DESIGN PRINCIPLES OF METAL-CUTTING MACHINE TOOLS

by

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TO JOHN, MARIE AND CARLOS

PREFACE

THE technical progress and development of manufacturing processes, tools, materials, measuring equipment and control devices never stops. It enables designers, manufacturers and users of machine tools to aim at ever-increasing qualitative and quantitative efficiency of their machines.

In consequence, it is almost impossible for an observer to view a general picture which is static for any period of time, however short. A book describing design features of well-known types of machine tools, such as lathes, milling or drilling machines, could easily have only historic interest for the reader because the designs described would tend to be obsolete before the galley proofs left the printer.

The application of transfer lines to mass production has given rise to a trend in design away from universal machines towards special purpose combinations of constructional units. These units are mounted on suitable base plates, beds, etc., to form single-purpose machine sets which later can be dismantled and re-assembled in different combinations to meet particular requirements. In moving with this change of approach from the design of the machine tool as an indivisible whole to the design of detail units, the German V.D.I. has published recommendations for standardizing the main dimensions of some of the principal units (Fig. I).

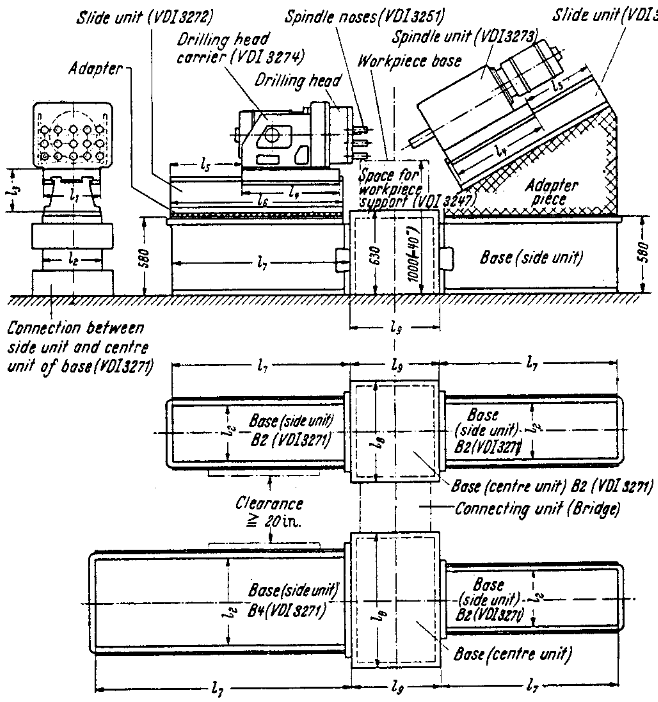
In view of such developments it appeared advantageous to discuss the basic principles and considerations that are applicable to the design of those constructional elements which form important parts of metal-cutting units, rather than to describe complete machine tools of present day design.

These constructional elements are not likely to be subjected to such extensive and radical change as whole machines. It may be observed, for example, that the control mechanism of the Brown and Sharpe automatic lathe is kinematically almost identical with that developed sixty years ago, although of course, detail developments of the design have led to ever-increasing speeds and overall accuracy.

General considerations concerning static and dynamic stiffness, operational speeds and their standardization, gearboxes, manual and automatic control, etc., are widely applicable and not limited to a particular type of machine tool. The same is also true for problems concerning the design of component units such as beds and columns, main spindles, slideways and bearings, drives for the cutting and feed movements and control devices, such design problems belonging exclusively to the field of machine tools. The machine tool designer has, of course, also to deal with questions of a more general technical character, problems of hydromechanics, kinematics, control engineering, electrical engineering, strength of materials, general machine design, etc. Examples of these are found in the calculation and design of clutches, gears, belt and chain drives, bearings, control mechanisms and servo-loops. Such problems of a general nature which are not specifically related to the design of machine tools should be studied from the numerous textbooks and papers that are available, many of which are indicated in the bibliography at the ends of Parts I and II.

This book discusses the fundamental principles involved in machine tool design and does so with the intention of arousing the reader's interest in more detailed study of specialized information. It is not the author's intention to spoon-feed the designer by giving complete instructions for a large number of calculations and specific design solutions.

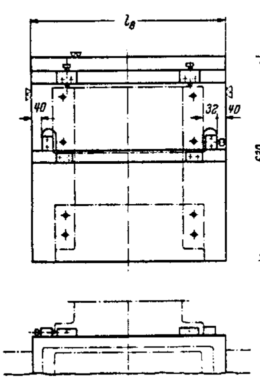
There are few design problems which cannot be solved after studying authoritative literature or calling on the advice of experts. However, a designer cannot take such action unless he is able to recognize the existence and the character of these various problems, and it is the lack of ability to do this which so often shows itself. For this reason, typical problems encountered in machine tool design are analysed and possible methods for their solution discussed. The field which it has been possible to cover within the limits of this book is, in consequence, wider than it would have been if numerous detailed descriptions of existing designs and calculations were given.



Main Dimensions

	l_1	l_2	l_3	l_4	l_5	l_6	l_7	l_8	l_9		
									a	b	c
B 1	320	340	280	560	400	1000	1050	630	450	630	800
B 2	400	420	320	710	500	1250	1300	710	450	630	800
B 3	500	520	320	900	500	1450	1500	850	450	630	800
B 4	630	650	350	1120	630	1800	1850	1000	450	630	800

FIG. Ia



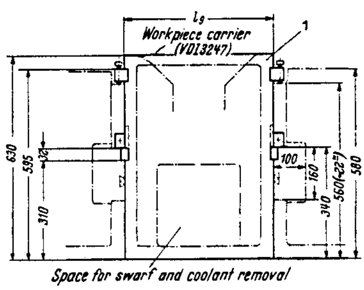


FIG. Ib. Centre units for base.

	l_8	l_9		
		a	b	c
B 1	630	450	630	800
B 2	710	450	630	800
B 3	850	450	630	800
B 4	1000	450	630	800

FIGS. Ia and b. Constructional units for machine tools (dimensions in mm).

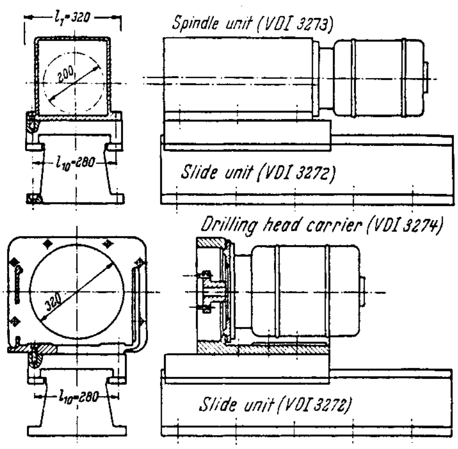


FIG. Ic. Location of spindle unit (VDI 3273) and drilling head carrier (VDI 3274) on the slide unit (VDI 3272).

a

b

c

d

e

f

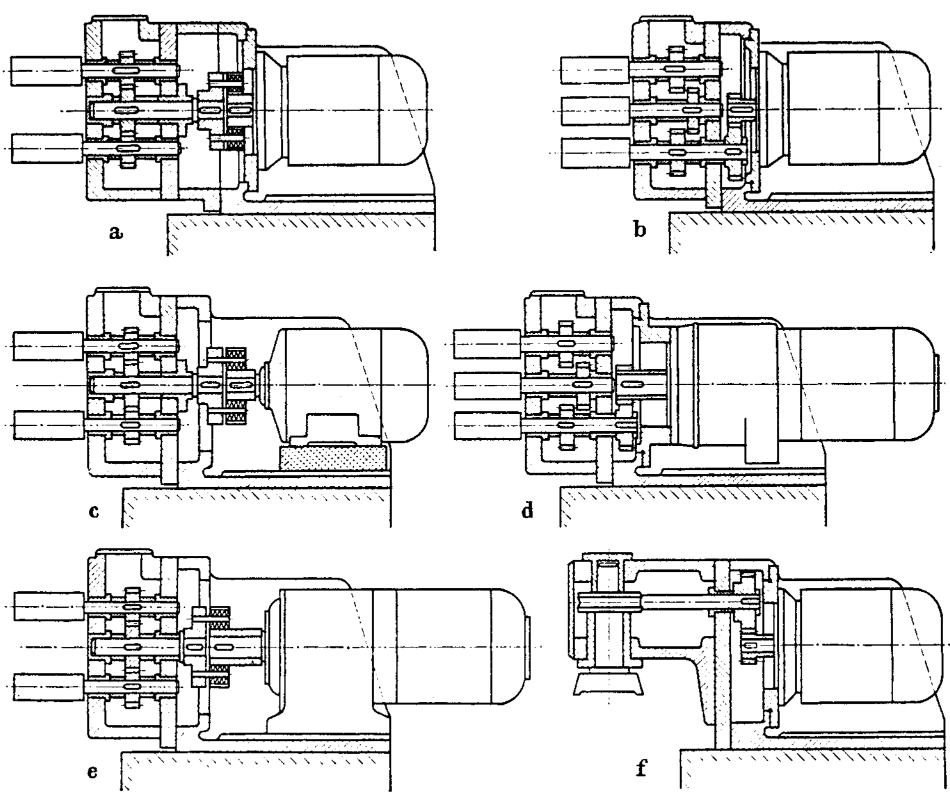


FIG. Id. Six different applications of the drilling head carrier (VDI 3274). Drive by (a) flange mounted motor and coupling; (b) flange mounted motor and spur gears; (c) base mounted motor and coupling; (d) geared flange mounted motor and spur gears; (e) geared base mounted motor and coupling; (f) milling head (special design).

FIGS. Ic and d. Constructional units for machine tools (dimensions in mm).

A certain minimal knowledge of the major machining processes (turning, milling, drilling, etc.), is necessary in order to appreciate the working conditions of a machine tool. A brief survey of these conditions and a discussion of the performance which the workshop engineer expects from a metal-cutting machine tool will be found in the introductory Part I of the book. A knowledge of the various types of machine tools which are used in the average machine shop is, however, assumed.

A designer cannot learn his job from books alone. Reading must be backed by practical design experience in industry. For the same reason, the problems discussed in this book have not been treated in complete isolation but with the backing of design solutions which have proved their success in practice. The author is, therefore, greatly indebted to the many machine tool manufacturers in various countries who have assisted him by putting at his disposal far more informative material, drawings, photographs, etc., than could be accommodated in the available space.

He would also like to thank his colleagues at The Manchester College of Science and Technology, especially Dr. M. M. Barash, Mr. J. P. Mabon, Dr. J. K. Royle and Dr. J. Skorecki, who generously helped him with their advice and assistance.

The author's interest and a great amount of his knowledge in the field of machine tools are due to his teacher, Dr.-Ing. G. Schlesinger, with whom he became closely connected, not only professionally but also personally. It is, therefore, with particular pleasure that he records his thanks to his mother-in-law, Mrs. Schlesinger, and to his wife for their indefatigable assistance in the preparation and checking of the original German script.

He would also like to acknowledge with grateful thanks the kind interest and advice of Professor Dr.-Ing., E.h. O. Kienzle, Hannover.

Finally, the author is indebted to his friend, J. H. Lamble, D.Sc., for his advice and invaluable help in preparing the English edition of the book, and last but not least to Miss Sheila Larkin and Miss Jeanne Vigurs who prepared the script.

F. KOENIGSBERGER

PART I

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).

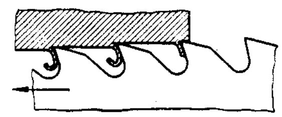


FIG. 1

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.

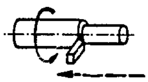
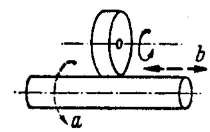
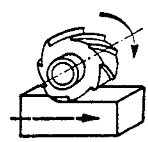
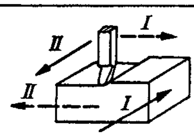
In general, a hyperbolic relationship exists between the *cutting speed* (v), which is the relative velocity of the cutting movement between cutting edge and workpiece, and the tool life (T) between re-grinds (Fig. 2). Taylor has expressed this relationship by the equation:

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

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.

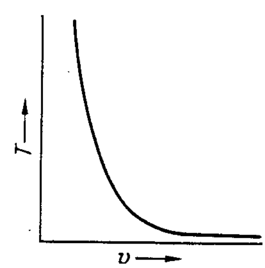


FIG. 2

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.

and for this reason, they have been used in the considerations which follow. It should not be difficult to apply more up-to-date values in the same manner to calculation and design work, as soon as such values become available.

Any new values must also cover cutting processes in which very high cutting speeds and feeds are applicable.* Cutting speeds of 1200–2500 ft/min are possible at the present time for the machining of steel with ceramic tools[4], and the possible increase in cutting capacity, which arises from these high cutting speeds at equal tool life, also results in greater accuracies in the workpiece because of the greatly reduced tool wear.

Research work concerned with the properties of ceramic tool materials, optimum conditions of use and power requirements are in progress all over the world.† Difficulties are encountered, especially during milling operations, due to their relatively low resistance to impact which has not only mechanical but also thermic causes.[5] In this respect the stiffness of the machine tool can have an immediate influence upon the performance of the cutting tools.

1. TURNING

(a) Cutting Force

The cutting force (P) which acts upon the tool is conveniently resolved into components in the following three directions (Fig. 3):

(1) Tangential to the turned surface, and at right angles to the axis of turning, i.e. in the direction of the cutting speed. This is the main component P_1 (lb), which together with the cutting speed v (ft/min) determines the net power required for the main spindle drive.

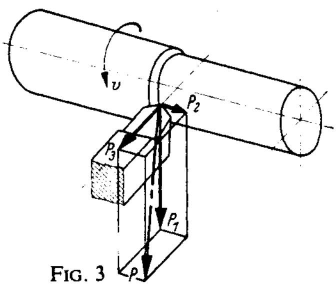


FIG. 3

(2) Parallel to the axis of turning, i.e. in the direction of the feed movement. This is the feed force P_2 which together with the feed velocity determines the net power required for the feed drive.

(3) Radial to the turned surface, i.e. in the direction of the depth setting movement. This component is called P_3.

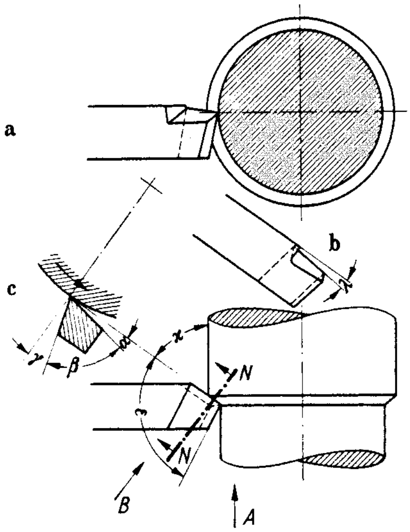


FIG. 4. Cutting angles of the turning tool.
a—view in direction of arrow "A";
b—view in direction of arrow "B";
c—section "N–N";
d—plan view;
α—clearance angle;
β—wedge angle;
γ—top rake;
ε—plan angle;
κ—approach angle;
λ—angle of obliquity (back rake).

As long ago as 1903, Nicholson[6] observed that even during simple turning operations the cutting force is not constant but pulsating. This is due to the elasticity of the tool, the workpiece and the machine, to the resulting changes in depth of cut, the effective rake angles, the relative velocity between tool and workpiece and to the chip formation itself.[7] A hard spot in the workpiece material, for instance, can produce elastic deformations in tool, workpiece and machine and thus initiate a vibration.

* See publications in *Stanki i Instrument* and books by Mozhaer & Saromotina and by Reznikow, all published in Russia.

† See for instance the publications of the Institute for Machine Tools, Technische Hochschule Aachen (Prof. H. Opitz).

Although the amplitudes of such forced vibrations can reach considerable values,[8] it is usual for the designer to assume a constant cutting force during turning operations, especially as far as the calculation of power requirements for the driving elements is concerned. It may, however, be important for him to know the frequency of the force pulsations, because resonance phenomena in the machine must be avoided at all costs. Kronenberg[1] quotes from his own experience, and from the work of other research workers, "frequencies from 80 to 200 c/sec (high speed steel tools), from 100 to 2000 c/sec (tungsten carbide tools), during the machining of steel and from 600 to 5000 c/sec during the machining of bronze".

The following parameters have an influence upon the magnitude of the cutting force and its components:

(1) The properties of the workpiece material,
(2) The size and shape (depth of cut and feed rate) of the chip section,
(3) The shape of the tool edges, especially the rake angles (Fig. 4), and to a limited extent the tool material,
(4) The cutting speed.

These will now be discussed in more detail.

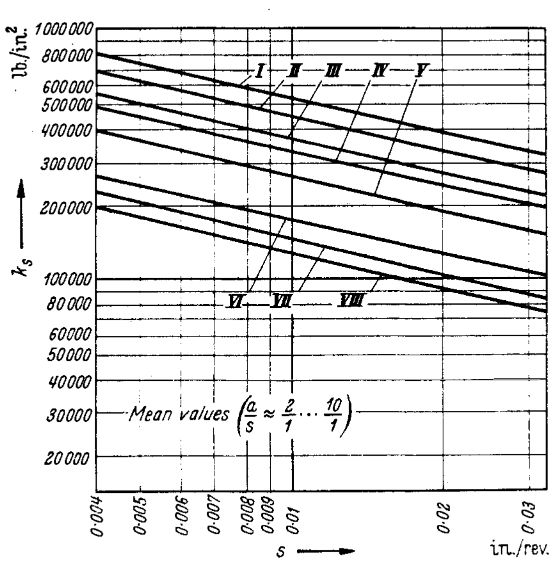


FIG. 5a. Specific cutting resistance values.
I Alloy steel up to 115 tons/in².
II Alloy steel up to 64 tons/in².
III Carbon steel up to 38 tons/in².
IV Cast steel and carbon steel up to 32 tons/in².

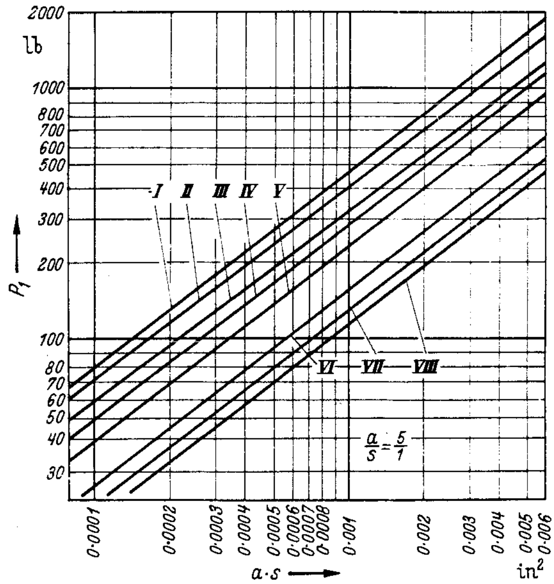


FIG. 5b. Cutting force components P_1.
V Cast iron 200–250 Brinell.
VI Cast iron up to 200 Brinell.
VII Brass 80–120 Brinell.
VIII Aluminium alloy.

(1) The specific cutting resistance k_s, i.e. the cutting force per unit chip section (lb/in²) is often specified as a material parameter. Although this value is not constant but depends upon the chip section, calculations can be carried out with the aid of approximate values.[1] Thus the order of magnitude of the specific cutting resistance is often indicated as a linear function of the tensile strength, and values of three to five times the U.T.S. are sometimes quoted.

(2) The specific cutting resistance decreases with increasing chip section ($a \times s$). It also increases slightly, however, with the ratio a/s, when the effective length of the cutting edge grows and the thickness of the chip decreases. For values of a/s between 2 and 10 typical values of specific cutting resistances for different materials can be shown as a function of the feed rate (Fig. 5a) and from these the corresponding values of the main force component (P_1) can be determined (Fig. 5b).*

(3) The cutting force components (P_1, P_2, P_3) and their ratios ($P_1 : P_2 : P_3$) are influenced by

* After Betriebsblatt AWF 158 (see p. 2).

the cutting angles. For a given chip section ($a \times s$) an increasing approach angle κ results in a growing chip thickness (h) and a decreasing chip width (b) (Fig. 6); in other words, the chips become shorter and thicker. For this reason, the specific cutting resistance, and with it the main cutting force component (P_1) and the radial component (P_3), decrease.[9] However, the feed force component (P_2) increases slightly (Fig. 7).[10] With increasing rake angle γ the cutting force decreases but the heat conduction deteriorates. If the angles κ and γ are too small, chatter results. Moreover, with decreasing angle γ the load carrying capacity of the tool increases more rapidly than the cutting force.

The use of negative rake tools is often advantageous, especially for the turning of materials which are difficult to machine. In this case, however, the power requirements are considerably greater and the design of the driving elements of the machine as a whole must satisfy the more severe stiffness requirements.

