saddle of a horizontal boring machine, as a function of the saddle speed.

FRICTION

friction (at $v = 0$) is reached, and if then the frictional resistance drops as soon as the movement starts, the energy initially stored in the strained driving members is suddenly released and drives the moving part beyond the intended distance. This makes accurate setting sometimes difficult, if not impossible, especially if the total setting movement is small.

Apart from the application of different materials for the slideways (cast iron, steel, bronze, plastic, etc., see page 143) and the use of suitable lubricants, it is possible to influence the friction conditions by appropriate design measures.[90] The sliding speeds usually encountered in machine tools are too low to obtain hydrodynamic lubrication conditions. If the designer can, however, ensure that a certain minimum oil quantity is supplied between the moving surfaces either by automatic means or by the operator, and if lubrication grooves are arranged in such a manner that the oil is distributed over these surfaces without breaking the oil film (Fig. 384), a state of semi-fluid friction may be

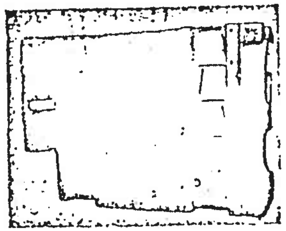


FIG. 384. Arrangement of oil grooves in the sliding surfaces of a horizontal boring machine (H. W. Kearns & Co. Ltd., Broadheath). Depth of groove $\frac{1}{8}$ in.; Width of groove $\frac{3}{8}$ in.

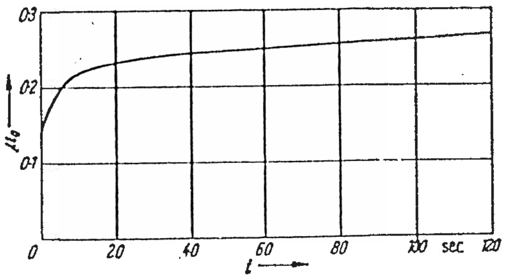


FIG. 385

obtained, which whilst not ensuring perfect working conditions makes them at least reasonable. It must be remembered in this connexion that under such conditions, the coefficient of static friction μ_0 depends upon the time interval between the latest supply of oil and the initiation of the movement. Under the weight of a stationary slide, the oil between the sliding surfaces is slowly squeezed out, so that with increasing time intervals t, the coefficient of static friction μ_0 increases (Fig. 385).[83] Under such conditions, the value of the frictional resistance is, therefore, not constant but depends upon the sliding speed and the time interval between the oil supply and the start of the working movement. Such a variation of the frictional resistance can sometimes give rise to greater difficulties than its absolute value.

Low friction resistance and constant friction conditions can be obtained by the application of anti-friction bearings (roller bearings, ball bearings, etc.), or by pressure lubrication of the slideways.

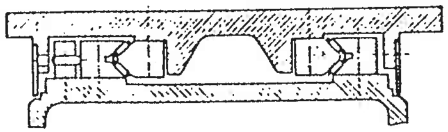


FIG. 386. Accurate setting made possible by roller-slides of a centreless grinding machine (Herminghausen).

(i) *Anti-friction bearings for slideways* have been used for some time in instrument technology, particularly when the loads to be carried are small. They have also been used in machine tools when very fine touch during setting operations was important and either the working loads were relatively small (e.g. grinding machines, Fig. 386),[91] or before any cutting forces were exerted, the full operational loads being taken by ordinary slideway surfaces (Fig. 387).

In the latter example, the freedom from play in the slideways is not critical, as the working accuracy is determined by the sliding surfaces and not by the roller bearings. If, however, the anti-friction roller bearing arrangement serves for transmitting the full working load during operations,

the requirements of accuracy must be fully satisfied and this can be obtained by adjustability, or preloading, or other arrangements. Roller bearing slideways can be divided into two groups,[92] i.e. slideways for limited traverses and slideways for unlimited traverses. An example of the former is the layout shown in Fig. 388, in which the rollers are normally held in a cage and traverse only