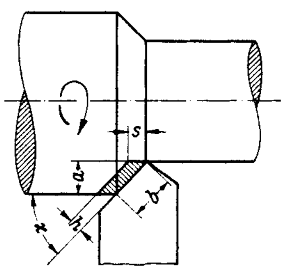


FIG. 6

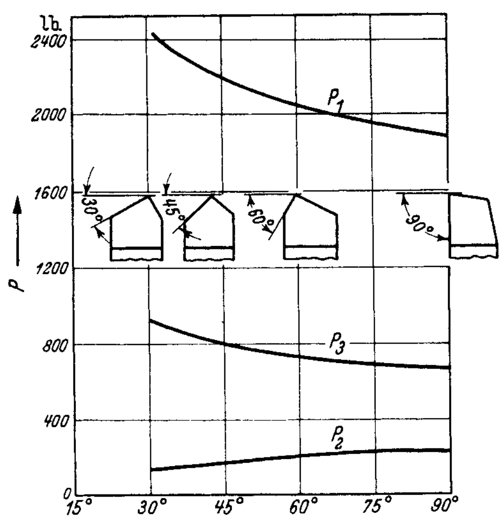


FIG. 7

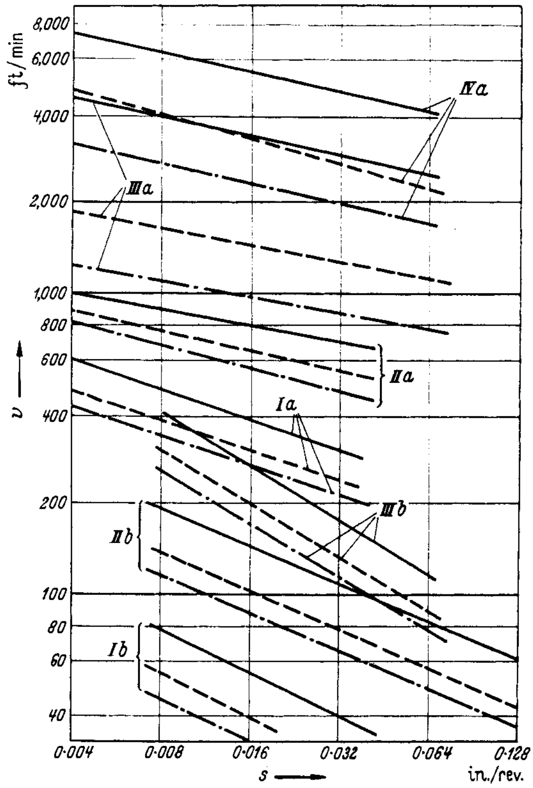


FIG. 8. Recommended cutting speeds.
—— v_{60}; – – – v_{240}; — · — v_{480}
a Carbide tools.
b High speed steel tools.
I Alloy steel up to 64 tons/in². III Brass.
II Carbon steel up to 32 tons/in². IV Aluminium.

The radial component (P_3) can be assumed as $0{\cdot}3P_1$. The feed component (P_2), however, depends to a greater extent upon the various cutting conditions and may vary between $0{\cdot}15P_1$ and $0{\cdot}50P_1$.

(4) Below cutting speeds of about 250 ft/min the cutting force decreases with increasing cutting speed. Above 250 ft/min (or when using carbide tools 300 ft/min), the cutting force is approximately constant over a wide range of speeds.

(b) Cutting Speed and Feed Rate

In order to obtain full economic efficiency the recommended cutting speeds for working with high-speed steel tools are usually based on a tool life of 60 min whilst those recommended for working with carbide tools are based on a tool life of 240 min. For turning operations on capstan, turret and automatic lathes, it is advantageous to work with tool lives of 240 or 480 min.

Figure 8 shows recommended cutting speeds as a function of the feed rate. The depth of cut has relatively little influence on the recommended cutting speeds as long as a is smaller than 0·2 in. For higher values of a it is advisable to reduce the recommended values by from 10 to 20 per cent. The cutting speeds mentioned in Fig. 8 are valid for the case where the approach angle $\kappa = 45°$ (see Fig. 7). If $\kappa > 45°$, the recommended cutting speeds can be obtained by multiplying these by a coefficient taken from Table 2.*

TABLE 2

CONVERSION TABLE FOR CUTTING SPEEDS WITH DIFFERENT APPROACH ANGLES

	Conversion Factor for v_{60}, v_{240}, v_{480} with an Approach Angle of		
	45°	60°	90°
High Speed Steel Tool Machining Steel and Steel Castings	1·0	0·80	0·66
High Speed Steel Tool Machining Cast Iron	1·0	0·89	0·72
High Speed Steel Tool Machining Other Materials	1·0	0·96	0·90
Carbide Tool Machining all Materials	1·0	0·96	0·90

2. DRILLING

In general, a twist drill is used. Facing tools, reamers and taps, which carry out a cutting and feed movement in the same manner as the twist drill, must also be considered.

The cutting action at the main edges (a, Figs. 9 and 10), is similar to that encountered in

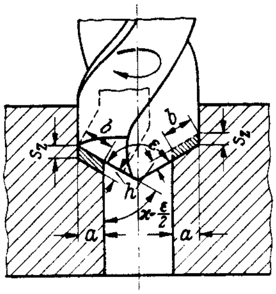


FIG. 9

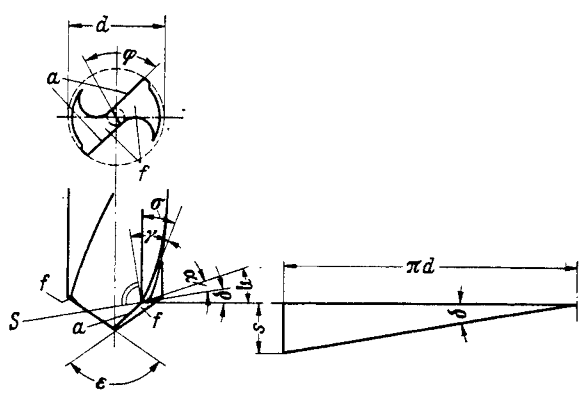


FIG. 10

turning.[9] The rake of the twist drill is determined by the helix angle. As this decreases towards the centre of the drill, the rake angle decreases equally from the circumference of the drill towards the centre. The effective rake angle γ depends also upon the direction of cutting (angle ρ, Fig. 10) where $\tan \rho = s/\pi d$. In this equation, s is the feed per revolution and d the diameter of the drill. The clearance angle α is further influenced by the relief angle η. Angle κ of the turning tool corresponds to angle ε of the twist drill $\varepsilon = 2\kappa$.

The intersecting line of the two relieved free surfaces (f, see Fig. 10) (the relief angle grows from the circumference of the drill towards its centre from about 6° to 20°) is the cross edge, which is

* After AWF 158 (see p. 2).

inclined by about 55°–60° to the main cutting edges. Its length depends upon the thickness of the drill core. For reasons of strength, this is increased from the point of the drill towards the shank by about 10 per cent. As the axial thrust of the drill increases with growing core thickness, it is important to point the drill after it has been shortened by frequent grinding.

The helix angle of a twist drill cannot be changed by tool grinding operations. Moreover, the shape of the helical groove is usually designed to suit the helix angle in such a manner that the main cutting edge, i.e. the intersection between the relief face and the top face, is a straight line. Although it would be possible, therefore, to change the point angle of the drill by suitable grinding, this might also result in a change of the shape of the main cutting edge. In order to use the correct rake angles, therefore, it is necessary to choose drills which have been designed to suit the machining of different materials. There is no universal twist drill for all materials!

(a) Cutting Forces

The cutting forces are conveniently resolved into a torque (T) and an axial thrust (P). If the drill does not cut symmetrically, for instance due to bad grinding, an additional radial component may exist, which results in unfavourable stressing of the machine and especially the spindle bearings. The magnitude of the cutting forces depends upon:

(1) Properties of the material to be drilled,
(2) Shape of drill (diameter, rake angles, length and positions of the cross edge),
(3) Chip section (drill diameter, feed rate),
(4) Cutting conditions (depth of drilled hole, coolant).

Consider (1). As the effective rake angle varies over the length of the cutting edge it is difficult to state approximate specific cutting resistances for different materials without knowing the drill diameter, because the specific cutting resistance depends upon the rake angle.

Now turn to (2) *and* (3). The chip section is determined by the drill diameter and the feed rate $d \times s$. The forces decrease with growing helix angle. The axial thrust decreases with decreasing angle ε, whilst the torque increases because at equal diameter the length of the cutting edge grows, and the chips are thinner at equal feed rate. The smallest forces occur if the cross edge lies at an angle between 55 and 60°. The influence of the relief angle upon the forces is negligible.

By suitable "pointing" of the cross edge it is possible to reduce the axial thrust by about 33 per cent with a simultaneous slight decrease of the torque. Figures 11 and 12 show values of torque and axial thrust when drilling several materials under different cutting conditions.

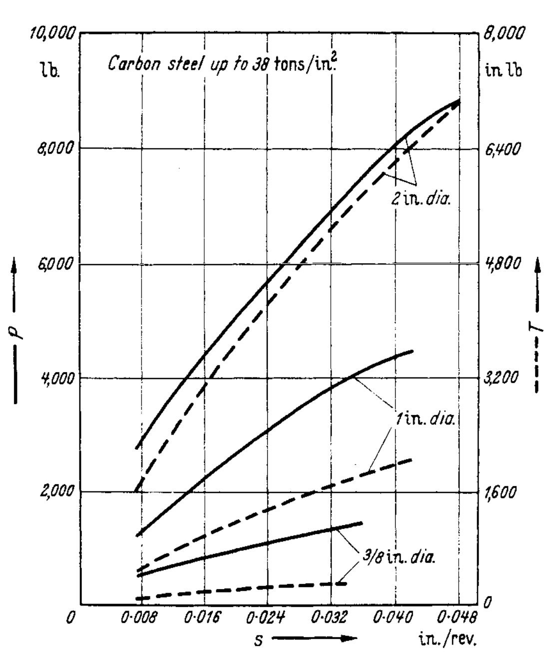


FIG. 11. Torque (T) and axial thrust (P) when drilling carbon steel up to 38 tons/in².

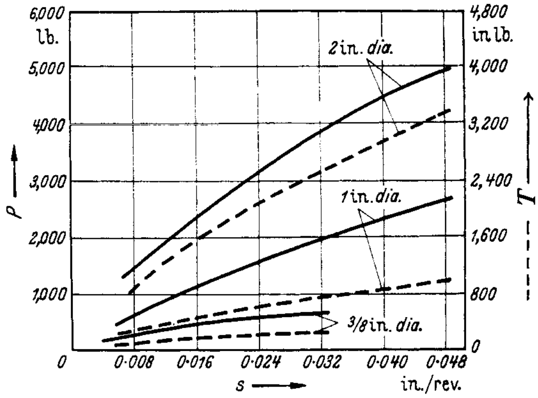


FIG. 12. Torque (T) and axial thrust (P) when drilling cast iron.

The axial thrust grows in proportion to the drill diameter, the torque with the square of the drill diameter. The increase of the forces as a function of the feed rate is, however, less than linear. If a large hole has to be drilled in steps of several smaller ones, it is better, from the point of view of load distribution, to use small diameter steps and high feed rates rather than vice versa.

Finally, consider (4). The friction at the outside diameter of the bore as well as the requirements of chip removal add to the cutting resistance, consequently the magnitudes of the forces and torques are affected by the depth of the hole and by the efficiency of cooling and lubrication during cutting (Fig. 13).

The forces which occur during facing, reaming and tapping are relatively small compared with those encountered when drilling into solid material with a twist drill, and for this reason the latter is the principal one which has to be considered by the drilling machine designer. As for operational speeds, the need for relatively low cutting speeds for reaming and tapping must be borne in mind (see Tables 5 and 6).

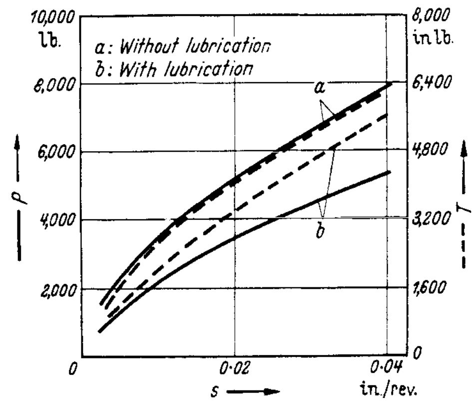


FIG. 13. Torque (T) and axial thrust (P) when drilling carbon steel up to 38 tons/in^2. with a 2 in. drill, with and without lubrication.

(b) Cutting Speed and Feed Rate

The cutting speed increases towards the outside of the drill, so that the extreme corner of the cutting edge is most liable to be damaged, and it is there that the drill loses its cutting ability first. For economic reasons long tool life in a twist drill is important. This depends not only upon the cutting speed but also upon the depth of the hole to be drilled (conduction of frictional heat, chip removal), the drill diameter (larger drills have longer life), the feed rate and the material properties.

Table 3 shows recommended cutting speeds and feed rates for different materials and drill diameters. Tables 4–6 show corresponding values for facing, reaming and tapping.

TABLE 3

CUTTING SPEEDS AND FEEDS FOR DRILLING INTO SOLID MATERIAL WITH HIGH SPEED STEEL TOOLS*

Workpiece Material	Cutting Speed ft/min†	Feed (in/rev) Drill Diameter (in)											
		0·2	0·25	0·32	0·4	0·5	0·63	0·8	1·0	1·25	1·6	2·0	2·5
Cast Iron	90–60	0·006	0·007	0·008	0·009	0·010	0·011	0·013	0·014	0·016	0·018	0·020	0·022
Carbon Steel up to 45 tons/in^2	90–80	0·004	0·005	0·005	0·006	0·007	0·008	0·009	0·010	0·011	0·013	0·014	0·016
Brass, Bronze	180–120	0·005	0·006	0·006	0·007	0·008	0·009	0·010	0·011	0·013	0·014	0·016	0·018
Light Alloy	500–400	0·006	0·007	0·008	0·009	0·010	0·011	0·013	0·014	0·016	0·018	0·020	0·022

* From DUBBEL, *Taschenbuch für den Maschinenbau*, 11th and 12th Editions, Springer 1956, 1961.
† Higher values for smaller, lower values for larger drill diameters.

TABLE 4

CUTTING SPEEDS AND FEEDS FOR FACING WITH HIGH SPEED STEEL TOOLS*

Workpiece Material	Cutting Speed ft/min†	Feed (in/rev) Tool Diameter (in)											
		0·2	0·25	0·32	0·4	0·5	0·63	0·8	1·0	1·25	1·6	2·0	2·5
Cast Iron	63–50	0·010	0·011	0·011	0·013	0·013	0·014	0·014	0·016	0·016	0·018	0·018	0·020
Carbon Steel up to 45 tons/in^2	70	0·014	0·014	0·016	0·016	0·018	0·018	0·020	0·020	0·022	0·022	0·025	0·025
Brass, Bronze	100	0·014	0·014	0·016	0·016	0·018	0·018	0·020	0·020	0·022	0·022	0·025	0·025
Light Alloy	250–200	0·010	0·011	0·011	0·013	0·013	0·014	0·014	0·016	0·016	0·018	0·018	0·020

* From DUBBEL, *Taschenbuch für den Maschinenbau*, 11th and 12th Editions, Springer 1956, 1961.
† Higher values for smaller, lower values for larger tool diameters.

TABLE 5

CUTTING SPEEDS AND FEEDS FOR REAMING WITH HIGH SPEED STEEL REAMERS*

Workpiece Material	Cutting Speed ft/min†	Feed (in/rev) Tool Diameter (in.)											
		0·2	0·25	0·32	0·4	0·5	0·63	0·8	1·0	1·25	1·6	2·0	2·5
Cast Iron	40–32	0·032	0·032	0·036	0·036	0·040	0·040	0·044	0·044	0·050	0·050	0·056	0·056
Carbon Steel up to 45 tons/in^2	25–20	0·016	0·018	0·020	0·022	0·025	0·028	0·032	0·036	0·040	0·044	0·050	0·056
Brass, Bronze	45	0·032	0·032	0·036	0·036	0·040	0·040	0·044	0·044	0·050	0·050	0·056	0·056
Light Alloy	80	0·032	0·032	0·036	0·036	0·040	0·040	0·044	0·044	0·050	0·050	0·056	0·056

* From DUBBEL, *Taschenbuch für den Maschinenbau*, 11th and 12th Editions, Springer 1956, 1961.
† Higher values for smaller, lower values for larger tool diameters.

TABLE 6

CUTTING SPEEDS FOR TAPPING WITH HIGH SPEED STEEL TAPS*

Workpiece Material	Cutting Speed (ft/min)	
	Small tap diameters	Large tap diameters
Cast Iron	30	20
Carbon Steel up to 45 tons/in^2	25	20
Brass, Bronze	45	32
Light Alloy	80	60

* From DUBBEL, *Taschenbuch für den Maschinenbau*, 11th Edition, Springer, 1956.

3. MILLING

A typical feature of the milling process is the fact that the rotating tool (the milling cutter) has a number of cutting edges each of which works over only part of its rotary path and travels over the remainder without cutting. The machine tool designer must always bear in mind the implications of these conditions, which concern the pulsations of the cutting forces, the vibrations of tool, workpiece and machine, the quality of the surface produced, etc.

The axis of rotation of the milling cutter remains usually stationary and the feed movement is carried out by the workpiece.

When the cutting edges are arranged on the circumference of the milling cutter, they describe cycloidal surfaces (Fig. 14) in relation to the workpiece, when they are arranged on the cutter face,

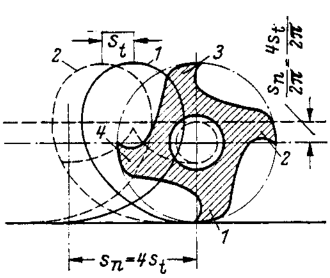


FIG. 14
s_n = Feed per rev.
s_t = Feed per tooth.

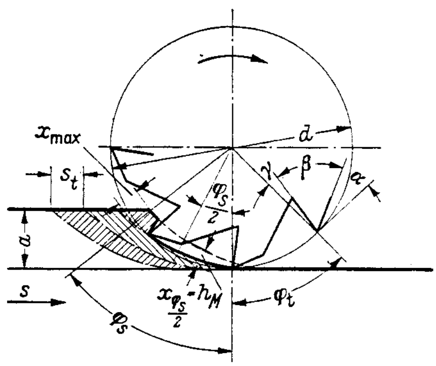


FIG. 15

they describe plain surfaces. If a surface is produced by edges which are arranged on the cutter circumference, whether straight or helical, the process is called slab milling. If the cutting process is carried out by cutting edges which are arranged on the cutter face, the process is called face milling.

(a) Chip Section

The thickness of the chip removed by each tooth varies during the cut approximately in the shape of a stylized comma (Fig. 15). It must be stressed that the "comma" is not the sectional area resisting the cutting force, but the boundary of the thickness, called the "chip thickness", as it changes during the cut. When a straight tooth cutter is used, the width of cut is constant (Fig. 16). In the case of a helical cutter the instantaneous chip thickness varies along the cutting edge, at least during part of the cut (Fig. 17). Under certain conditions, however, the total width of cutting,

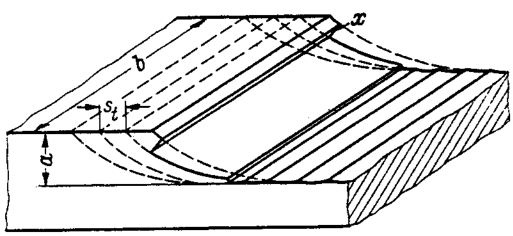


FIG. 16

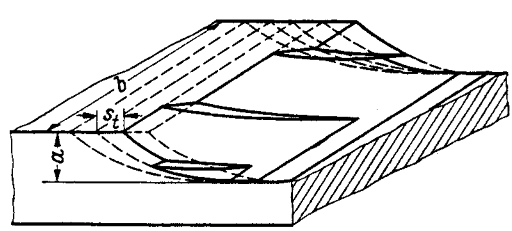


FIG. 17

i.e. the sum of the widths of cuts of all those teeth which are simultaneously in action, may remain constant.

The cutting conditions (depth of cut, number of teeth, etc.) determine whether each tooth may complete its action before the next one comes into operation, or that the action of several teeth may become superimposed. The magnitude of the instantaneous total chip section and the corresponding periodic variations of the cutting resistance depend, therefore, upon the cutting conditions.

The angle of rotation of a cutter between the entry of two successive teeth into the workpiece material can be assumed to be equal to the pitch angle of the cutter teeth $\varphi_t = 360°/n_t$, (see Fig. 15), where n_t is the number of teeth of the cutter. The angle of rotation of the cutter during which each tooth remains engaged in the workpiece material depends upon the cutter diameter d and the depth of cut a:

$$\cos \varphi_s = \frac{d/2 - a}{d/2} = 1 - \frac{2a}{d}$$

(b) Cutting Force

Although the chip section and with it the cutting resistance rises to a maximum and drops sharply as a straight cutting edge leaves the material (Fig. 18), the maximum may be maintained over a certain period if a helical tooth cutter is used, because the cutting action travels in this case parallel to the axis of the cutter and across the workpiece material (see Fig. 17). In this case the cutting force changes as a function of the cutter rotation in accordance with the curves shown in Figs. 19 and 20. The total resulting cutting force can then be resolved into an alternating component superimposed on a steady component. It is also possible to choose the cutting conditions in such a manner that the resulting total cutting force is practically constant (Fig. 21).

The foregoing shows that the load conditions in a milling machine depend not only on the width and depth of cut, but also on the size and type of cutter used, as well as upon the cutting speed and feed rate applied in a particular operation.

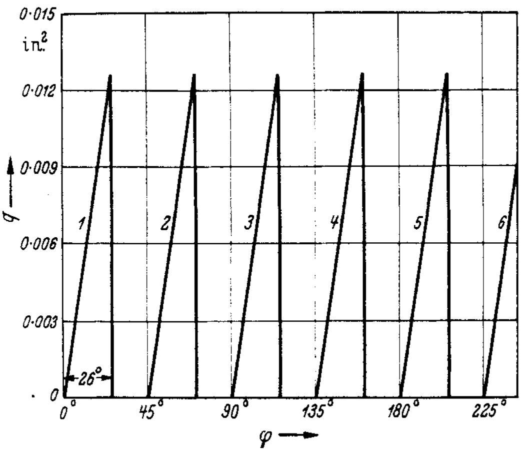


FIG. 18. Chip section during milling (straight tooth cutter).

Cutter diameter	$d = 4$ in.
Number of teeth	$n_t = 8$.
Cutting speed	$v = 60$ ft/min.
Feed rate	$s = 4$ in./min.
Depth of cut	$a = 0{\cdot}2$ in.
Width of cut	$b = 4$ in.

FIG. 19

Cutter diameter	$d = 4$ in.
Number of teeth	$n_t = 4$.
Helix angle	$\delta = 45°$.
Cutting speed	$v = 60$ ft./min.
Feed rate	$s = 4$ in./min.
Depth of cut	$a = 0{\cdot}2$ in.
Width of cut	$b = 4$ in.

ABCD ——— Tooth 1;
EFGH — — — Tooth 2;
IK, etc. — · — · — Tooth 3;
——— Total chip section.

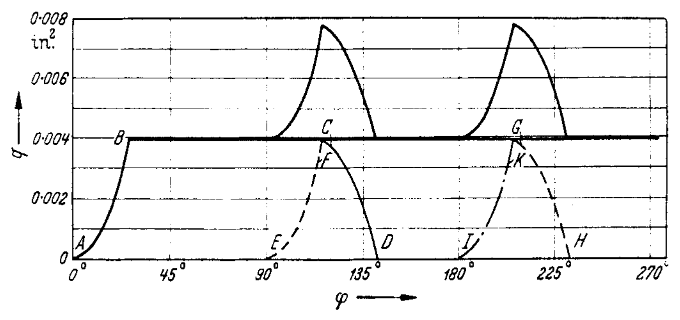
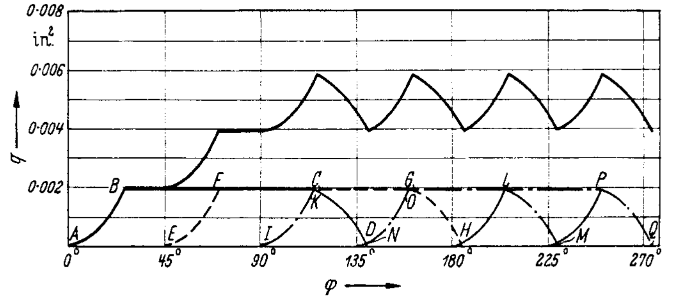


FIG. 20

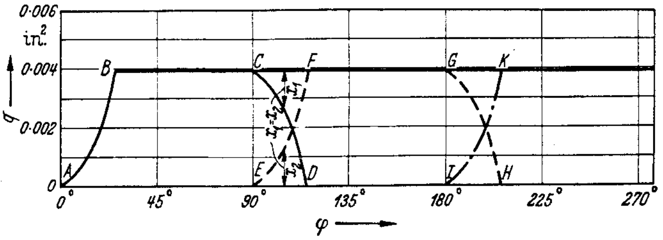
$d = 4$ in. $s = 4$ in./min.
$n_t = 8$. $a = 0{\cdot}2$ in.
$\delta = 45°$. $b = 4$ in.
$v = 60$ ft/min.

ABCD ——— Tooth 1; *EFGH* — — — Tooth 2; *IKLM* — · — · — Tooth 3; *NOPQ* — · — Tooth 4, etc. ——— Total chip section.



FIGS. 19 and 20. Chip sections during milling (helical cutters).

The cutting resistance depends upon the properties of the workpiece material, the shape and size of the cutter and the cutter teeth, the width and depth of cut, and the feed rate (per tooth or per revolution of the cutter).



ABCD ——— Tooth 1; *EFGH* - - - - Tooth 2; *IK*, etc. —·—·— Tooth 3; ——— Total chip section.

FIG. 21. Chip sections during milling (helical cutters).

When the cutter tooth has entered the workpiece material by an angle φ (see Fig. 15)

$$x = s_t . \sin \varphi$$

where s_t is the feed per tooth.

If the cutter rotates at n rev/min, the feed per minute is equal to:

$$s = n . n_t . s_t$$

and

$$x = \frac{s}{n . n_t} . \sin \varphi$$

The maximum chip thickness is then:

$$x_{\max} = \frac{s}{n . n_t} . \sin \varphi_s$$

(see Fig. 15).

With

$$\sin \varphi_s = \frac{2\sqrt{[a(d-a)]}}{d}$$

$$x_{\max} = \frac{s}{n . n_t} . \frac{2\sqrt{[a(d-a)]}}{d}$$

Even during milling operations with straight teeth, and therefore, constant width of cut, the cutting force and the power required are not proportional to the depth of cut, because the specific cutting resistance is not constant and varies with the changing chip thickness.[11]

It is possible to determine[12] the mean cutting torque and the mean power required at the milling cutter by means of a cutting resistance k_M which is related to the "middle chip thickness" h_M. This is the chip thickness in the middle of the arc described by the tooth during its cut. At this point the cutting resistance equals the mean value of the total cutting force taken over the full angle φ_s.[13]

The middle chip thickness is:

$$h_M = \frac{s}{n . n_t} . \sin \frac{\varphi_s}{2}$$

In the case of slab milling the ratio a/d is small, and for this reason it is possible to assume[11,12,14]

$$h_M = \frac{s}{n . n_t} \sqrt{\frac{a}{d}}$$

Figure 22a shows specific cutting resistance values for several materials as functions of the middle chip thickness and for normal rake angles.[14,15] The specific cutting resistance is independent of the helix angle δ.[11]

When, for given cutting conditions s, n, n_t, a, d the specific cutting resistance has been determined as a function of h_M the mean power required at the cutting edge can be related to the rate of metal removed per minute $a \times b \times s$ as follows:

$$N_M = \frac{k_M . a . b . s}{530\,000} \quad [\text{kW}]$$

The mean circumferential force acting at the cutter (tangential force P_{TM}) can then be calculated:

$$P_{TM} = \frac{N_M}{12v} . 530\,000 \quad [\text{lb}] \qquad (v \text{ in ft/min})$$

as well as the mean torque:

$$T_M = \frac{P_{TM} . d}{2} \quad [\text{lb in.}]$$

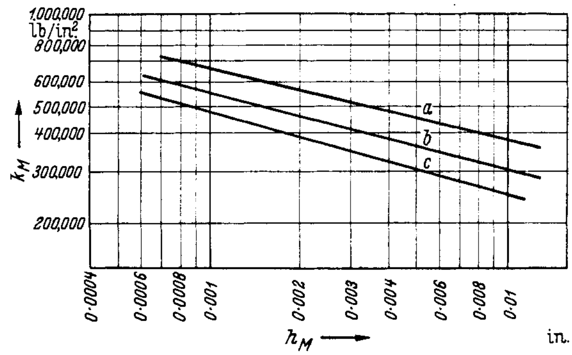


FIG. 22a. Specific cutting resistance during milling.
a Alloy steel, about 63 tons/in².
b Carbon steel, about 38 tons/in².
c Cast iron, about 200 Brinell.

The above equation for N_M gives the impression that for a specified width of cut the mean power required is proportional to the depth of cut and the feed rate, in other words, that it depends entirely upon the rate of metal removal and is independent of whether this is obtained with a large feed rate and a small depth of cut, or vice versa. This, however, is not the case because the specific cutting resistance decreases with growing middle chip thickness, whilst the middle chip thickness in turn is proportional to the feed rate and to the square root of the depth of cut. The power required is, therefore, smaller if a small depth of cut and a large feed rate are used.

A knowledge of the middle chip thickness will not enable the designer to determine the maximum cutting force, and this is essential when deciding upon the strength and stiffness of the machine. Under normal cutting conditions it may be assumed that the maximum circumferential force $P_{T\max}$ acting on the cutter is from 1·2 to 1·8 times the mean tangential force P_{TM}. The ratio $P_{T\max}/P_{TM}$ can be plotted as a function of the dimensions of the cutter, the width and depth of cut and the other cutting conditions (speed, feed rate).

For a helical tooth cutter, the length of the cutting path of a tooth is $r(\varphi_b + \varphi_s)$, where φ_b = effective angle between the leading and trailing edges, φ_s = tooth engagement angle and r = radius of the cutter.

The maximum number of teeth which can cut simultaneously is:

$$m \approx \frac{r(\varphi_b + \varphi_s)}{r\varphi_t}$$

or

$$m \approx \frac{\varphi_b + \varphi_s}{\varphi_t}$$

in which m is raised to its nearest whole number and where $\varphi_t = 2\pi/n_t$ the angle between two consecutive teeth of the cutter (n_t being the total number of teeth in the cutter).

In the case of a multi-tooth helical cutter, the leading edge of the first tooth enters the material and travels through an angle of rotation φ_t before the leading edge of the next tooth enters. After a further rotation by φ_t, this is followed by the leading edge of the third tooth, etc. After the leading edge of the first tooth has travelled through an angle φ_s, which depends upon the diameter of the cutter and the depth of cut, $\cos \varphi_s = 1 - 2a/d$, it leaves the material. This does not mean, however, that the whole tooth has ceased to be in action, as its trailing edge will still be cutting for a further angle φ_b, so that the total angle of engagement of one tooth is $\varphi_s + \varphi_b$. As soon as the cutter, after an initial period of "running into" a fresh block of material, reaches the depth of cut a, the cycle is repeated at intervals of φ_t.

A nomogram (Fig. 22b) can be drawn by means of which the ratio between the maximum and the average force can be found.[16]

Let a = depth of cut, b = width of cut, $d = 2r$ = cutting diameter, δ = helix angle.

Then

$$\varphi_b = \frac{2b . \tan \delta}{d}$$

$$\cos \varphi_s = 1 - \frac{2a}{d}$$

For given values of δ, d, and b, the value of φ_b can be determined. The stages to be followed in using the nomogram are shown by means of the arrows in Fig. 22b for the following example:

Cutter:	Diameter	$2\frac{1}{2}$ in.
	Number of teeth	8
	Helix angle	30°
Workpiece:	Width	2 in.
	Depth of cut	$\frac{5}{32}$ in.

On the right-hand side, the ratio P_{ac}/P_{dc} is plotted, where

$$P_{dc} = \text{average force}$$
$$P_{ac} = \text{maximum force–average force.}$$

From Fig. 22b, it can be seen that when φ_b is an integral multiple of φ_t, the ratio P_{max}/P_{TM} becomes unity and P_{ac}/P_{dc} is zero, i.e. there is no alternating component of the force. Also, for a given value of φ_t, the ratio P_{max}/P_{TM} is reduced by increasing a/d, i.e. φ_s.

The cutting force (P) acting on a helical cutter (helix angle δ) can be resolved into components either in the directions of the main dimensions of the cutter (tangential component P_T, radial component P_R and axial component P_A, Fig. 23a) or into components in the directions of the main movements of the milling machine table (longitudinal, vertical and transverse direction: horizontal component P_H, vertical component P_V, and axial component P_A, Fig. 23b). With the exception of the power calculation for the drive (see above), it is advantageous for the designer to know the force components in the direction of the main movements of the machine table. In this connexion it must be remembered that not only the force components acting on one edge but the sum of all of the forces acting on all edges which are simultaneously in action, must be considered. The component acting in the direction of the feed movement is approximately 90 per cent of the cutting force, and for this reason it is permissible to assume for calculation purposes $P_H = P$.

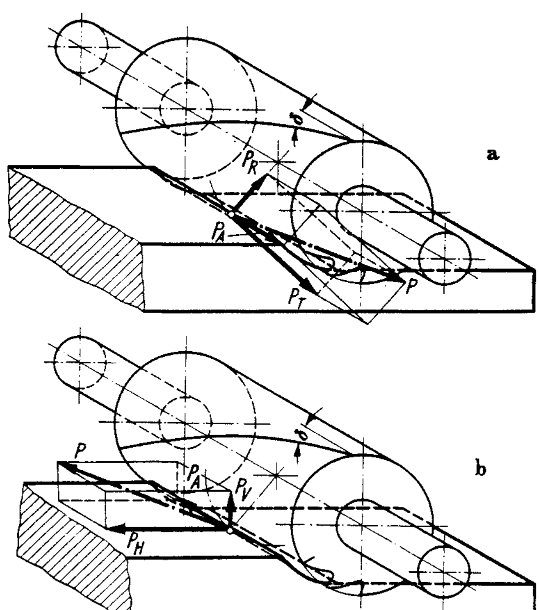


FIG. 23a and b. Resolution of the cutting force. a—acting on the cutting edge; b—acting on the workpiece.

The magnitudes of the force components depend on:

(a) The shape and size of the cutter; diameter, number of teeth, rake angles, helix angle,
(b) The properties of the workpiece material,
(c) The cutting conditions: depth of cut (a), width of cut (b), cutting speed v (spindle speed n), feed (s) (per minute, per tooth or per revolution of the cutter).

The selection of the cutter, and the choice of the cutting conditions depend on considerations

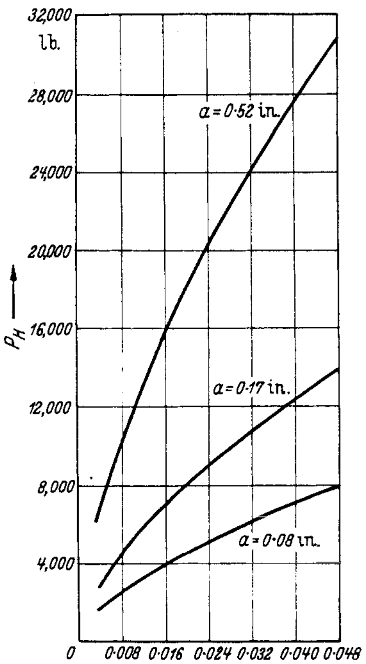


FIG. 24. Horizontal component P_H of the cutting force when milling carbon steel (45 tons/in²).

Cutter diameter $4\frac{3}{8}$ in.
Number of teeth 11.
Width of cut 5 in.
Various depths of cut (*a*)

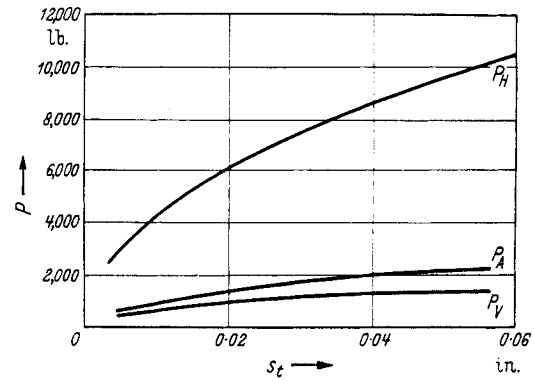


FIG. 25. Cutting force components when milling cast iron.

Cutter diameter $4\frac{3}{8}$ in. Depth of cut $\frac{5}{32}$ in.
Number of teeth 11 Width of cut 5 in.
Helix angle 25°

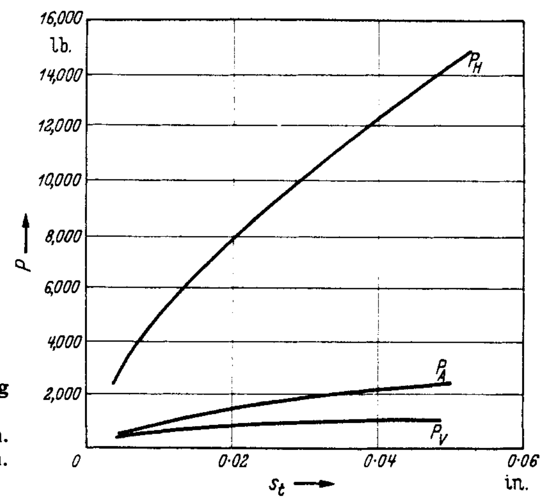


FIG. 26. Cutting force components when milling carbon steel (45 tons/in²).

Cutter diameter $4\frac{3}{8}$ in. Depth of cut $\frac{5}{32}$ in.
Number of teeth 11 Width of cut 5 in.
Helix angle 25°

(accuracy and finish of the machined surface, productivity, cost, etc.) which the designer cannot always predict. However, he must consider the most unfavourable loading conditions to which the machine may be subjected and base his calculations and designs on these.

He must also remember that the maximum values of the three components need not necessarily occur simultaneously and that not only the geometry of the cutting process, but also a possible eccentricity of the cutter teeth may cause periodic pulsations of the cutting forces.

The effect of some of these parameters as well as the cutting force components which have been observed under normal cutting conditions during the machining of several materials are shown in Figs. 24 to 26. Recommended cutting speeds are shown in Table 7.

TABLE 7

CUTTING SPEEDS (FEET/MIN.) FOR MILLING CUTTERS, ASSUMING RIGID WORKPIECES AND MILLING CUTTERS

Material	High Speed Steel Cutters	Carbide Cutters
Carbon Steel up to 38 tons/in².	50 to 100	160 to 400
Cast Iron	32 to 50	128 to 200
Light Alloys	640 to 1280	1280 to 2000

As the specific cutting resistance decreases with growing chip thickness, and as the forces do not increase in proportion to the feed rate (see Figs. 24 to 26), it is advantageous to use high feed rates per tooth. These can be obtained by applying high feed rates and low cutting speeds or by using milling cutters with small numbers of teeth. The teeth of such milling cutters can be made stronger than the teeth of those with larger numbers of teeth, and there is, therefore, little danger arising from the load per tooth increasing with smaller numbers of teeth. Low cutting speeds also result in longer tool life which is of advantage from an economic point of view because of the cost of re-grinding milling cutters. A further advantage is the fact that the frequency of force pulsations decreases in proportion both to the spindle speed and the number of teeth in the cutter.

Although under these conditions the ratio between cutting force and rate of metal removal is favourable, the absolute values of the force will, of course, increase and the machine, the milling arbor and the milling cutter must be capable of withstanding the resulting higher loads and stresses. Whilst the ratio between maximum and minimum cutting force becomes smaller with increasing helix angle (see Fig. 22b), the maximum cutting force (especially the axial component) and power requirements increase.

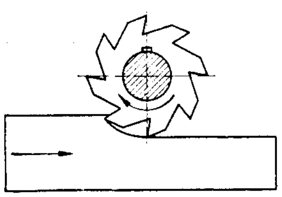


FIG. 27a. Up-cut milling

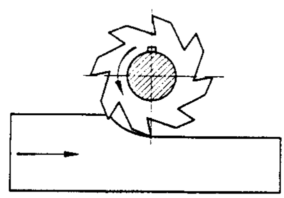


FIG. 27b. Down-cut (climb) milling

With increasing depth of cut the total force also increases but the force pulsation is reduced because of several edges cutting simultaneously. It must be realized, however, that with too large a depth of cut the vertical component (P_V) can become negative and create chatter.

In the process so far described the cutting movement of the milling cutter is opposed to the feed movement (Fig. 27a), and this is known as up-cut milling. Advantages can often be gained by down-cut milling (Fig. 27b), in which the cutting movement and the feed movement act in the same direction.

During up-cut milling the chip thickness is infinitely small at the beginning of the cutting action and this results in the cutting edge often pressing or rubbing rather than cutting. During down-cut milling, the cutting action starts with a relatively large chip thickness and the cutting conditions are, therefore, more favourable. In addition the milling cutter holds the workpiece down against its locating faces and does not work against clamping elements, a fact which is particularly important when machining thin and flexible workpieces which are difficult to clamp.

There exists, however, a danger in the case of down-cut milling, because the feed component of the cutting force acts in the direction of the feed movement of the workpiece and may draw the machine table towards the cutter if backlash exists in the table drive or in the slideways. This may result in an instantaneous increase of the feed rate, and therefore, the cutting force, thus overloading the cutter tooth. Under such conditions, the milling cutter may climb upon the workpiece (climb milling), bend the milling arbor and damage the cutter and the machine. Down-cut milling is, therefore, only possible with the more rigid machine tools which are also equipped with devices for the elimination of any backlash in the drive and in the slideways. Figure 27c shows the specific cutting resistances during up and down milling of steel.[11]

(c) Milling with Cutter Heads

For the machining of plain surfaces, face milling, especially with cutter heads, is gaining more and more ground. In contrast to the slab milled surface which consists of cycloidal patterns the face milled surface is plane as it is made up from the plane areas which are described by the cutting edges of the tool. The cutter head with interchangeable tools is also advantageous from the point of view of tool maintenance. It is, of course, important to use great care in the setting of the inserted teeth especially if they are ground separately and not in the cutter head itself.

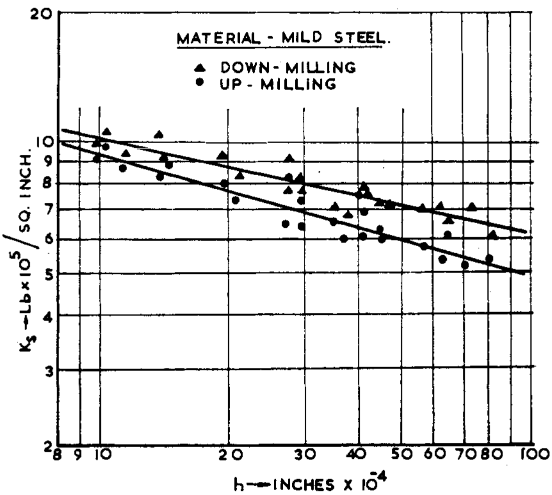


FIG. 27c.