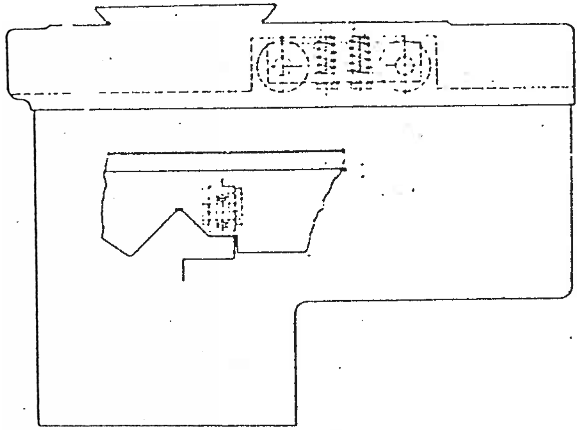


FIG. 387. Spring loaded rollers carry the lathe saddle, whilst the cutting force compresses the springs and is thus taken by the Vee-slide (Dean, Smith & Grace Ltd., Keighley).

half the distance which the slide B covers on the fixed slideway A. The cage strip C must be shorter than the fixed slideway by an amount equal to half the traverse l ($L_C = L_A - l/2$, Fig. 388a). If the slide B is in its centre position, the roller bearing cage C must be in the middle of the fixed slideway (distance $l/4$ from each end, Fig. 388b).

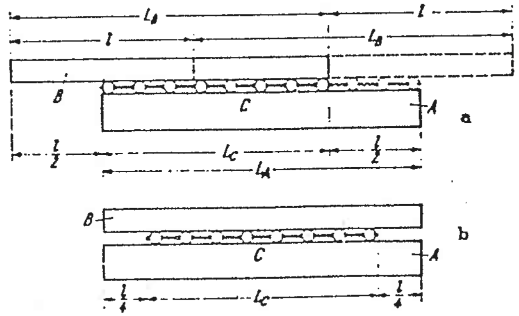


FIG. 388

Balls, needles, or, for higher load carrying capacity, rollers are used and these run between hardened guiding strips (H_R = 60 to 62 C) which are suitably shaped. Play can be eliminated by making one of the guiding strips adjustable. Open (Fig. 389), or closed arrangements (Figs. 390 and 391), are used. The ball bearing arrangements shown in Figs. 389a and 390 serve for light loading conditions when the supporting forces can be taken simultaneously in two directions at right angles to each other. In the case of needle or roller bearing arrangements, other steps are necessary, such as a setting of the roller axes at 45° to the direction of loading (Figs. 389b and 391). In such an

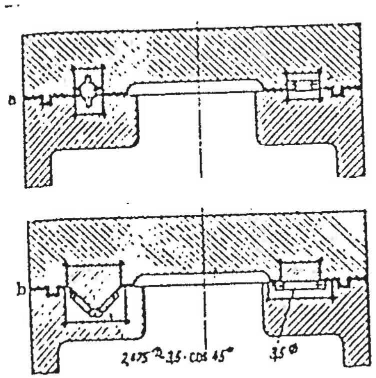


FIG. 389. Open arrangement. *a*) Ball and roller slide (Robert Kling, Wetzlar, Germany); (*b*) Needles of different diameters (Industrie Werke Schaeffler, nr. Nürnberg, Germany).

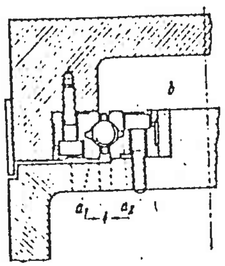


FIG. 390. Ball slide (W. Schneeberger A.G., Bern, Switzerland) a_1 and a_2—hardened and ground races; *b*—Adjusting strip.

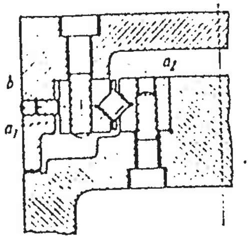
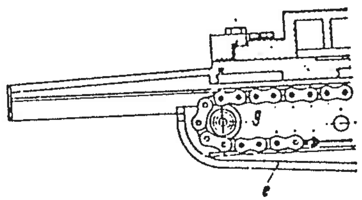


FIG. 391. Roller slide (W. Schneeberger A.G., Bern, Switzerland). a_1 and a_2—hardened and ground races; *b*—Adjusting screw.

arrangement the diameter of the needles can be less than that of needles loaded at right angles to their axes (see Fig. 389b).

Instead of using needles the axes of which lie at 90° to each other and in two different cage strips (Fig. 389b), it is also possible to employ rollers in a crossed arrangement and in one single cage strip (Fig. 391, see also Fig. 394).