Face milling is basically similar to slab milling,[17] although the ratios between the three components of the cutting force, vertical, horizontal (longitudinally and transversely) and the loading of the machine are not the same because of the different relative position between the axis of the cutter and the machined surface. The width of the face milled surface corresponds to the depth of cut a during slab milling. The ratio a/d and the angle of engagement of each tooth are, therefore, considerably larger during face milling than they are during slab milling, and the simplified equation for the middle chip thickness, which is valid for small values of a/d, is not applicable. It is also important to consider the distance of the cutter axis from the centre line of the milled surface because this has an effect on the load conditions.[16]

If the cutter diameter is not much larger than the width of the machined surface the conditions may become similar to those of down-cut milling and the designer of the machine has to take these conditions into consideration.

If the cutter diameter is considerably larger than the width of cut the chip thickness at the beginning of the cut is not infinitely small as it is in the case of slab milling, and the performance is considerably better. The conditions prevailing at the beginning of the cut, i.e. when the cutting edges enter the workpiece material not only influence the tool life[18] but also the loading conditions of the machine tool.

The periodicity of the cutting force waveform depends upon the angle φ_s (see page 11) which is a function of the diameter of the cutter and the width of the workpiece. The angle φ_s depends also upon the relative position between the centre line of the cutter and the centre line of the workpiece. From Fig. 28a:[16]

$$\cos \varphi_1 = \frac{B + 2e}{2R}$$

and

$$\cos \varphi_2 = -\frac{(B - 2e)}{2R}$$

where B = width of cut and $\varphi_s = \varphi_2 - \varphi_1$.

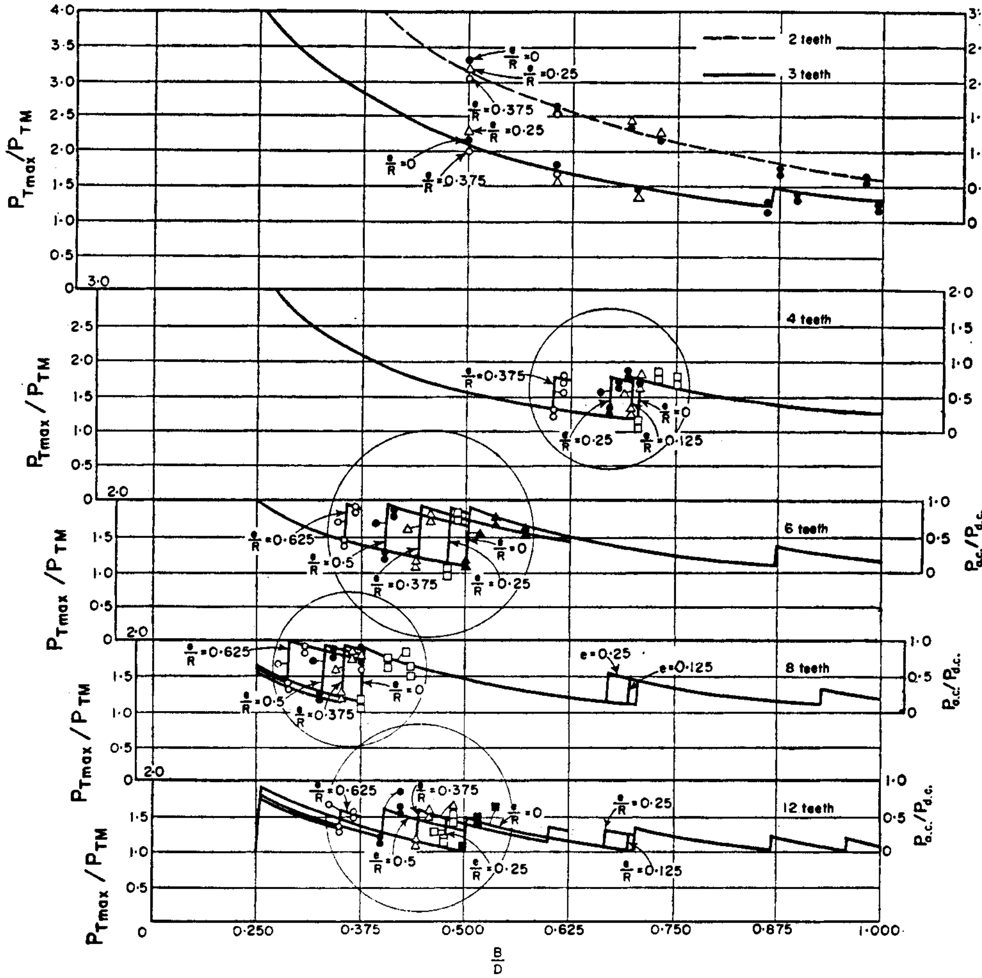

2 teeth
3 teeth
4 teeth
6 teeth
8 teeth
12 teeth
P_{Tmax}/P_{TM}
$P_{a.c.}/P_{d.c.}$
$\frac{B}{D}$

FIG. 28b.

φ_s changes with differing positions of the cutter with respect to the workpiece. The average chip thickness is used to determine the power and the average force (Fig. 28a).

$$h_M = \frac{s_t(\cos\varphi_1 - \cos\varphi_2)\sin\theta}{\varphi_2 - \varphi_1}$$

$$= \frac{s}{n.n_t}.\frac{\sin\theta}{\varphi_s}.\frac{2B}{D}$$

where n = rev/min of the spindle, n_t = number of teeth of the cutter.

The values of k_M as a function of h_M in face milling are identical with those found for the case of slab milling for the same cutter and workpiece materials.

When k_M is known:

$$N_M = \frac{B.a.s.k_M}{530\,000} \quad [\text{kW}]$$

and

$$P_{TM} = \frac{N_M.530\,000}{12v} \quad [\text{lb}] \qquad (v \text{ in ft/min})$$

The ratio $P_{T\max}/P_{TM}$ decreases with decreasing values of φ_t and becomes a minimum if φ_s is an integral multiple of φ_t. Hence, in order to reduce the alternating component of the force, the ratio B/D should be chosen in such a manner that for a given ratio e/R, the angle φ_s is just below a multiple of φ_t (Fig. 28b).[16]

4. GRINDING

The grinding operation is similar to the milling operation in that the grinding wheel may be considered as a rotating tool with a large number of cutting edges distributed over the periphery. These carry out the cutting movement, whilst the depth setting and the feed movement are allocated either to the tool or the workpiece, according to the design of the grinding machine. However, the actual operation is basically different from those of turning or milling due to the order of magnitude of forces, chip sections, etc., the cutting conditions and the rake angles.[19]* The cutting edges are formed by the edges of the abrasive grains which are held together by the bonding material. Abrasives used are: natural and synthetic corundum (aluminium oxide), emery, silicon, carbide and boron carbide. They are held together by mineral, vegetable or ceramic bonds. The grinding wheel is specified by the grain size of the abrasive (i.e. size of sieve) and the hardness of the bond (the resistance to breaking out of the abrasive grains). The structure of the grinding wheel depends upon the ratio between grain size and bond thickness.

Although the grinding wheel resharpens itself, when the force acting on a blunted grain increases until it exceeds the strength of the bond and the grain breaks out, it is necessary to dress grinding wheels from time to time with a diamond in order to maintain true running conditions. In order to prevent undesirable vibrations, grinding wheels must also be statically and dynamically balanced.

Generally speaking, hard workpiece materials are best ground with soft grinding wheels and vice versa. It is, however, necessary to consider the dimensions of the grinding wheel and the grinding conditions when selecting a grinding wheel[20] for a specific task.

(a) Grinding Forces[21]

The grinding forces (cutting force components) are relatively small, and therefore of secondary importance from the point of view of strength and stiffness of the machine. However, the deformations of the workpiece especially during cylindrical grinding are of major importance. In contrast

* See also the numerous publications on work carried out at the Machine Tool Laboratory of the T. H. Braunschweig, Germany (Prof. G. Pahlitzsch), and by J. Peklenik, of the Machine Tool Laboratory of the T. H. Aachen.

to the conditions during turning, the radial component P_3 is from 1·6 2·4 times to the magnitude of the tangential component P_1 when grinding with an emulsion coolant, and both components increase as the grinding operation proceeds.[22] If an oil is used as coolant the tangential component is found to be smaller and it increases only slightly as grinding proceeds. The ratio P_3/P_1 is, therefore, larger and grows considerably as grinding proceeds.

The power requirements of the machine depend on the tangential force component P_1 acting on the grinding wheel. During cylindrical grinding the mean instantaneous chip section q_m increases with growing width of cut, longitudinal feed rate, depth of cut and circumferential velocity of the workpiece, and with decreasing circumferential velocity of the grinding wheel:

$$q_m = \frac{a \, . \, s \, . \, v_w}{(v_s + v_w)}$$

where a = the depth of cut, in.
s = feed, in/rev
v_w = circumferential velocity of the workpiece, ft/min
v_s = circumferential velocity of the grinding wheel, ft/min.

Figure 29 shows some tangential force components P_1 measured during cylindrical grinding of a 40 ton/in² steel.[23] Figure 30 shows the tangential force components P_1 encountered during surface grinding (peripheral cut).[24] The grinding force during plunge grinding can be estimated by assuming that the width of the grinding wheel equals the longitudinal feed during cylindrical grinding, and that the force increases in proportion to the width of the grinding wheel.

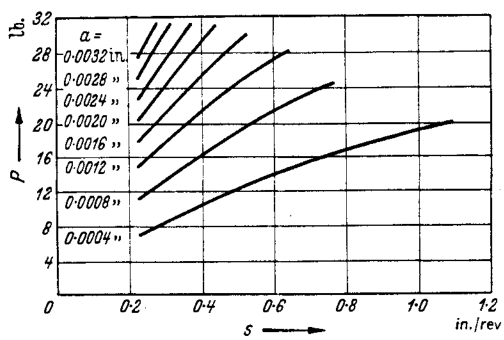


FIG. 29. Tangential force components when cylindrical grinding carbon steel (40 tons/in²) (a = depth of cut) (from Kurrein, *loc. cit.*).

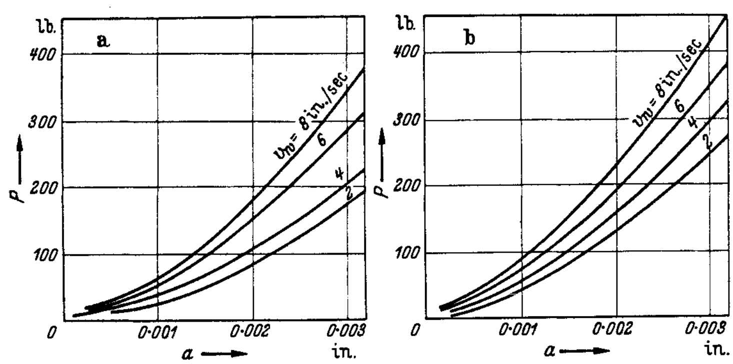


FIG. 30a and b. Tangential force components during surface grinding (peripheral cut) (from Krug, *loc. cit.*).
a—Carbon steel (26 tons/in²), wet ground with wheel Ek white 30 I Ke.
b—Cast iron, wet ground with wheel SiC dark 30 I Ke.
Width of workpiece: 2 × 2 in. = 4 in. Workpiece velocity: v_W
Circumferential velocity of grinding wheel: v_s = 100 ft/min Depth of cut: a

(b) Cutting Speed, Feed Rate and Depth Setting

The circumferential speed of the grinding wheel is limited by the strength of the bond. The permissible speeds in the case of mineral bonds may be up to 50 ft/sec, and in the case of ceramic bonds up to 100 ft/sec. Higher speeds (up to 250 ft/sec in the case of cutting off wheels and special machines) are permissible for bakelite bonded grinding wheels. With increasing circumferential speeds, soft grinding wheels behave as if they were harder.

The choice of the optimum circumferential speed depends upon the operating conditions. If the circumferential speed is too low the wheel wear increases.

As the length of the arc of contact between grinding wheel and workpiece increases, the circumferential speed of the grinding wheel must be reduced. The following values are recommended by Whibley.[25]

External cylindrical grinding	Large diameter grinding wheel, small diameter workpiece	100 ft/sec
	Grinding wheel diameter smaller than workpiece diameter (roll grinding)	75 ft/sec
Surface grinding		60 ft/sec
Internal grinding		50 ft/sec

During cylindrical grinding longitudinal feed rate and circumferential speed of the workpiece must be balanced. Generally a longitudinal feed per revolution of the workpiece equal to two-thirds of the grinding wheel width is chosen, because this is found to result in the wheel wear being concentrated on the centre path of the grinding wheel periphery. If the feed rate is too small the greatest wheel wear occurs at the edges. If the circumferential speed of the workpiece is too high a relatively large instantaneous chip thickness results, and the wheel wear is again high. On the other hand, if the circumferential speed of the workpiece is too low, the resulting local heating and deformation of the workpiece affect the accuracy.

For the external cylindrical grinding of steel, Whibley[25] recommends a circumferential speed of about 60 ft/min for the workpiece. Brödner[20] mentions lower circumferential speeds (20 to 30 ft/min) for finish grinding of steel, circumferential speeds of up to 60 ft/min for the external cylindrical grinding of brass and up to 220 ft/min for the external cylindrical grinding of aluminium. For internal grinding operations on steel Whibley[25] recommends up to 120 ft/min, and for surface grinding, he recommends workpiece traverse speeds of 30 to 40 ft/min (roughing) and about 20 ft/min (finishing). The depth of cut depends upon the working conditions and can vary between 0·0002 and 0·002 in.

5. PLANING AND SHAPING

The cutting process, the working conditions and the tools are very similar to those used in turning. However, although the cutting forces follow the same laws they are not always the decisive factor affecting the design of the machines.

The feature by which the planing and shaping process differs mostly from others is the reciprocating movement comprising both the cutting stroke and return stroke. The interruption of the cutting action after each forward stroke and the intermittent feed movement, which occurs during the return stroke, result in impact loads on the tool which are transmitted to the elements carrying the tool and the workpiece as well as to the driving mechanism (hydraulic, pinion or worm and rack, linkage, etc.).

Moreover, it is important to obtain as closely as possible uniform velocity of the moving parts during the cutting stroke and this depends upon the driving mechanism selected. Very frequently the choice of the cutting speed is determined not by the tool life, but by the dynamic problems of the driving elements which also affect the speed of the return stroke.

6. BROACHING[26]

During the working stroke the broaching tool which is either pulled through the workpiece or (in the case of external broaching) along the workpiece, carries out a straight line cutting action, after which it returns to its initial position. In this respect broaching is similar to planing and shaping. However, the feed is not obtained by moving the workpiece or the tool, but by the arrangement in series of suitably stepped cutting edges (see Fig. 1). The total cutting resistance exerted by all chip sections which lie in front of the cutting edges has to be overcome during the working stroke. This total varies with depth of cut, stepping and pitch of the broach and with the length of the surface to be machined. The total cutting force does not, therefore, remain constant during one working stroke. In the example (Fig. 31), L is the length of the machined surface, l the length of the tool entry and l' the length during which the total chip section remains constant, i.e. the length over which the largest number of cutting edges, corresponding to the machined length L

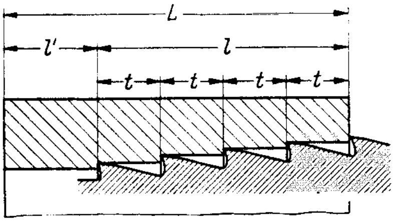
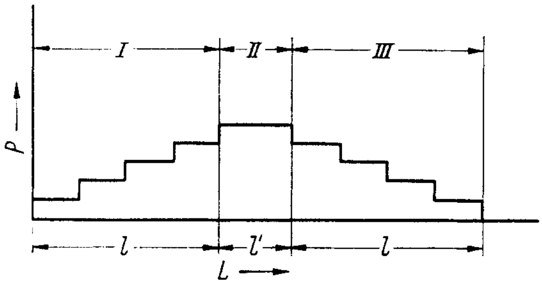


FIG. 31



I = Entry of the Tool
II = Full Cut
III = Exit of the Tool

FIG. 32

and the pitch t, are engaged. As the cutting edges are usually arithmetically stepped, the chip sections and, therefore, the cutting resistance exerted on all cutting edges are approximately equal. The resulting cutting force acting on the broach varies with the broaching length in accordance with Fig. 32.

For a correctly designed and well-lubricated broach, the maximum broaching force, which determines the design parameters of the machine, can be expressed as:

$$P = k \,.\, k_s \,.\, a \,.\, b \,.\, \frac{l}{t}$$

in which

k is a coefficient which takes friction into consideration (1·1 to 1·3)
k_s is the specific cutting resistance (ton/in^2)
a is the depth of cut per tooth (in.)
b the width of cut (in.)
l the length of tool entry (in.)
t the pitch of the cutting edges (in.)
l/t the largest number of cutting edges which are simultaneously in action.

The value of l/t has to be rounded off to the next higher whole number.

Brödner[20] gives the following values for specific cutting resistances and cutting speeds.

Specific cutting resistances (depth of cut 0·01 in.):

Steel	130 to 180	tons/in^2
Cast iron	120	tons/in^2
Aluminium–silicon	65	tons/in^2
Electron	30	tons/in^2

If the depth of cut is only 0·004 in., these values have to be increased by about 30 per cent.

Cutting speeds for broaching steel:

Internal broaching: 12 to 20 ft/min.
External broaching: 20 to 30 ft/min.

The smaller values are applicable for long broaching lengths. For a more accurate determination of the broaching forces, especially for external broaching operations, Nalchan[27] gives the following formulae:

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:

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

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.

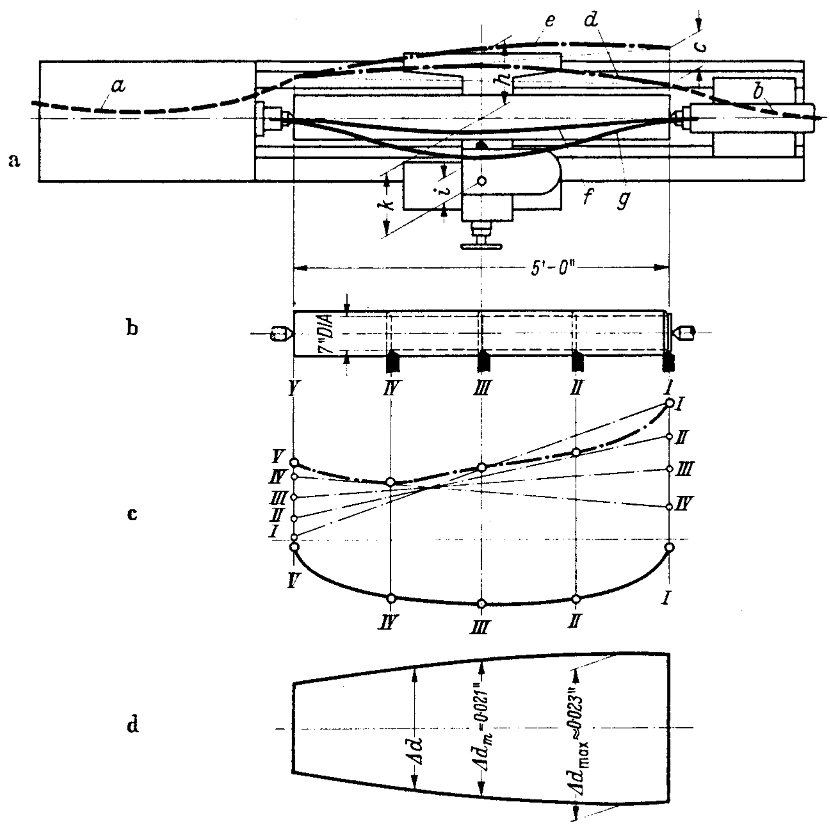


FIG. 33a–d

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

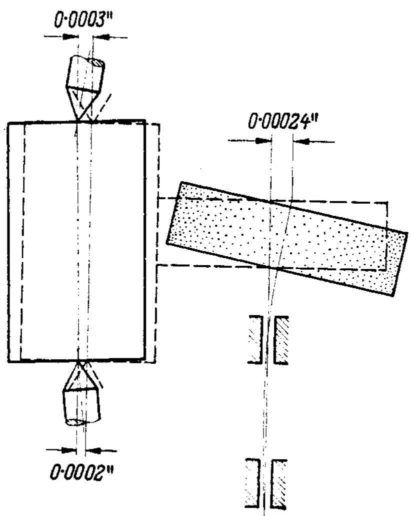


FIG. 34

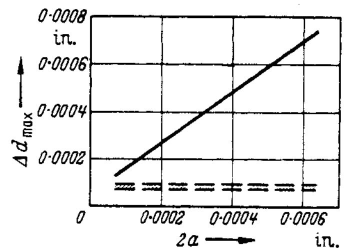


FIG. 35

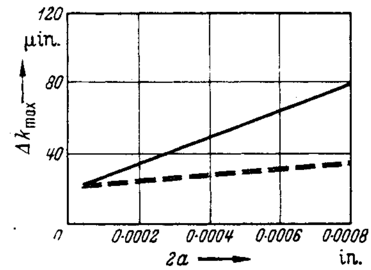


FIG. 36

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.