If the stroke becomes too long compared with the length of the slide, either normal ball or roller bearings, rolling on hardened rails or recirculating elements, such as used in recirculating ball nuts (see page 148), can be applied. A combination of both ideas is shown in the arrangement (Fig. 392), where the guidance in the horizontal plane is provided by two bearing sets *a* and *b* and in the vertical plane by two recirculating sets of rollers *c* and *d*. The rollers *c* and *d* are carried in chains which act as recirculating cages *e* and *f* and are tightened by pulleys *g*. After their disengagement at one end of the sliding surface the rollers are thus guided to the other end, where they again enter the sliding surfaces and continue their function.



For fine adjustment or fitting, the ball bearings are located on eccentric pins. However, standard ball bearings provide only line contact between their outer races and the sliding surfaces and they are unable, therefore, to carry heavy loads. In order to make use of the relatively higher load carrying capacity of rollers, for a two-directional slideway, the SKF has developed a so-called cross roller chain which, differing from ordinary roller chains (Fig. 392), carries rollers whose axes lie alternately at 90° to each other (Fig. 393).[93] The rollers are guided over the full length of the slideway (not only at the point of reversal as in Fig. 392) on a guiding rail of adjustable length (Fig. 394) and this (*a*, Fig. 393)

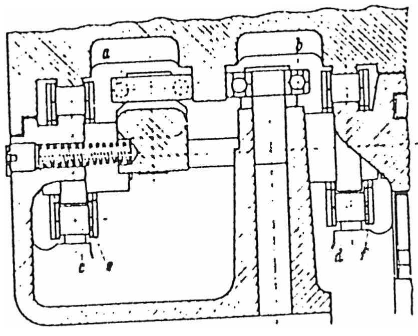


FIG. 392. Ball and roller bearing slides with precision roller chain (Ludw. Loewe and Co. A.G., Berlin).

the spool valve is controlled by the bearing pressure in such a manner that the ratio p_0/p_1 and with it the bearing gap (the oil film thickness *h*) is kept constant.

When spool valves, however, are used for controlling the oil film thickness, some undesirable characteristics arise which limit their application. These are spool stiction, leakage, especially when low viscosity fluids are employed, manufacturing problems, and slow dynamic response.

In order to overcome this difficulty, M. E. Moshin has developed a control device (Fig. 403a),

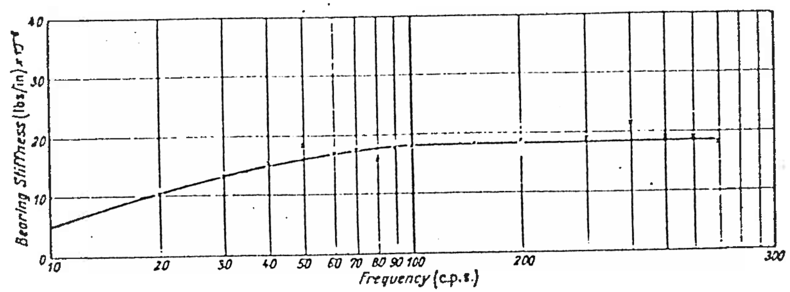


FIG. 403c.

which acts as a restrictor and controls the oil film thickness in the hydrostatic bearing. The resistance of the control device is determined by the deflexion of a diaphragm. This deflexion changes with the bearing pressure in such a manner as to permit just the desired amount of oil discharge thus ensuring a practically constant gap *h* of the bearing.

The lubricant flows at a constant supply pressure to the bearing through the gap of the circular restrictor *a*. This restrictor is located at the centre of a circular diaphragm *b*, which is rigidly fixed at its periphery. At atmospheric bearing pressure ($p = 0$), the gap of the restrictor is adjusted to a set value by means of the low stiffness spring *c*. If a load is applied to the bearing the pressure *p* will increase to p_1, deflect the diaphragm and thus increase the gap of the restrictor. This will result in an increased flow *Q*.

The design proportions of the restrictor can be chosen so that the oil film thickness *h* remains almost constant over a wide range of load values (see graph in the corner of Fig. 403a). Figure 403b shows the static and Fig. 403c the dynamic characteristic (oil film thickness *h* as a function of load *P*).[99]

3. SPINDLES AND SPINDLE BEARINGS

The main spindle serves for centring and holding the cutting tool (drilling, grinding, milling) or the workpiece (turning) under the effects of weights and cutting forces on the one hand and driving forces and torques on the other. It fulfils, therefore, two functions, i.e. it not only locates the tool or workpiece respectively but also drives and guides them with the required accuracy and stiffness in their operational movements (rotation and sometimes axial feed movement).

The centring elements usually arranged at the front end, the spindle nose, are either external or internal cylinders or tapers. When cylindrical centring devices are employed, thrust faces have to be provided in order to determine the axial position of the located part. The taper, which alone ensures backlash-free centring, determines also the axial position, although not positively because this depends upon the relative sizes of the male and female tapers.

Both the metric (exactly 1 : 20) and the morse tapers (approximately 1 : 20) are practically irreversible. They can transmit frictional torques up to a certain magnitude and are, in most cases, separated easily by an axial impact. Intermediate adaptor pieces can be used in order to fit tapers of various sizes into one spindle. Figures 404 and 405 show lathe spindle noses equipped with taper bores *1*, adaptors *2* and centres *3*.

A long taper is especially suited for the lathe spindle because the centre is loaded not only axially, but also radially (see Fig. 297). In drilling and boring machine spindles (Figs. 406 and 407), it is frequently found that the 1 : 20 taper is used for transmitting the torque to a limited extent. However, in drilling spindles, driving flats (*2*, Figs. 406 and 407) are added at the rear end of the taper *1* for light loading and special flats (*3*, Fig. 407) at the front end for heavier loads, in order to prevent loosening of the taper under the effect of large torques and possible vibrations. This is important because loosening or slipping in these tapers would lead to damage of the tapered surfaces, to inaccurate locating and possibly seizing. Relatively small tapers can be assembled securely by a light axial blow, whilst for larger tapers, special tightening devices (*a*, Fig. 407) are often provided

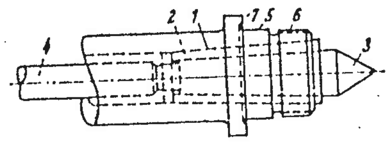


Fig. 404

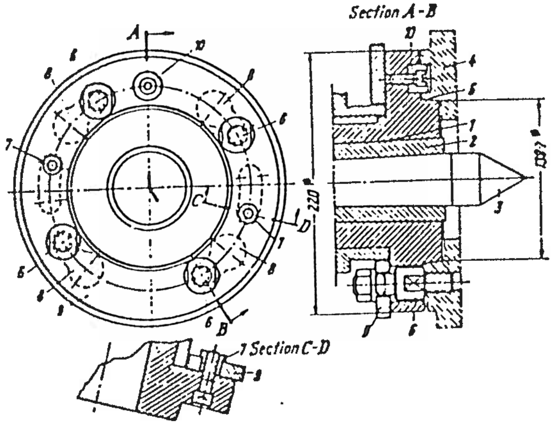


Fig. 405a. Spindle nose according to DIN 55022 with driving plate (Schaerer lathe).

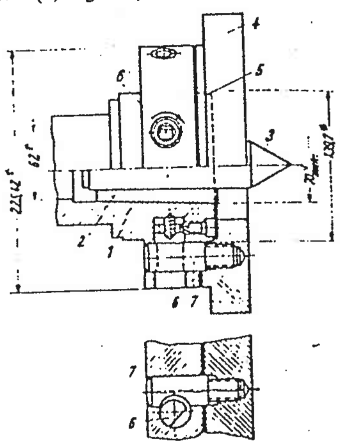


Fig. 405b. Spindle nose with short taper and "camlock" for the driving flange (Schaerer lathe).

by the designer. If the spindle is hollow, it is possible to loosen the taper by a sharp axial blow on a bar (*4*, Fig. 404). Otherwise wedges (*4*, Fig. 407), acting through slots (*2*, Figs. 406 and 407) are employed for this purpose.