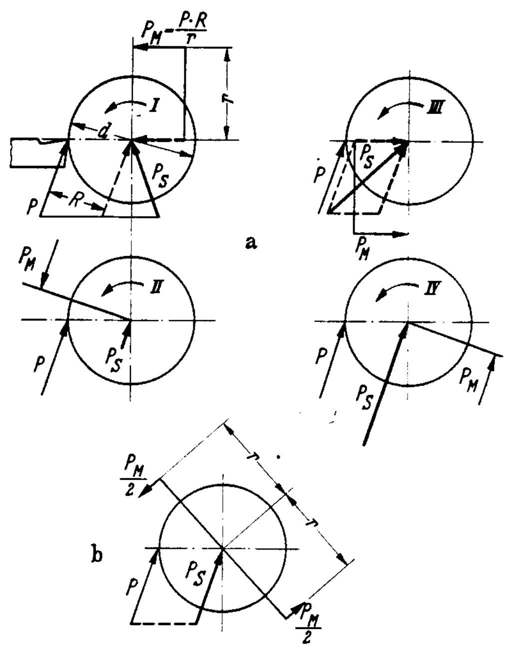


FIG. 37a and b.

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.

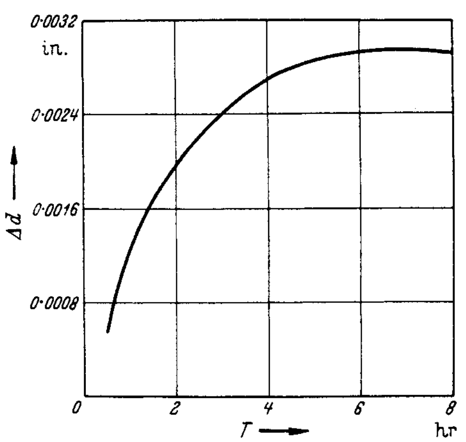


FIG. 38

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]

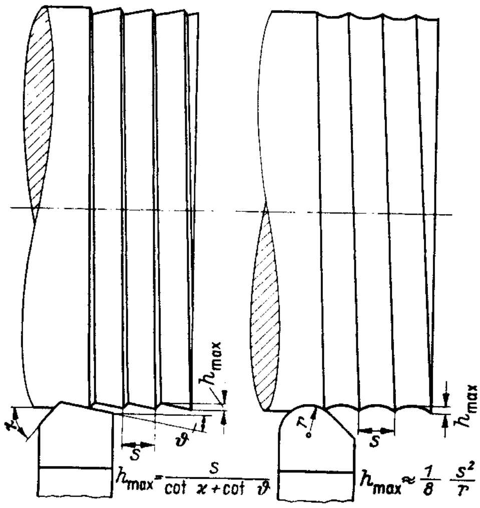


FIG. 39

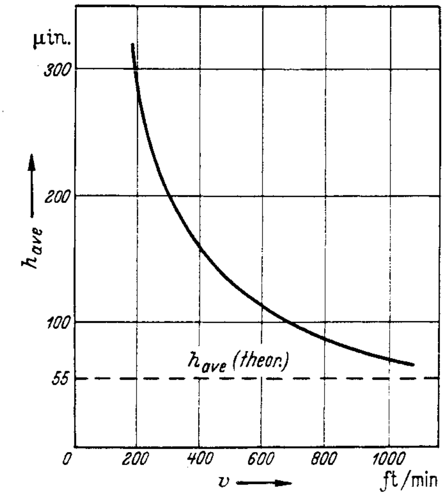


FIG. 40. Effect of cutting speed upon surface finish (from Kumar). Tool—tungsten carbide, negative rake (−3°); Material—mild steel (28 tons/in^2); Depth of cut 0.002 in.; Feed—0.00875 in./rev.

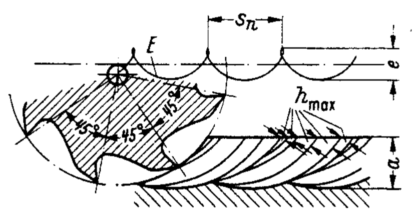


FIG. 41. Surface machined with eccentric milling cutter: a = depth of cut; E = path of cutter axis relative to workpiece; e = eccentricity of cutter; h_{max} = maximum chip thickness per tooth; s_n = feed per revolution of cutter. Number of cutter teeth: 8.

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]

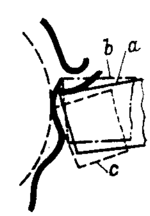


FIG. 42. Influence of tool vibrations on the turned surface (from Kronenberg): ——a) normal position of turning tool; —·—·— b) highest position of turning tool during vibration; - - - - c) lowest position of turning tool during vibration; —) turned surface.

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).

From Figs. 5(a) and 8, the specific cutting resistance k_s and the recommended cutting speed v, both corresponding to the required feed rates, can be found, and if the depth of cut a is known, the cutting force P_1 can be calculated.

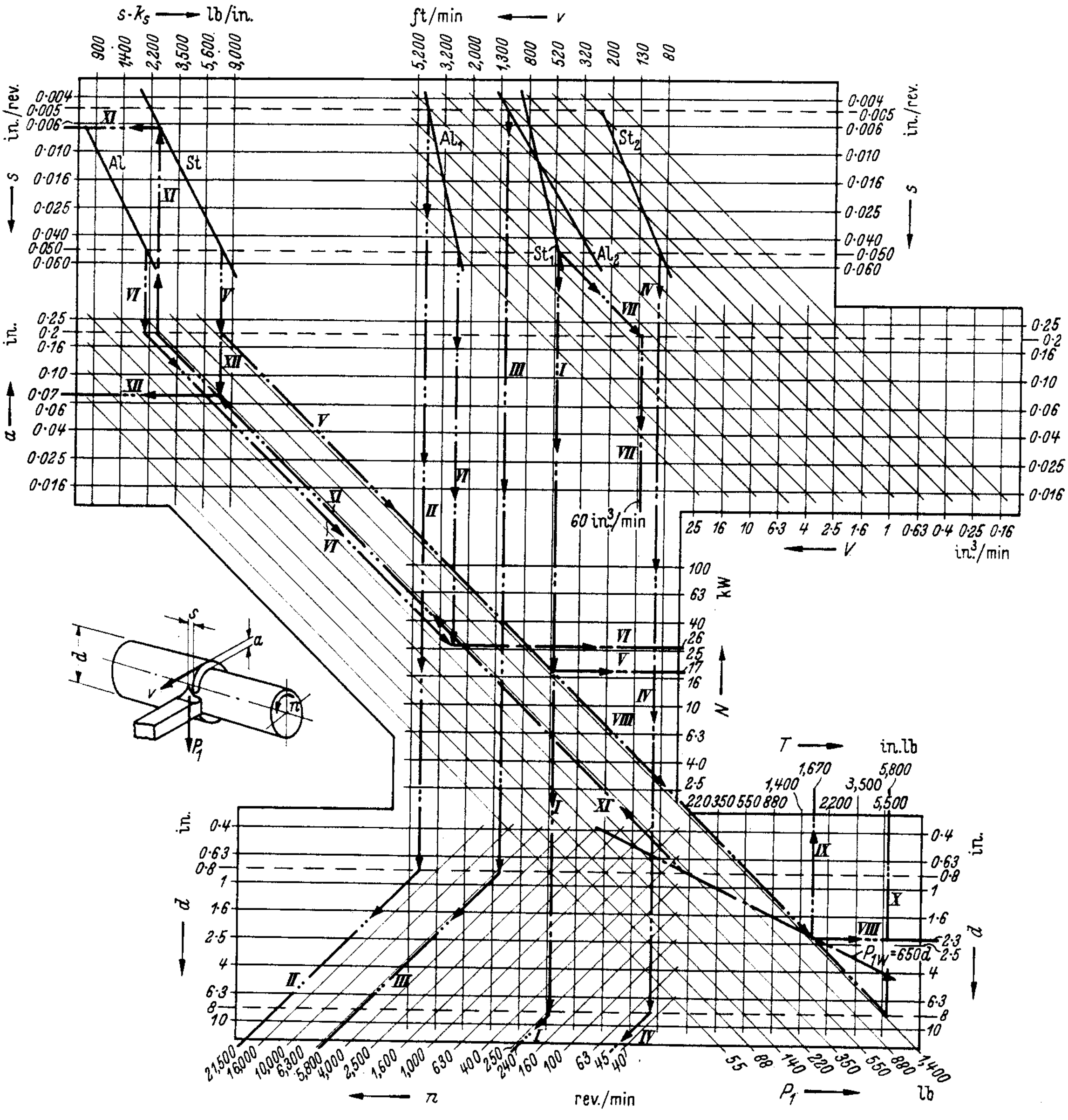


FIG. 43. Nomogram for the centre lathe.

In the nomogram, Fig. 43, $s.k_s$ is plotted as a function of s in the top left-hand corner and below this $s.k_s$ together with a shows the straight lines sloping at 45° from left to right, which indicate the cutting force component $P_1 = a \times s \times k_s$. At the top right-hand side, v is plotted as a function of s, and lower down, the intersections between the vertical lines for v and the sloping lines for P_1 determine the net power:

$$N = \frac{P_1 . v}{530000} \quad [\text{kW}] \qquad (P_1 \text{ in lb}; v \text{ in in./min})$$

The lines $s \times v$ which slope at an angle of 45° from left to right, start at the intersections between the vertical lines (v) and the horizontal lines (s), and their intersections with the horizontal lines (a) define the rate of metal removal $V = a \times s \times v$.

In the lower part of the nomogram, the spindle speeds $n = v/\pi d$ can be found from the intersections of the vertical lines (v) and the horizontal lines (d). The intersections of the sloping lines (P_1) and the horizontal lines (d) at the right-hand side of the nomogram determine the torque acting on the workpiece or the spindle:

$$T = P_1 \times \frac{d}{2}$$

The accuracy of the workpiece obtained during finishing operations is affected mainly by deflexion in the horizontal plane (see page 24).

Although the deflexion of the workpiece in the vertical plane under the force component P_1 is of less influence upon the accuracy, it is important in the case of roughing operations because of its effect upon the relative height of the cutting edge in relation to the workpiece axis.

This deflexion is

$$\delta_1 = \frac{P_1 \times l^3}{48EI}$$

where

$$I = \frac{\pi d^4}{64}$$

In the case of steel ($E = 30 \times 10^6$ lb/in^2) P_1 then becomes

$$P_1 = \frac{48 \times 30 \times 10^6 \times \pi \times d^4 \times \delta_1}{64 \times l^3} = 70{\cdot}5 \times 10^6 \times \delta_1 \times \left(\frac{d}{l}\right)^3 \times d$$

If it may be assumed that d/l will not be less than $\frac{1}{6}$, and that the maximum permissible deflexion of the workpiece in the vertical plane is 0·002 in, the maximum permissible cutting force defined by the maximum permissible deflexion of the workpiece becomes $P_{1w} = 650\ d$. This relationship between P_1 and d is plotted as a chain dotted line in the right-hand bottom corner of the nomogram.

The use of the nomogram may be demonstrated for the example of a centre lathe which satisfies the following specification:

(a) *Workpiece*

1. *Materials*—Steel up to 32 ton/in^2 (St) and aluminium (Al).
2. *Dimensions*—Maximum diameter to be turned: $d_{max} = 8$ in.
   Minimum diameter to be turned: $d_{min} = 0{\cdot}8$ in.
   Maximum length between centres in the case of maximum diameter: $l = 48$ in.
3. *Deformation*—Maximum permissible deflexion in the vertical plane $\delta_1 = 0{\cdot}002$ in.

(b) *Tool*

1. *Materials*—Tungsten carbide (St_1 for machining steel, Al_1 for machining aluminium). High-speed steel (St_2 and Al_2 respectively).
2. *Rake angles*—Standard, (approach angle $\kappa = 45°$).*

(c) *Cutting conditions*

These are taken from the nomogram.

* For other values of κ a correction factor can be applied (see p. 6).

The cutting resistance and cutting speeds applicable for the above-mentioned workpiece materials and tools (high-speed steel tools 60 min life, tungsten carbide tools 240 min life) are plotted (straight lines) at the top left-hand and top right-hand sides respectively of the nomogram. Feed rates, $s_{max} = 0{\cdot}05$ in/rev, $s_{min} = 0{\cdot}005$ in/rev and maximum depth of cut $a_{max} = 0{\cdot}2$ in. have been entered as dotted lines.

If 32 ton/in^2 steel (St) and aluminium (Al) have to be machined at the maximum permissible feed rate with tungsten carbide tools (chain dotted lines *I* and *II*), the recommended cutting speeds (St_1 and Al_1) are obtainable for all diameters only if a spindle speed range from 240 ÷ 21,500 rev/min is available. If the above materials are to be machined with high-speed steel tools (St_2 and Al_2), a speed range from 45 ÷ 5,800 rev/min is required (Lines *III* and *IV*).

A speed range of $\frac{21500}{45} = 480$ is difficult to achieve economically. It is possible, however, to design a gear box on the assumption that the maximum diameter of a steel workpiece would be machined with tungsten carbide tools and the smallest possible diameter of an aluminium workpiece would be turned with high speed steel tools. In this case, the speed range would become $\frac{5800}{240} = 24$.

If steel is to be machined with tungsten carbide tools employing the maximum permissible feed rate and depth of cut, the net power at the cutting edge would be 17 kW (Lines *V*). If aluminium is to be machined (Lines *VI*), the cutting power would be 26 kW. For the case of steel, the rate of metal removal would be about 60 in^3/min (Lines *VII*). If steel is machined at the maximum permissible feed rate and depth of cut (Lines *V*), the cutting force component P_1 is so large that under the conditions stated before the minimum diameter of the workpiece must not be less than 2·3 in. (Lines *VIII*). This corresponds to a torque of 1670 in. lb at the spindle (Line *IX*). If the maximum diameter of 8 in. is to be machined under the same conditions, the torque at the spindle would be 5800 in. lb (Line *X*). On the other hand, the limit for the deflexion of a workpiece of minimum diameter (0·8 in.) determines the maximum cutting force component P_1 and limits it to a feed rate of 0·006 in/rev (if the maximum depth of cut is to be taken) (Lines *XI*). However, it is unlikely that a workpiece of the minimum diameter of 0·8 in. would be machined with a depth of cut of 0·2 in. and it is, therefore, more probable that the depth of cut would be limited to 0·07 in. so that the maximum feed rate could be used (Lines *XII*).

2. MILLING MACHINE

The performance of the milling machine is less influenced by the dimensions of the workpiece than by the dimensions of the tool, and especially the tool carrier (the milling arbor). As a result of standardization the variety of available milling arbor diameters is small, so that in the interests of simplicity such diameter need not be considered as a variable in the nomogram. The parameters are, therefore:

(a) *Workpiece*
 1. Materials

(b) *Tool*
 1. Material
 2. Diameter
 3. Number of teeth
 4. Helix and rake angles
 5. Life

(c) *Cutting conditions*
 1. Cutting speed (v)
 2. Feed rate (s)
 3. Depth of cut (a)
 4. Width of cut (b).

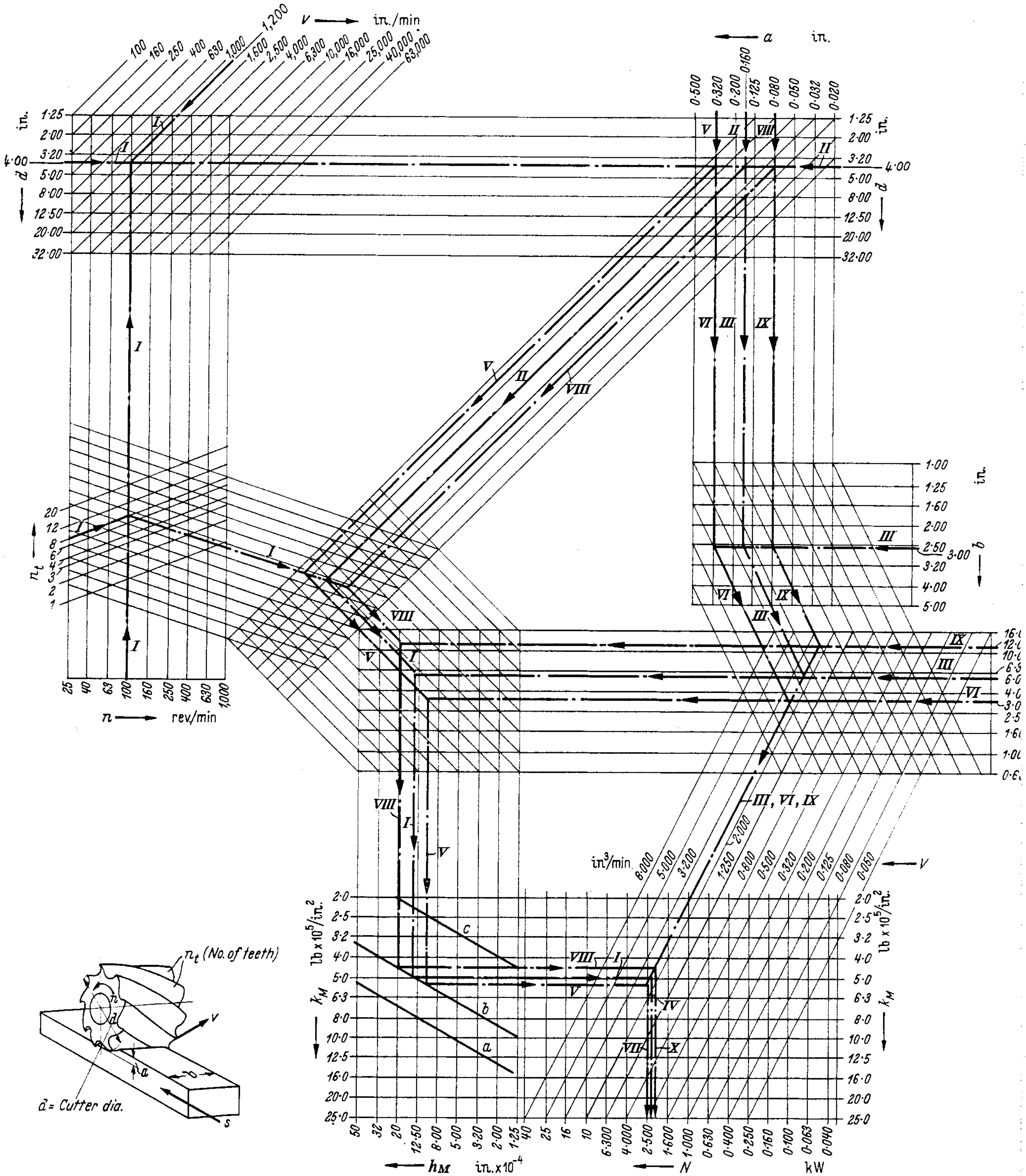


FIG. 44. Nomogram for the milling machine.
k_M values for: a—Alloy steel (100 tons/in²).
b—Carbon steel (38 tons/in²).
c—Light Alloy.