Driving plates or chucks are often located on an external cylinder (*5*, Fig. 404), the driving plate

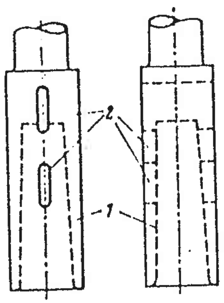


Fig. 406

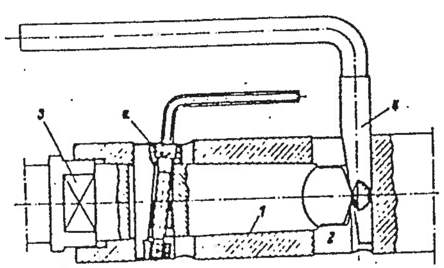


Fig. 407. Patented spindle nose of a horizontal boring machine (Collet & Engelhard, Offenbach, Germany).

or the chuck being held axially against a shoulder *7* by a screw thread *6*. It is important that the screw thread *6* does not interfere with the centring action of cylinder *5*. The screw thread must, therefore, be sloppy rather than too tight. Any play in the screw thread appears, in any case, always on one side only by virtue of the axial pressure against shoulder *7*. Loosening of the thread is, therefore, prevented. The centring by means of an external cylinder on a spindle nose cannot be without any play, and the fitting of a heavy chuck on a spindle nose with a tight cylinder fit and a screw thread is fatiguing and time-consuming.

For this reason, spindle noses of heavy lathes are frequently designed for locating and fastening the chucks by means of an external taper and a ring nut (see Fig. 424a). Figure 405a shows an external

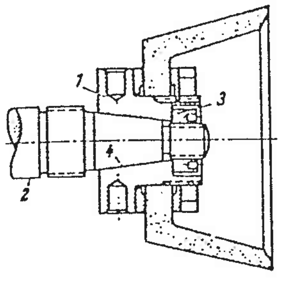


Fig. 408

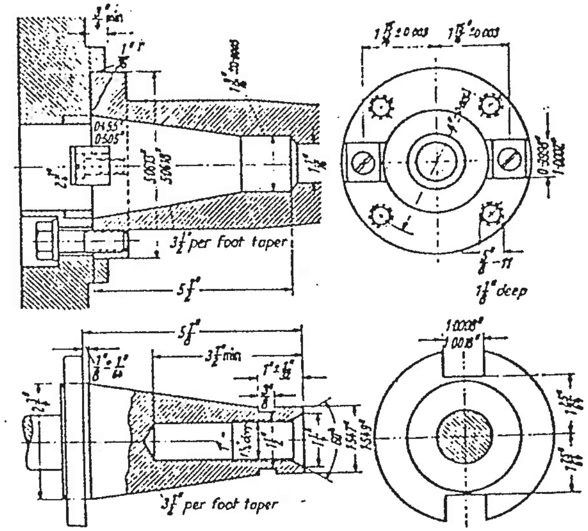


Fig. 409. Milling spindle taper.

taper location with bayonet fixing, in which the adaptor plate for the chuck or the driving plate *4* is located by the taper *5* and held axially by four bolts *6* in slots *8*. These bolts are held in the bayonet disc *9*, which is guided by two pegs *7*. The torque is transmitted by bush *10*.

An even more rapid fixing device for the adaptor plate *4* on the taper *5* is shown in Fig. 405b. Here, the plate *4* is held in position by the eccentric pegs *6*, which engage in semi-circular slots of the fastening bolts *7*.

Concentricity of location is of particular importance when grinding wheels are fitted to their spindles (Fig. 408, see also Fig. 433). The grinding wheel is fitted to an adaptor *1*, which is pressed on to the external taper *4* of the spindle *2* by means of a nut *3*, and thus located accurately and held tightly.

The location and fastening of milling arbors and cutter heads must be able to resist the pulsating and frequently high cutting forces. Even the shallow taper (1 in 20) could not possibly transmit the torques by friction alone. Moreover, such a taper is more liable to seizure than a steeper one. For this reason, the steep taper ($3\frac{1}{2}$ in/ft, Fig. 409), originally introduced in the U.S.A., has now been accepted generally and standardized for the internal centring in the milling spindle nose.

The torque is transmitted by means of two tenons which are fastened by screws to the spindle nose. As this taper is not irreversible, it is held axially in the spindle by means of a screw thread. The fastening bar which passes through the full length of the milling spindle (*1*, Fig. 410) is held by a collar *2*, against a shoulder *3* of the spindle bore. It can also be used for ejecting the taper by unscrewing it against the shoulder of collar *4*.

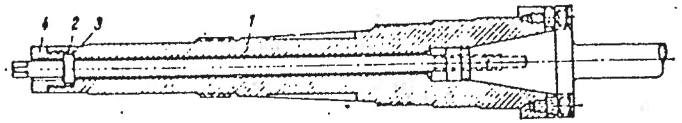


FIG. 410. Milling spindle.

If a spindle is very long, it may be cumbersome and time-consuming to fasten the arbor by means of draw bars and may even lead to accidents. The difficulty is avoided in the quick change device (Fig. 411),[109] developed by the Cincinnati Milling Machine Company, and consisting mainly of a

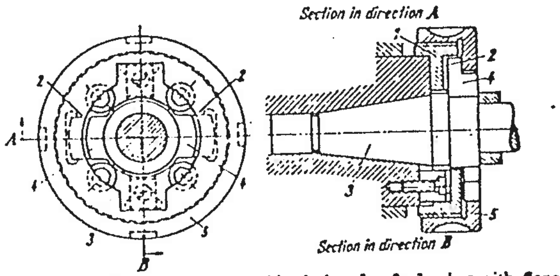


FIG. 411. Milling spindle nose. *1*—Driver with machined slots for *2*; *3* arbor with flange and driving tongue *4* machined from solid; *5*—Clamping nut.

holding flange and driving flaps machined from solid. This results, however, in rather high material and manufacturing costs. In a less costly version, pegs with machined driving faces are fitted to the arbor (Fig. 412).[100]

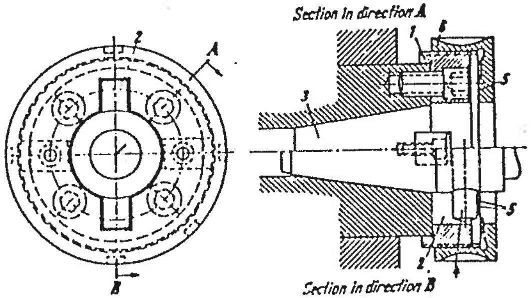


FIG. 412. Milling spindle nose. *1*—Driver with machined slots *2*; *3*—Arbor with driving peg *4*; *5*—Clamping faces; *6*—Clamping nut.

In a manner similar to that of the driving devices shown in Figs. 411 and 412, cutter heads are usually located on the external cylinder of the spindle nose and fastened axially to the front face by means of clamping screws (Fig. 413). The centring accuracy achieved in this manner is, however, frequently insufficient for the requirements of precision milling with large cutter heads, and for this reason internal centring (Fig. 414) is sometimes preferred. More rapid tool changes are possible

into the headstock (*a* and *b* respectively). The outer races of the two needle roller bearings (*c* at the front and *d* at the end) are carried in slotted tapered rings *e* and *f*. These can be axially displaced in the housing bushes *a* and *b*, thus providing the possibility of radial play adjustment.

Another method of play adjustment for needle roller bearings, which avoids the danger of distorting the races by the tapered adjusting rings, is the application of an elastic outer race whose internal diameter can be reduced by axial pressure (Fig. 429).[109] The use of an elastic outer race requires, however, a very stiff housing, as otherwise the stiffness of the arrangement as a whole would suffer.

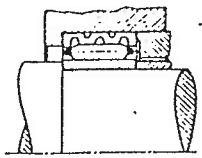


FIG. 429

(2) *Plain Bearings*[110]

Due to the great variation in working conditions encountered in spindle bearings, it is impossible in many plain bearings to avoid lubrication conditions in which metallic contact occurs between the moving parts. Under these conditions, the spindle climbs up the inner wall of the bearing in a direction opposite to its rotation and then drops from a height which is determined by the so-called sliding angle (Fig. 430a). With varying lubrication conditions, this angle changes and with it the displacement of the spindle axis, resulting in unstable running. Whilst the amount of this movement

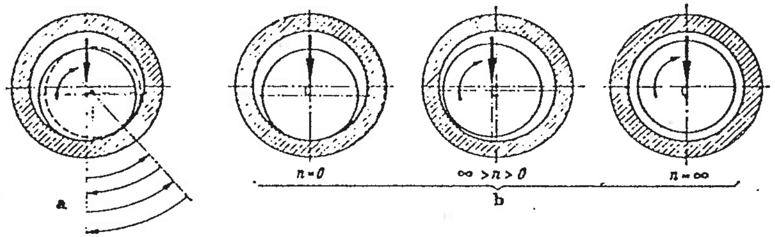


FIG. 430

can be kept within permissible limits by appropriately small play in the bearing, the change to hydrodynamic lubrication and vice versa causes changes of the friction resistance and of the direction of displacement (Fig. 430b) and this again results in unstable running. One of the main requirements for a plain bearing is, therefore, that the lubrication condition at a given speed should remain unchanged.