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 = 0·16 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 = 2·9 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:

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

and

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

Hence

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

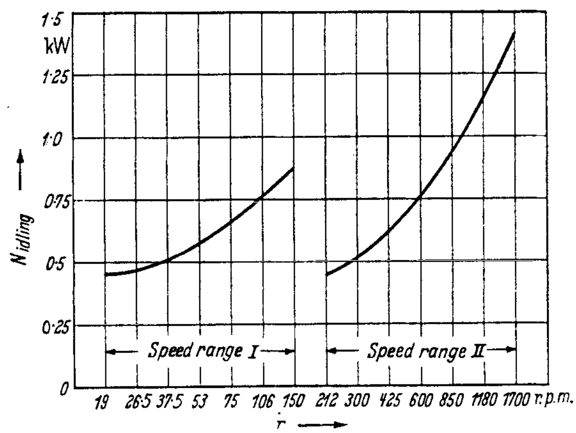


FIG. 45

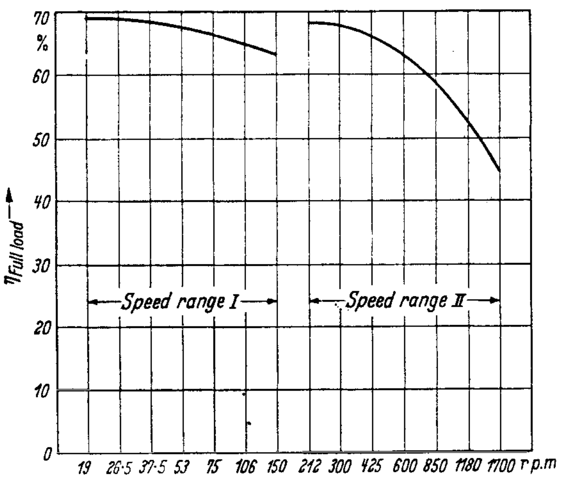


FIG. 46

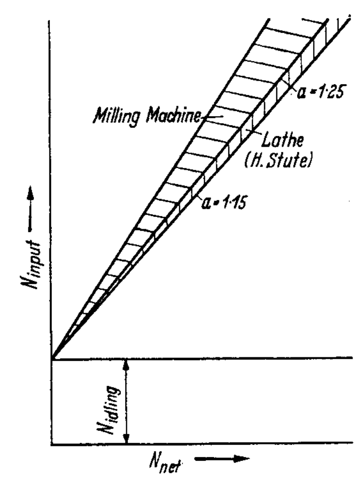


FIG. 47

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):

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

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

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

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

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

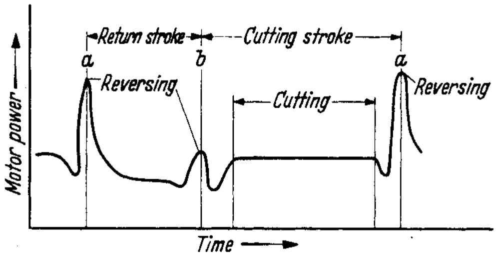


FIG. 48

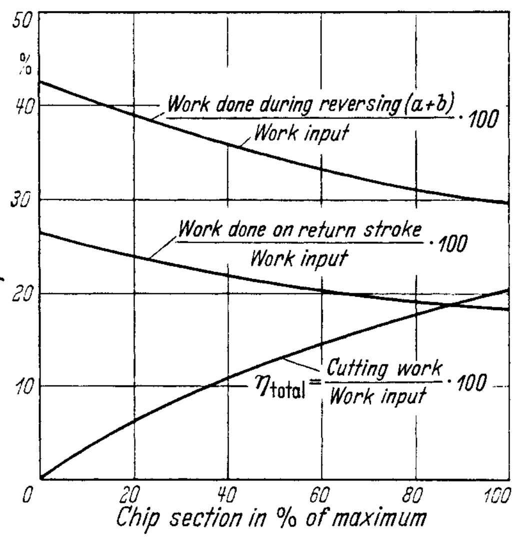


FIG. 49

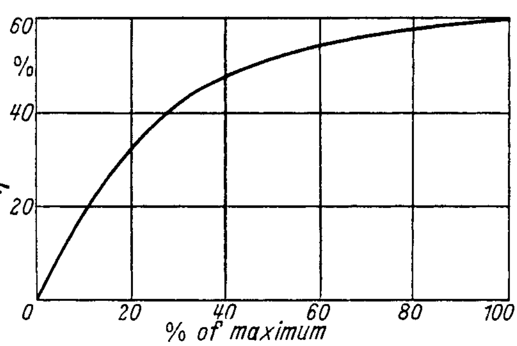


FIG. 50

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]

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 (Fig. 51). In general, the force required

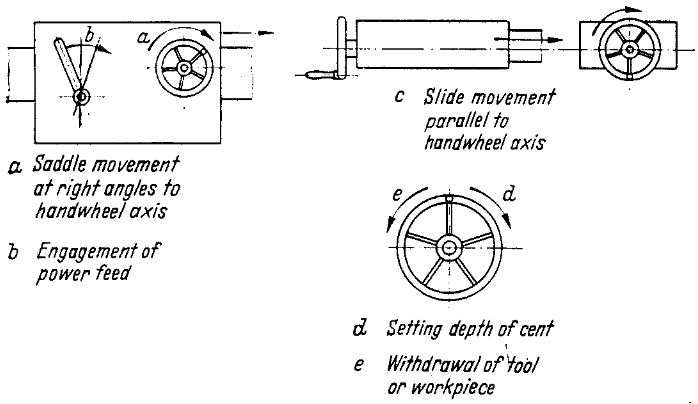


FIG. 51

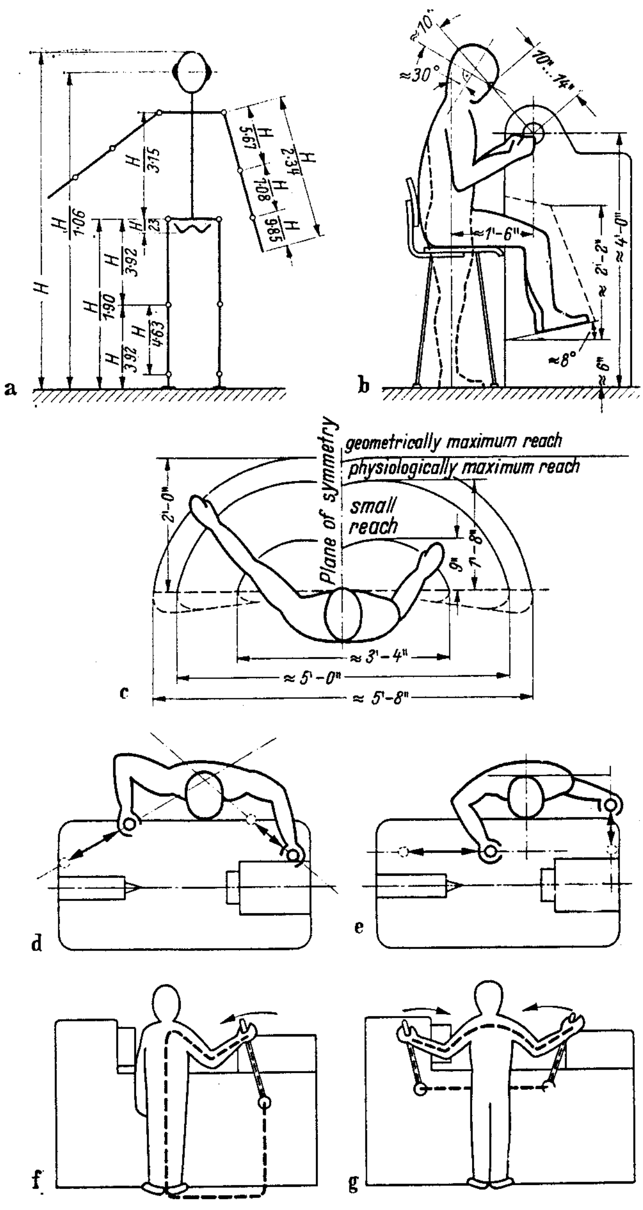


FIG. 52. a—proportions of the human body (after B. Schulte: Arbeitserleichterung, München, Hanser, 1952); b—normal body position during operation of machine; c—reach in the horizontal plane; d—arrangement of operating levers so that the operational movement is directed towards the longitudinal axis of the body; e—arrangement of operating levers so that the operational movements are directed parallel to the axis; f—closing the flow of force through the body in a large loop; g—closing the flow of force through the body in a small loop.

to operate levers and wheels should not exceed 30–40 lb. If foot pedals have to be operated frequently, the maximum force required should be about 20–30 lb; if their operation is less frequent and if the operator can apply his weight directly, an operating force of 50–60 lb may be permitted. If a lever has to be moved from a sitting position, the maximum force required should not exceed 8–16 lb. If machines are to be operated by women, these values may have to be reduced, sometimes considerably. Unnecessary movements and fatigue can be avoided if the layout of levers and handwheels is adapted to the geometry of the human body (Fig. 52).[41]

(b) The possibilities of repair and maintenance work[42] must be borne in mind during the design stage of a new machine. Unnecessary work, loss of time and of productivity in the workshop can be greatly reduced, for instance, if parts which are subjected to heavy wear and are likely to cause other difficulties, are designed and arranged in such a manner that repair work can be

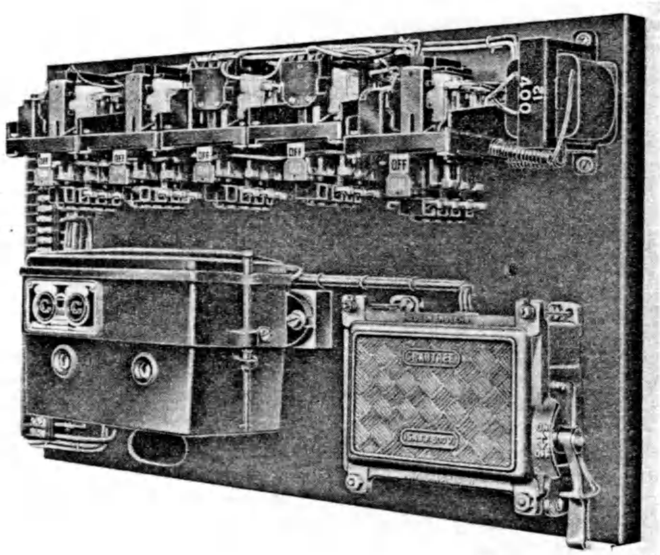


FIG. 53. Arrangement of electrical control gear for a production milling machine (see Figs. 133, 134) as a self-contained unit. (Cooke & Ferguson Ltd., Manchester).

carried out easily and rapidly. Quite apart from the great importance of standardizing sizes and types of screws and nuts in such a manner that the required quantity of spanners can be reduced to a minimum, the use of self-contained units can be of great value. Typical applications of such units will be found in electrical and hydraulic control gear, etc. An example is the electric control panel for a production milling machine (Fig. 53). This panel carries all the control gear required for starting, stopping and reversing the spindle, the feed and the quick traverse drive, together with all interlocking circuits (see page 104). If some electrical fault occurs, the machine need not be taken out of the production programme whilst the fault is being investigated. If in a given workshop several similar machines are installed it is only necessary to carry in stock one spare control panel which can be quickly exchanged for the faulty one. The latter can then be taken to the electricians' workshop for the fault to be located and repaired, the machine remaining operative through the use of the replacement panel. After repair, the faulty panel serves as a replacement panel for use in any future case of fault.

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Part II

DESIGN PRINCIPLES

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]

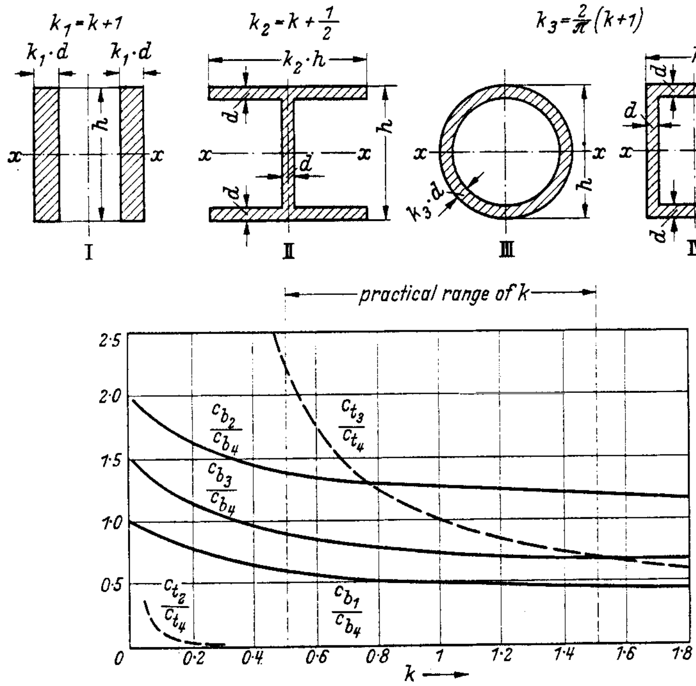


FIG. 54. Bending and torsional stiffness of different cross-sections. c_{b1}—Bending stiffness of section I; c_{b2}—Bending stiffness of section II; c_{t2}—Torsional stiffness of section II; c_{b3}—Bending stiffness of section III; c_{t3}—Torsional stiffness of section III; c_{b4}—Bending stiffness of section IV; c_{t4}—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. (It should be pointed out that in the case of this example shear stresses and shear deflexions have not been considered.) The calculation shows that the ratio between the depth of the beam and the square of the free length is important, and this ratio will have to be entirely different if a cast iron beam is employed rather than a welded steel beam, and if the latter is to be neither too stiff ($\delta_{\text{permissible}}$) nor too strong ($\sigma_{b\text{permissible}}$). For a given permissible maximum stress and a permissible maximum deflexion there is only one optimum ratio l^2/h for each material, and this is determined by the intersection of the V_δ and V_{σ_b} curves.

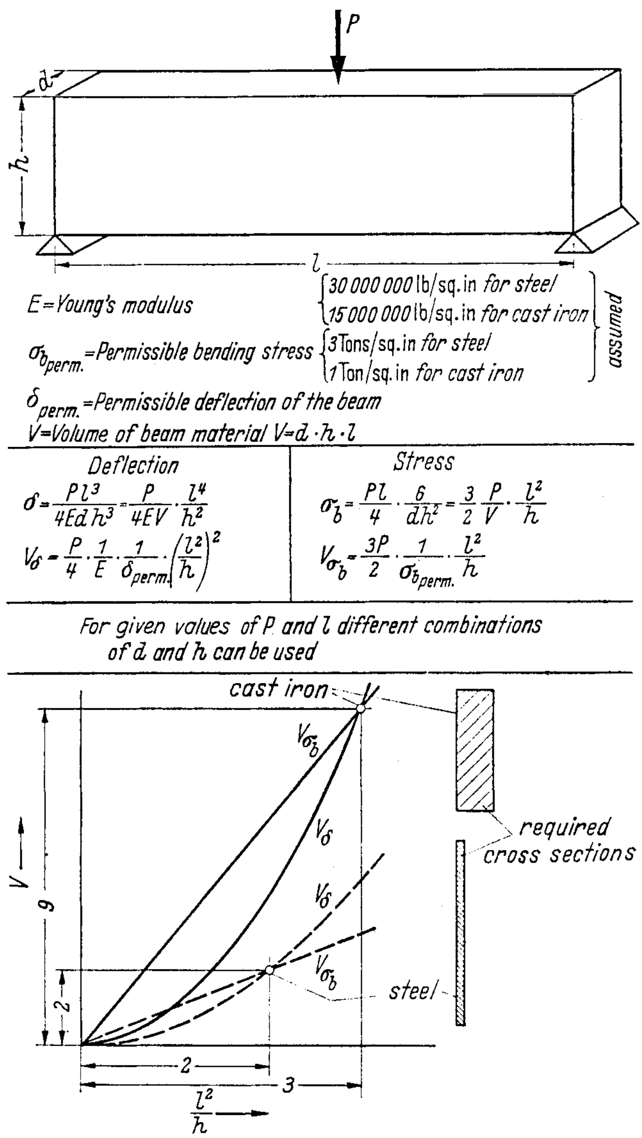


FIG. 55

If the designer is not prevented by other considerations from applying this optimum ratio l^2/h then the saving in material obtained by employing mild steel instead of cast iron will reach its maximum value, which—in the example shown in Fig. 55—is more than 70 per cent.

In order to satisfy these conditions, the steel beam has to be thin and deep compared with its free length, and this creates some difficulty. The height of the machine bed is usually limited by other design considerations, and small wall thicknesses require stiffeners whose presence increase not only the material consumption, but also the cost of labour. The ideal combination of free length, depth and wall thickness is, therefore, often difficult, if not impossible, to obtain. However, the designer should always bear in mind the principle that the wall thicknesses of steel structures should be smaller and the sections deeper than in corresponding cast iron structures.

In practice, the ratio l^2/h will in the majority of cases be found to lie to the right of the intersection between the V_δ and V_σ curves, so that stiffness and not load carrying capacity is the decisive factor for determining the dimensions and the amount of material required for a machine tool structure. Where this is the case the theoretical material consumption for a steel structure will be approximately half that required for a cast iron structure, because E for steel is approximately twice that of cast iron. The use of high tensile alloy steels does not present any advantage here because the value of E for most steels cannot vary by more than about ± 3 per cent.

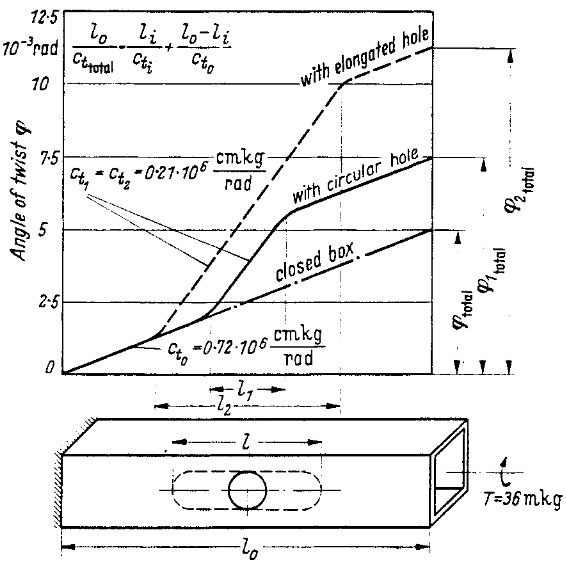


FIG. 56. Angle of twist and torsional stiffness of box beams with apertures in one wall (from Bielefeld).

The load carrying parts of a machine tool cannot always be designed and built with cross-sections which are constant over their whole length. The weakening effect of apertures, especially upon the torsional stiffness is well known. Interesting model experiments in the Machine Tool Laboratory of the Technische Hochschule, Aachen, have provided the results which are reproduced in Fig. 56.[3] These show that the effect of a circular hole affects a length of about twice the diameter of the hole ($l_1 \approx 2d$, see Fig. 56). An elongated aperture (length l) affects the total stiffness even more because the disturbed range is still greater compared with the total length of the structure ($l_2 = l + d$, see Fig. 56). Such apertures are, however, often unavoidable in machine structures, beds, etc., where for example, gearboxes, etc., have to be assembled and fitted into the structure. These apertures are then closed by suitable cover plates which re-establish the appearance of the closed box section. Although a further series of experiments (Fig. 57) has shown that the reduction of the bending stiffness is relatively small and almost completely compensated by the application of a suitably designed cover plate, this is not the case as far as torsional stiffness is concerned. The loss of about 72 per cent of torsional stiffness due to the aperture is only partly made up, and even with the most favourably designed cover plate the torsional stiffness is still only 41 per cent of its original value.

The weakening effect of apertures can be reduced by suitable arrangements of stiffeners. The diagonal arrangement of stiffeners (see Fig. 317c) originally suggested by Peters[4] is superior to an arrangement of straight transverse stiffeners and produces higher stiffness not only against bending, but also against torsion.*

* For the calculation of the increase in torsional stiffness see ref. 5.

The effect of apertures and stiffeners upon the deformations of wall panels in a box section has been investigated in Russia, where experimental measurements were compared with the results of

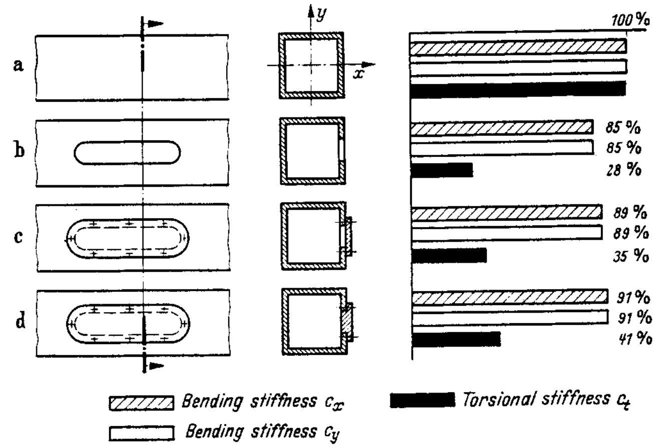


FIG. 57a–d. Static stiffness of a box beam, apertures and different cover plates (from Bielefeld).

suggested methods of calculation.[6] The magnitude of deformations of a wall panel with constant wall thickness is given by:

$$\delta_0 = K_0 . \frac{P . a^2(1 - \mu^2)}{Eh^3}$$

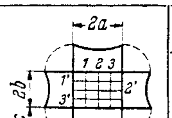
TABLE 10

THE NUMERICAL VALUES OF THE COEFFICIENT K_0

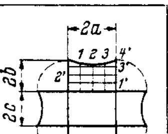
The deformation δ_0 of a panel with the dimensions $2a$, $2b$ and the wall thickness h of an open box is:

$$\delta_0 = \frac{P . a^2(1 - \mu^2)}{E . h^2} . K_0$$

(a) The loaded panel $2a \times 2b$ is connected on 4 sides with the box.

Ratio $a : b$ of the loaded panel		1 : 1									1 : 0·75					
Ratio $a : b : c$ of the box dimensions		1 : 1 : 1			1 : 1 : 0·75			1 : 1 : 0·5			1 : 0·75 : 0·75			1 : 0·75 : 0·5		
	Coordinates of the point of load application	1	2	3	1	2	3	1	2	3	1	2	3	1	2	3
	1′	0·18	0·24	0·18	0·20	0·28	0·20	0·21	0·31	0·21	0·13	0·18	0·13	0·13	0·20	0·13
	2′	0·24	0·35	0·24	0·28	0·44	0·28	0·31	0·50	0·31	0·21	0·30	0·21	0·22	0·33	0·22
	3′	0·18	0·24	0·18	0·20	0·28	0·20	0·21	0·31	0·21	0·13	0·18	0·13	0·13	0·20	0·13

(b) The loaded panel $2a \times 2b$ is connected on 3 sides with the box.