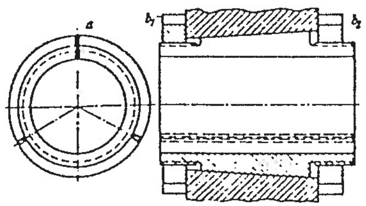


FIG. 431

The adjustment of the play in the bearing is therefore most important. Even to-day, bushes with cylindrical bores and external tapers are used for this purpose, the adjustability being obtained by the provision of slots (Fig. 431). Such adjustment can be carried out only by very experienced fitters if to-day's requirements are to be satisfied. When the play has to be reduced and the bush is pressed into the conical housing the segments are deformed and the originally circular bore takes on a triangular or other shape, according to the number of slots. This means that the bearing ought to be rescraped after each adjustment. It is also important to make certain that the bearing bush cannot "collapse" on to the spindle and clamp it. For this purpose, separating screws or shims (*a*, Fig. 431) made of leather or wood, are usually provided. It would appear advantageous to place the open slot at the top of the bearings, so that the lubricating oil cannot escape. However, the radial load of the bearing must not be directed against the slots. The adjusting nuts b_1 and b_2 at the ends of the tapered bush, which serve for axially displacing the bush, must be provided with rectangular threads, as otherwise radial forces are exerted on the male thread and these may make the bush collapse on to the spindle.

Bearing bushes without slots and with tapered bores are used for adjusting the play by axial displacement of the bush on a tapered spindle. This design is more expensive to manufacture and requires great care during adjustment, because of the danger of pressing the bush too far on to the spindle. However, the load carrying capacity of such bearings is higher than that of slotted bushes and their circular shape is ensured. Figure 432 shows such a bearing which is provided with a flooded

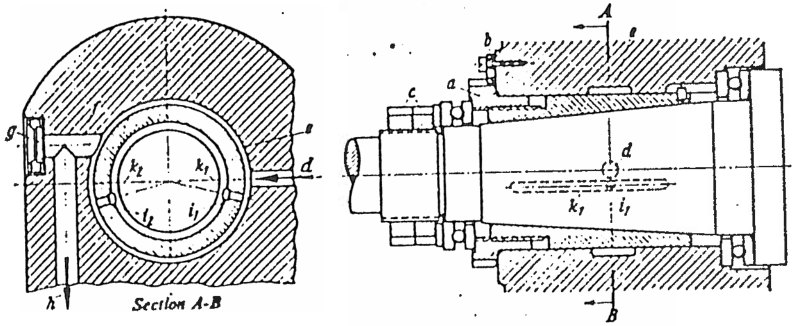
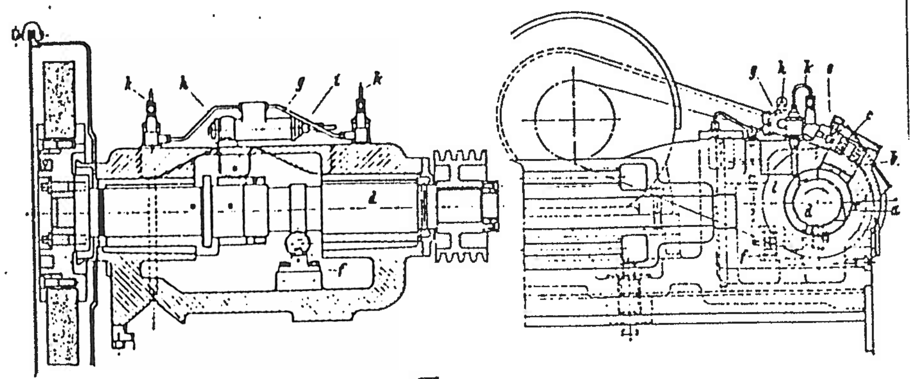


FIG. 432

lubrication and was used by the author in a heavy turret lathe. The ring nut a serves for adjusting the play and can be secured in any position by the bracket b. The adjustment of the radial play is independent of that for the thrust bearings which is carried out by the nut and lock nut c. An oil tank with a capacity of about 12 gal is arranged in the base of the machine and lubricating oil (about 4 gal/min) is delivered to the inlet d by a pump. This oil surrounds the bearings continuously (circular groove, e) and runs back to the tank through bores f and h. As a result, the oil level, which can be observed through the window g is kept constant. The oil reaches the bearing surfaces via