Ratio $a : b$ of the loaded panel		1 : 1			1 : 0·75						1 : 0·5					
Ratio $a : b : c$ of the box dimensions		1 : 1 : 1			1 : 0·75 : 1			1 : 0·75 : 0·75			1 : 0·5 : 1			1 : 0·5 : 0·75		
	Coordinates of the point of load application	1	2	3	1	2	3	1	2	3	1	2	3	1	2	3
	1′	0·16	0·25	0·16	0·15	0·20	0·15	0·15	—	0·15	0·08	0·09	0·08	0·08	—	0·08
	2′	0·30	0·48	0·30	0·29	0·45	0·29	0·28	0·42	0·28	0·19	0·28	0·19	0·18	0·27	0·18
	3′	0·43	0·70	0·43	0·39	0·62	0·39	—	0·62	—	0·34	0·51	0·34	—	0·48	—
	4′	0·95	1·40	0·95	0·77	1·16	0·77	—	0·16	—	0·62	0·92	0·62	—	0·69	—

— Hinged support. ∩ Unsupported side of the panel.

where P = load
a = half the maximum length of the wall under consideration
E = Young's Modulus
μ = Poisson's ratio
h = wall thickness of the panel
K_0 = a coefficient, the value of which depends upon the coordinates of the point of load application and upon the dimensions of the panel edges. Values of K_0 are given in Table 10.

If a wall panel is interrupted by apertures and stiffened by bosses and stiffeners, the deformation can be calculated approximately by means of the equation:

$$\delta = \delta_0 . K_1 . K_2 . K_3$$

where δ_0 = deformation of an uninterrupted panel
K_1 = a coefficient which depends upon the effect of the bore and the boss through which the load is transmitted
K_2 = a coefficient which depends upon the effect of unloaded bores and bosses
K_3 = a coefficient which is determined by the type of stiffener.

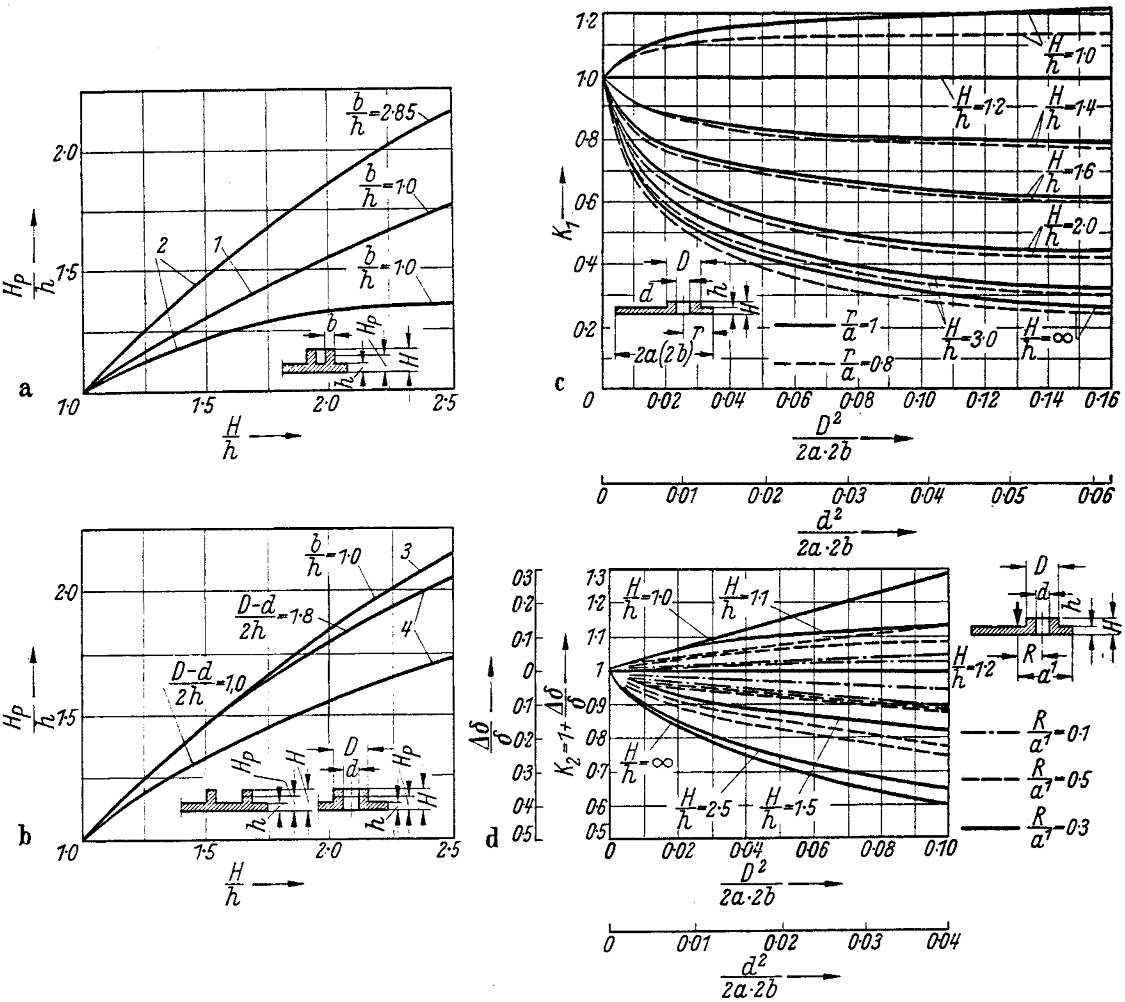


FIG. 58. Determination of coefficients K_1 and K_2 (from Reschetow and Kaminskaja). Effective height of stiffeners and bosses: a—For beams with: 1) diagonal stiffeners, 2) transverse stiffeners; b—For plates with: 3) stiffeners, 4) bosses; c—Coefficient K_1 as a function of main dimensions. Force acting on boss; d—Coefficient K_2 as a function of the dimension of unloaded bores. δ = Deformation of loaded panel without holes. $\Delta\delta$ = Increase of deformation due to holes and bosses.

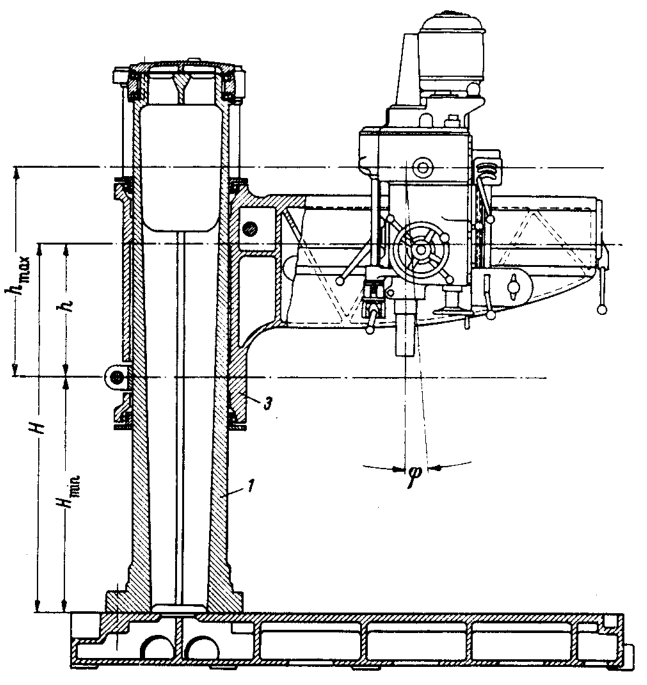


FIG. 59

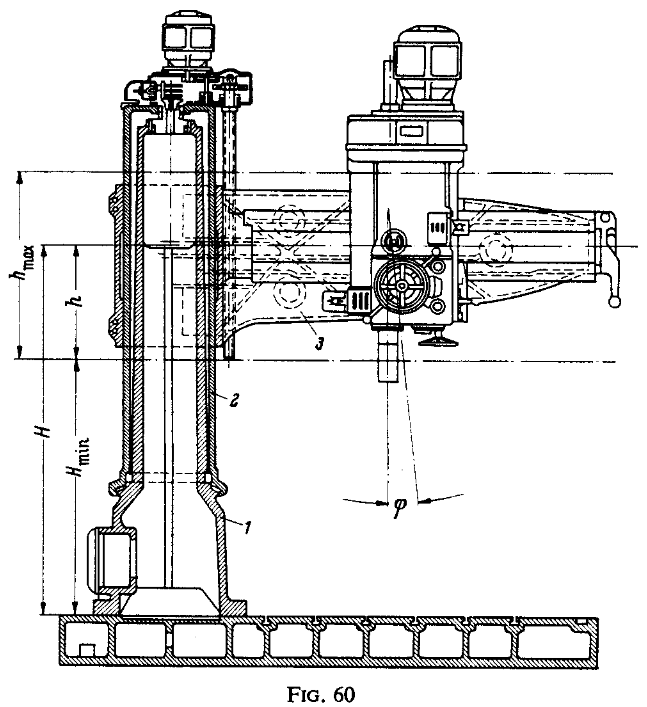


FIG. 60

The values for K_1 and K_2 can be determined from curves given in Fig. 58, whilst K_3 varies between 0·75 and 0·9 according to the design adopted for the stiffeners.

The effect of the general design layout, i.e. the combination of different design elements, upon the total stiffness of a machine structure can be shown by the example of a radial drilling machine structure. This consists basically of two superimposed cantilevers, viz:

1. The column which is fixed to the base by means of bolts.
2. The radial arm which is guided on the column.

In present designs the radial arm (*3*) is guided either directly on a column (*1*) on which it can be swivelled and axially displaced (Fig. 59), or on a sleeve (*2*) inserted between radial arm (*3*) and column (*1*) (Fig. 60). Both arrangements have certain advantages and disadvantages which can be of a technical and of an economic nature. A simple calculation[7] for a given axial thrust on the spindle shows that in the arrangement of Fig. 59, the maximum inclination φ of the spindle axis occurs when the radial arm is in its top position, and the minimum inclination when it is in its lowest position. In the case of the arrangement shown in Fig. 60, however, the minimum inclination of the spindle axis occurs when the radial arm is half way up its vertical traverse (Fig. 61). This point must be considered not only during the design of a machine but also when its performance is being judged, especially during acceptance tests.

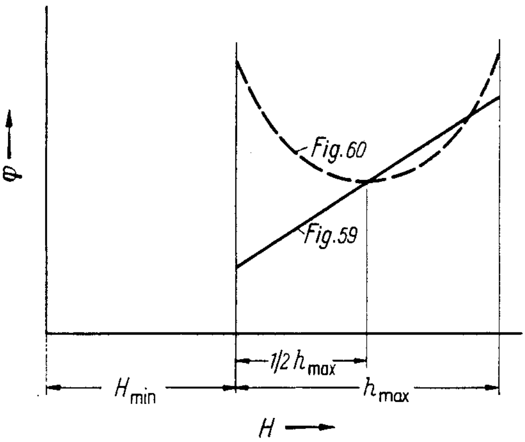


FIG. 61

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.

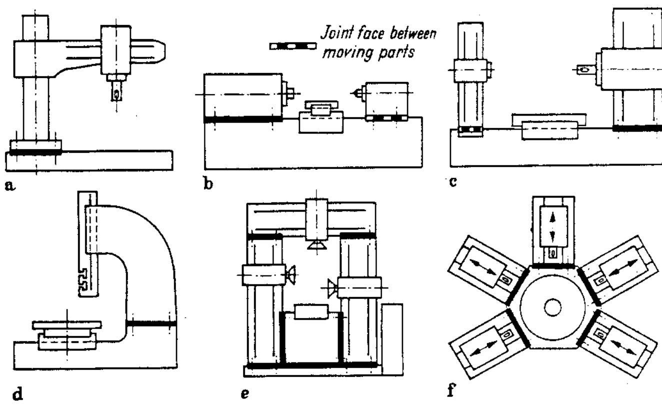


FIG. 62. Examples of bolted joints in machine tool structures. a—Radial drilling machine; b—Centre lathe; c—Horizontal boring machine; d—Vertical slotting machine; e—Plano milling machine; f—Multi-spindle drilling units.

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

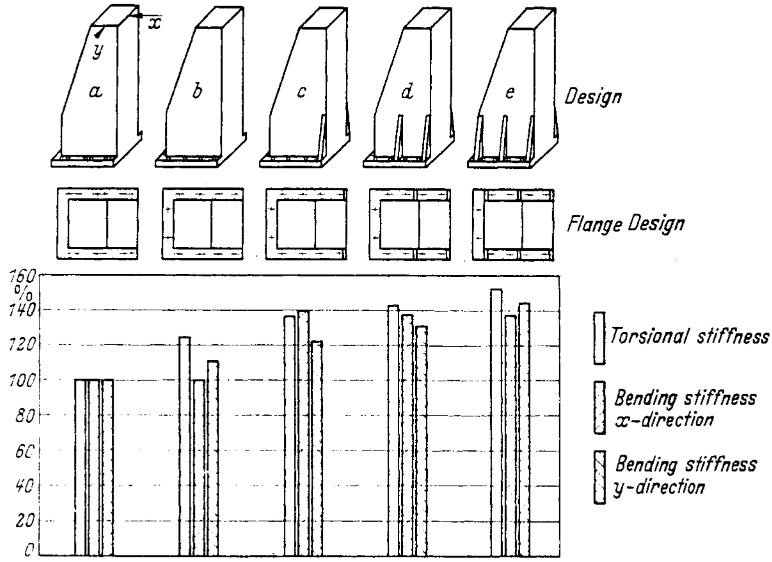


FIG. 63. Effect of flange design on stiffness (from Bielefeld).

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

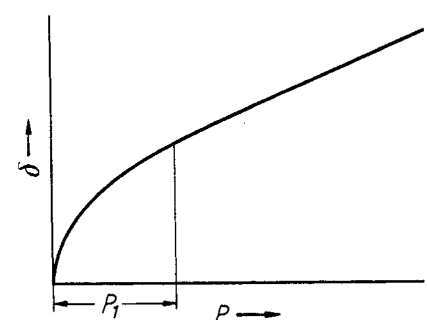


FIG. 64. Deflexion of lathe spindle. P—Transverse force acting on the spindle nose; δ—deflexion measured at the spindle nose; P_1—recommended preload.

* 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

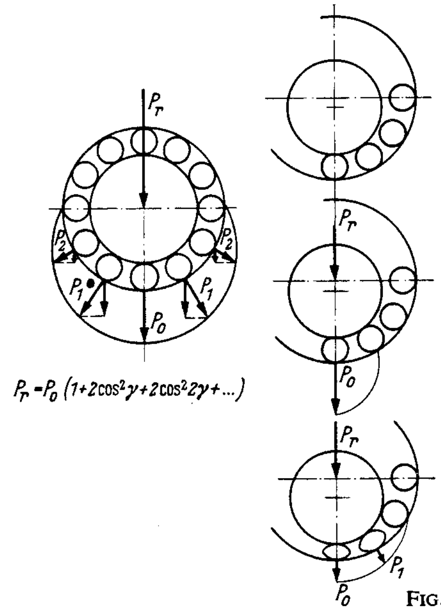


FIG. 65. Load distribution in a ball bearing (from Honrath).

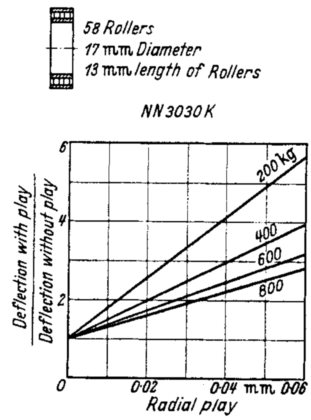


FIG. 66. Stiffness as a function of play in a roller bearing (from Honrath).

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].

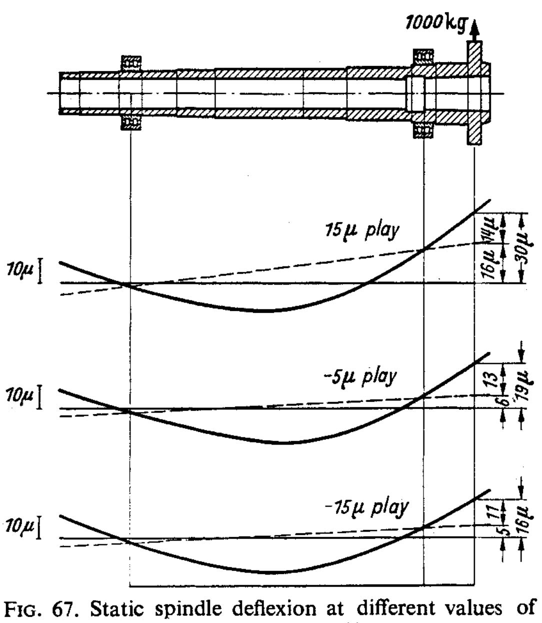


FIG. 67. Static spindle deflexion at different values of play (from Honrath).

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).

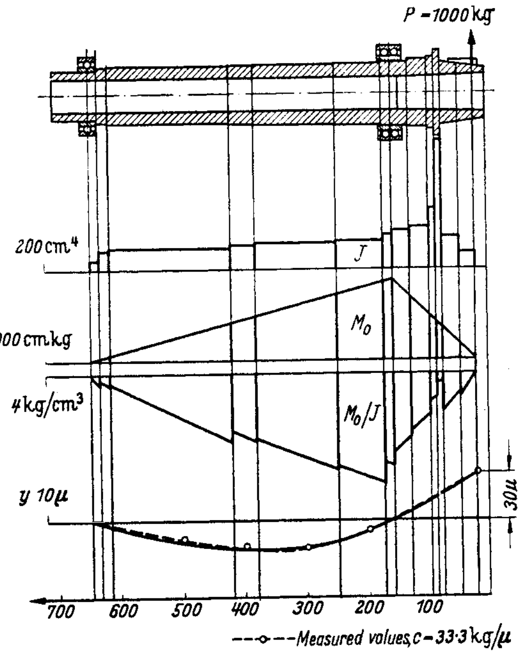


FIG. 68. Comparison of measured and graphically determined spindle deflexion (Mohr's graphical method) (from Honrath).

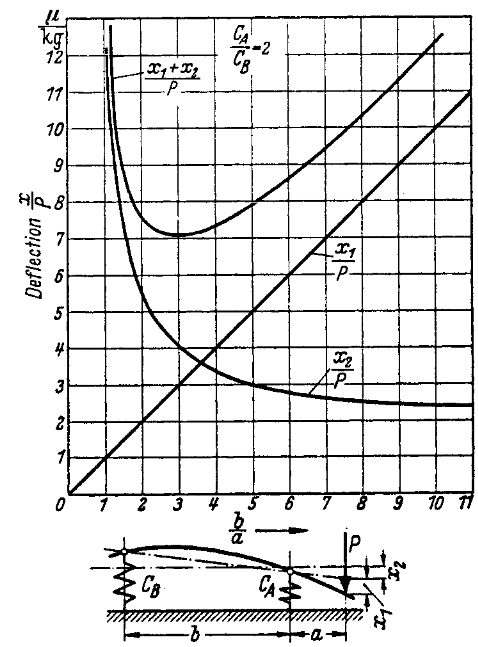


FIG. 69. Determination of optimum beaming spacing (from Honrath).

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

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

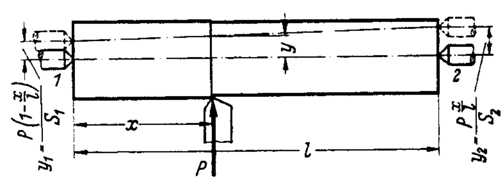


FIG. 70

The stiffness of the machine parts carrying the centres may be S_1 and S_2 respectively, the ratio S_1/S_2 being α. The displacements of the centres are, therefore:

$$y_1 = \frac{P.[1-(x/l)]}{S_1}$$

$$y_2 = \frac{P.x/l}{S_2}$$

The displacement of the workpiece axis relative to the position of the cutting edge (Fig. 70) is then

$$\begin{aligned} y &= y_1 + (y_2 - y_1).\frac{x}{l} \\ &= P\left[\frac{1-(x/l)}{S_1} + \left(\frac{x/l}{S_2} - \frac{1-(x/l)}{S_1}\right)\frac{x}{l}\right] \\ &= P.\frac{S_2[1-(x/l)]^2 + S_1(x/l)^2}{S_1 S_2} \end{aligned}$$

By substituting

$$S_1 = \alpha . S_2$$

$$\begin{aligned} y &= P.\frac{[1-(x/l)]^2 + \alpha(x/l)^2}{S_1} \\ &= \frac{P}{S_1}\left[\left(1-\frac{x}{l}\right)^2 + \alpha\left(\frac{x}{l}\right)^2\right] \end{aligned}$$

and

$$\frac{y}{y_{1\,\max}} = \left(1-\frac{x}{l}\right)^2 + \alpha\left(\frac{x}{l}\right)^2 \quad \text{(Fig. 71)}$$

The deviation of the workpiece shape from that of a right circular cylinder depends upon the difference between the maximum and minimum displacement of the workpiece axis relative to the cutting edge of the tool. In Fig. 72, this difference referred to the displacement (y_1) of centre 1 is

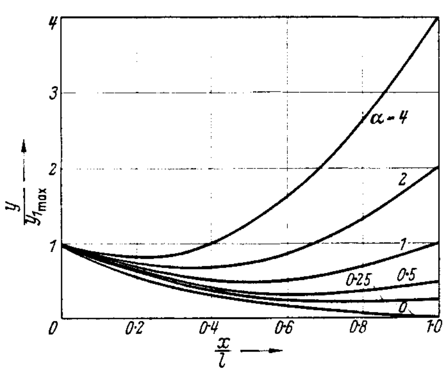


FIG. 71

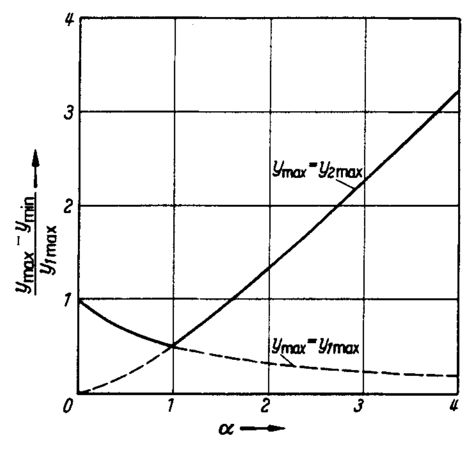


FIG. 72

plotted as a function of α. It will be observed that when $\alpha < 1$, the maximum displacement $y_{\max}$ occurs at the headstock centre ($y_{\max} = y_{1\,\max}$), whilst for $\alpha > 1$—i.e. if the stiffness at the tailstock centre is less than that at the headstock centre as is generally the case—the maximum displacement occurs at the tailstock centre ($y_{max} = y_{2\ max}$). The deviation will be a minimum when $\alpha = 1$, i.e. when the stiffnesses at both centres are equal. This shows that the stiffnesses of various parts of a machine must be balanced, and that high stiffness of one single part differing from that of any other part ($\alpha \neq 1$) is of relatively little value.