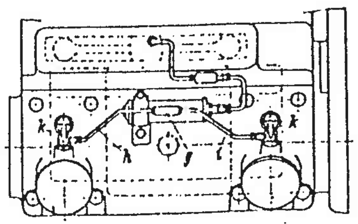


FIG. 433. "Hydrauto" grinding spindle bearing (The Churchill Machine Tool Co. Ltd., Manchester).

bores l_1 and l_2 in the quantities which are required at any moment and is distributed over the whole length of the bearing by grooves k_1 and k_2. According to the spindle speed, boundary or semi-hydrodynamic lubrication is obtained. At the same time, the continuous supply of fresh oil surrounding the bearing maintains the temperature almost constant.

A plain bearing with automatic play adjustment is the "Hydrauto" bearing (Fig. 433), in which a loose segment a is held against the spindle d with minimum possible play, by means of piston b and springs c. Oil is supplied, under pressure, via non-return valve e to the piston b, which thus prevents the lifting of the segment and with it the spindle. A pump f supplies the pressure oil for the bearing adjustment (pipeline k) via filter g and pipelines h and i and also the lubricating oil for the bearing (bore l).*

The growing requirements concerning the running quality of main spindles, the spindle speeds and the life of the bearing underline the necessity for using hydrodynamically or hydrostatically lubricated bearings, in which the position of the spindle axis changes very little or not at all. The design of hydrostatically lubricated bearings, which in principle work similarly to the hydrostatically lubricated slideways described earlier (see page 259) is still undergoing development. If single surface bearings (see Fig. 430b) are used, the position of the rotating shaft in the bearing varies with the speed and the transverse load. This positional change would not be permissible in machine tool spindles. The difficulty is avoided in the multi-surface bearing (Fig. 434), in which the oil pressure

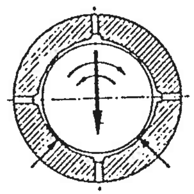


FIG. 434

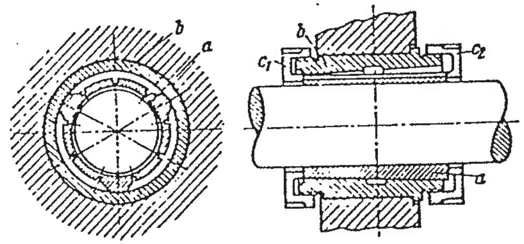


FIG. 435. Mackensen bearing.

is applied in four directions and thus holds the spindle in its central position.[111] Even if the play in the bearing is very small (0·0002 to 0·0004 in.), hydrodynamic lubrication is obtained over a wide speed range,[112] and the heating remains within permissible limits. If the spindle axis is horizontal and the weight of the spindle is the main part of the total bearing load, it is advantageous to design the bearing in such a manner that the load is taken by two surfaces which lie at angles of 45° to the direction of loading (see Fig. 434).

In order to ensure the small play which can be obtained with multi-surface bearings, it is usual to fit the shafts to the bearing bore by a lapping process. Under hydrodynamic conditions, metal contact and wear cannot occur and it is, therefore, not necessary to provide means for adjusting such a bearing once it has been correctly fitted. However, in some cases adjustability may be preferable to fitting by lapping, if only for economic reasons. The designer of the Mackensen bearing (Fig. 435) obtains the desired effect by allowing an elastic bearing bush a, whose elasticity is increased by the provision of nine longitudinal grooves distributed over the circumference, to deform into a triangular shape. As a result, three wedge-shaped oil pockets are generated, which initiate the formation of load carrying oil wedges. The radial position of the three supporting points of the bearing in the housing b, can be changed by axial displacement of the bush (ring nuts c_1 and c_2) in a tapered bore. This adjustment makes it possible to reduce the minimum play in the bearing down to about

* This bearing has proved very valuable for grinding operations, when the weights of the grinding spindle and the grinding wheel are considerably greater than the grinding force component acting in the opposite direction. In recent years, however, the requirements of productivity and working speeds have been greatly increased, demanding high surface quality even at maximum rates of metal removal. The Churchill Machine Tool Company considers that under these conditions, which result in considerably higher grinding forces, an undivided bearing bush is preferable, and for this reason, all Churchill grinding machines are today equipped with such bushes.