(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).

$$c_{\text{dyn}_{\min}} = 2\varrho c$$

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).

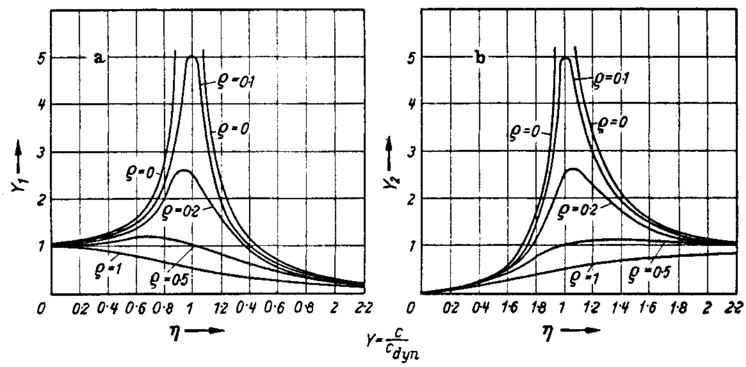


FIG. 73a and b.

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".

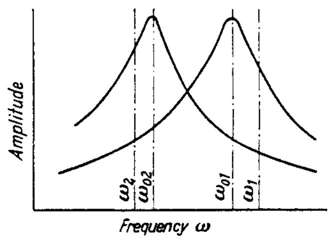


FIG. 74

Another example for relating the natural frequency to the possible exciting frequencies as a design criterion was shown by Piekenbrink[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 $m \times \ddot{\delta}$ precedes the damping force $\varrho . \dot{\delta}$ by $\pi/2$ and the spring force $c \times \delta$ by π.

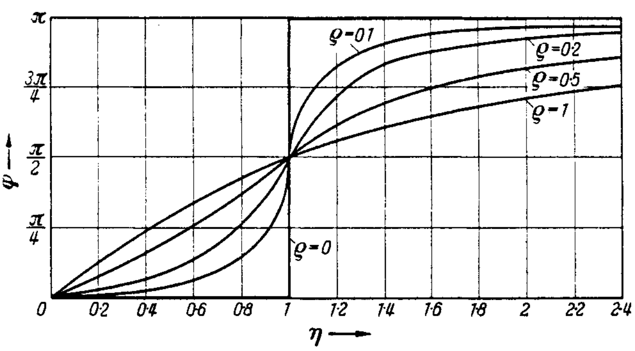


FIG. 75

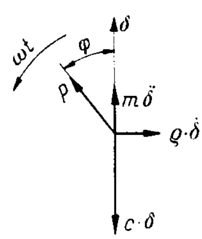


FIG. 76

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

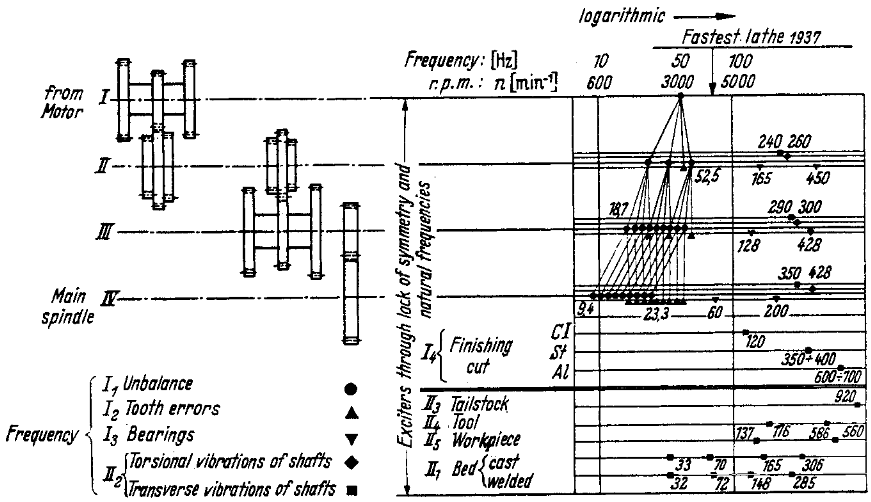


FIG. 77. Kienzle diagram.

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 possiblc 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.

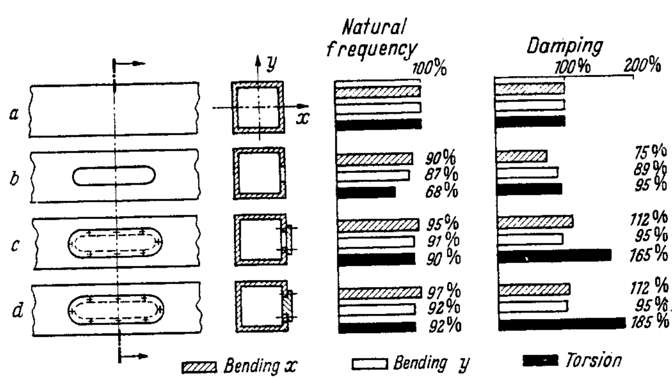


FIG. 78. Dynamic parameters of a box sectioned upright with apertures and cover plates (from Bielefeld).

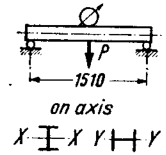
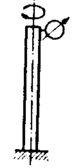
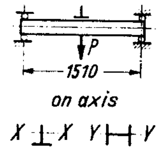
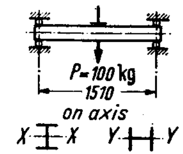
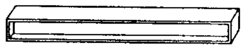
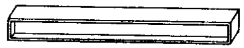
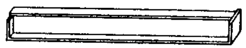
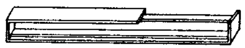
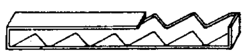
			Static stiffness			Natural frequency			Damping		
			Bending 		Torsion 	Bending 		Torsion 	Bending 		Torsion $T = 1$ mkg 
No.	Sketch of beam	Weight kg	c_B		c_D	ω_B		ω_T	ρ_B		ρ_T
			kg/μ	kg/μ	10^{-3} mkg/$\frac{\mu}{m}$	c.p.s.	μ c.p.s.	c.p.s.	10^{-3}	10^{-3}	10^{-3}
Ia		42	3·2	1·6	1·0	195	135	50·5	1·12	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 required 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.

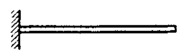
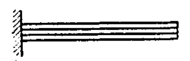
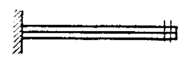
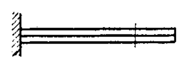
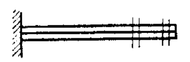
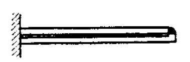
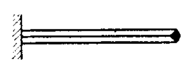
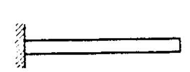
No.	Description	Sketch of beam and beam joints	Second moment of area compared with J_0	Natural frequency c.p.s.	Joint —	Increase in damping %
1	Single bar 10 mm thick		J_0	ω_0	—	—
2	Double bar not joined		$J_1 = 2J_0$	ω_0	free	0
3	Spot welded bars 1. 4 Spot welds		$J_2 = 5{\cdot}4J_0$	$1{\cdot}42\omega_0$	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).

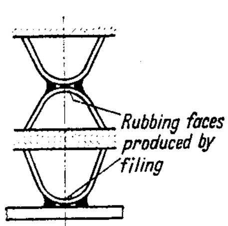


FIG. 81

(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]

(Fig. 80). It is important to remember[20] that the damping in welded built-up beams is high if (a) the contact pressure (or pre-load), which produces friction between contact faces, is as high as possible, and if (b) the joint effect of the welds is small so that the rubbing contact faces can move relatively to each other.

In an American publication,[21] the possibility of using welded joints as damping elements has been described. Apart from the "rubbing" effect, the damping found in such joints (Fig. 81) was explained by the presence of shrinkage stresses arising from the welding operation.

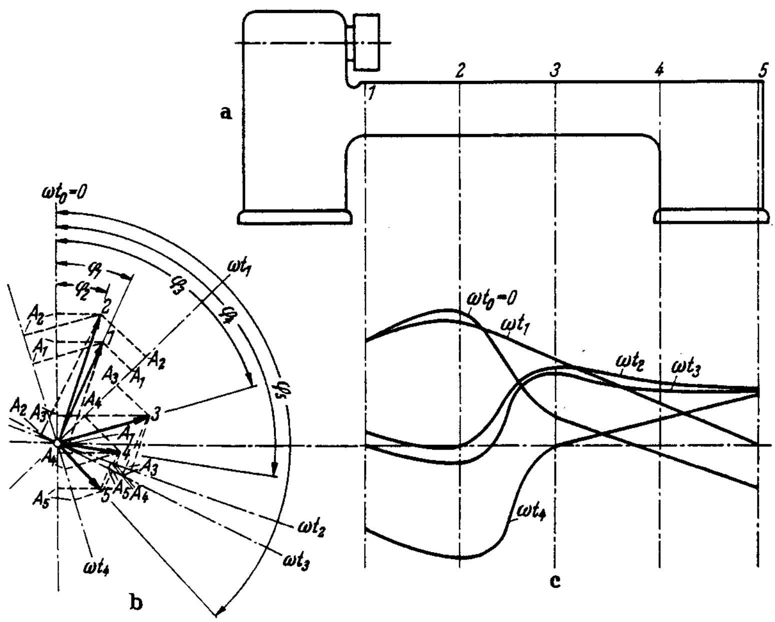


FIG. 82

(4) *Mode of vibration.*

Vibration systems with more than one degree of freedom arise in the machine tool as a whole and also in its constructional elements. As a result, there are phase differences between the vibration amplitudes at different points of one and the same element and of different elements in relation to each other. The mutual influences affect natural frequencies, damping, vibration amplitudes and phase angles. For a lathe bed which is excited by a pulsating bending force, Salje[22] has shown that no fixed nodes exist and that the dynamic deflexion shape becomes a function of position and time. The amplitudes, A_1, A_2, A_3, A_4 and A_5, measured at points 1, 2, 3, 4 and 5 respectively (Fig. 82a), are plotted in the vector diagram (Fig. 82b), in accordance with their phase angles (φ_1, φ_2, φ_3, φ_4 and φ_5) relative to the exciting force P. By drawing also on Fig. 82b the time axes for $\omega t_0 = 0$, ωt_1, ωt_2, ωt_3 and ωt_4, shown chain dotted, it is possible to determine the bending lines of Fig. 82c which occur at the times t_0, t_1, t_2, t_3 and t_4.

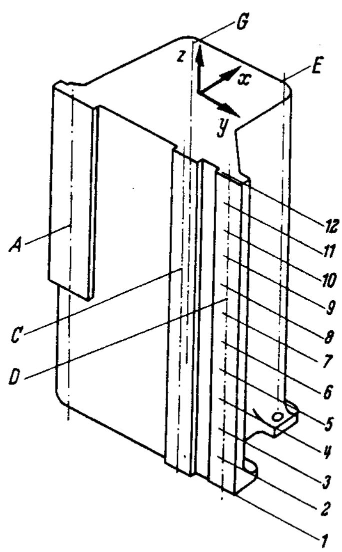


FIG. 83. Upright of a vertical borer

Figure 84 is a three-dimensional sketch showing the dynamic deflexions of a vertical boring machine upright (Fig. 83) for four different cases of resonance.[22] Although the upright was excited only by forces in the Y–Z plane (in the Z-direction at point 12 of slideway C and in the X-direction at point 6 of guide C), vibrations were observed both in the Y–Z and X–Z planes. The reason for these complex modes of vibration is thought to be in the shape of the upright, the main axes of which do not lie in one plane.

The conditions become even more complex when several elements are acting together. Honrath[23] investigated the conditions of a main spindle and its bearings. The increase of stiffness obtained by reducing the clearance (see page 52), not only results in higher natural frequencies (Fig. 85a), but also in increased damping (Fig. 85b). It is probable that both these effects are due to the increased friction between balls or rollers and their races. Figure 86 shows the dynamic bending lines of a lathe spindle with the workpiece supported between centres (Fig. 86a), and a comparison between static (c_{stat}) and dynamic (c_{dyn}) stiffness (Fig. 86b). This is taken from Hölken,[24] who also found that the spindle was the main cause of chatter vibrations because, although the spindle was subjected to a self-excited vibration, forced vibrations with a frequency below the critical one were produced in the tool holder through coupling with the cutting action. Coupling also caused torsional vibrations of small amplitude in the spindle.

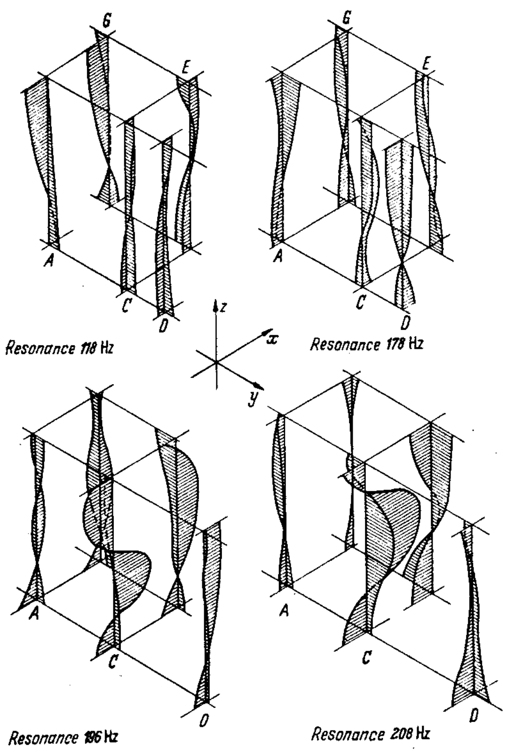


FIG. 84. Dynamic bending of a vertical borer at different resonance conditions (from Saljé).

The dynamic stiffness of the oil film in bearings and slideways may usually be assumed to be higher than the static one, as it depends upon the static stiffness and the damping capacity which is considerable.

If a machine is composed of many separate parts, not only the mutual effects of the various elements, but also the varying conditions of load application are again important.

For a universal milling machine the modes of vibration of various parts were measured[25] under the influence of an exciting force acting at an angle of 45° between table and milling arbor (Fig. 87). Measurements were taken in three transverse (A, B, C) and three longitudinal (1, 2, 3) positions of

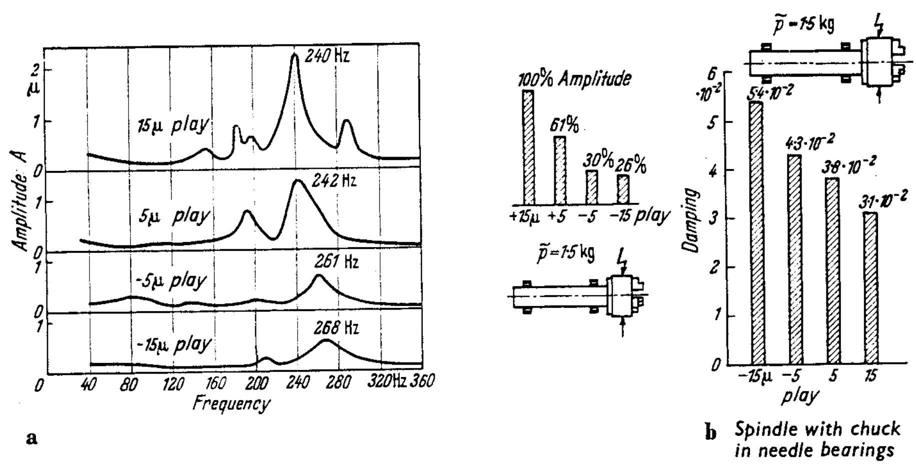


FIG. 85. Resonance amplitude (a) and damping (b) as a function of play in bearings (from Honrath).

the table (Fig. 88). Figures 89 to 97 show some of the dynamic shapes of base plate, knee, table, overarm and milling arbor as measured during the investigation. This investigation also showed

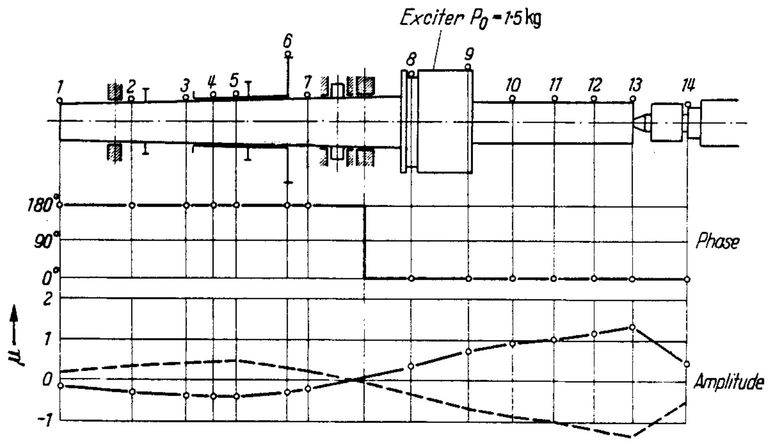


FIG. 86a. Dynamic bending line of a lathe spindle with workpiece (ω_0 = 168 c.p.s.) (from Hölken).

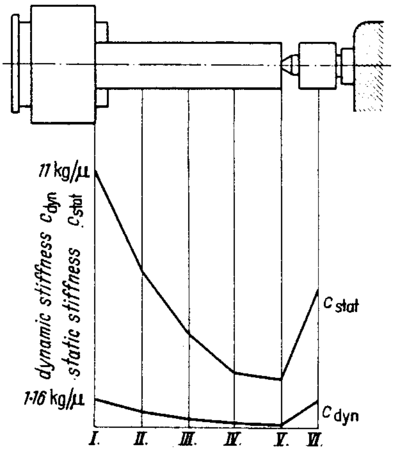


FIG. 86b. Static and dynamic stiffness at various points of the system spindle–workpiece–tail stock (from Hölken).

that vibrations occur in the direction of three axes even if the components of the exciting force act only in the direction of two axes, a fact which was later confirmed during cutting tests with a straight tooth milling cutter.

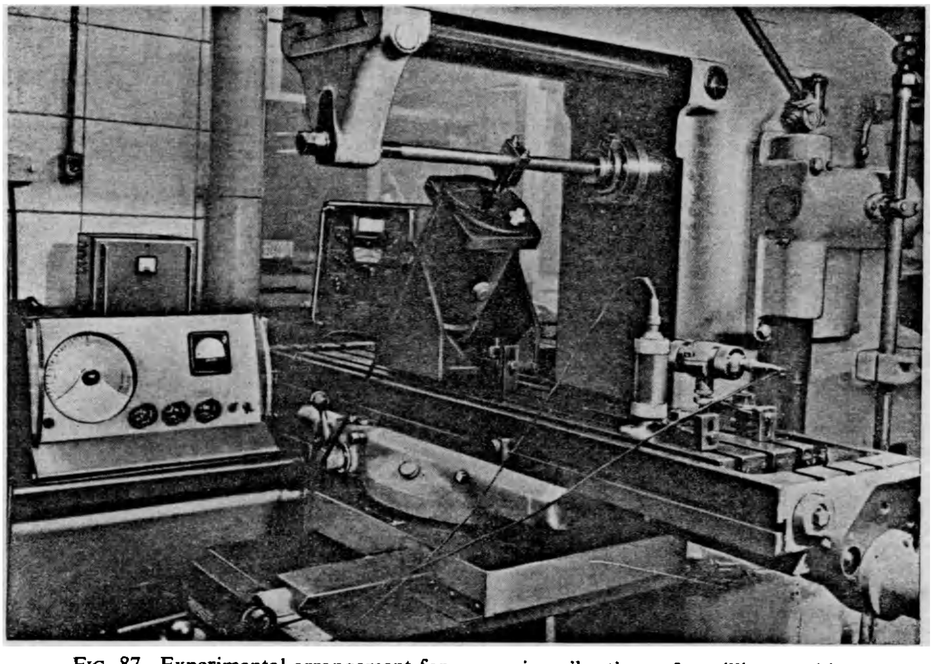


FIG. 87. Experimental arrangement for measuring vibrations of a milling machine.

The Kienzle diagrams for the spindle and feed drive of the milling machine used in this investigation are shown in Figs. 98 to 101.

As was to be expected, the cutting tests resulted in complex vibration configurations. However, it was possible to recognize clearly the forced vibrations, the frequency of which corresponded to the cutting force pulsations (rev/min times number of teeth of the milling cutter).

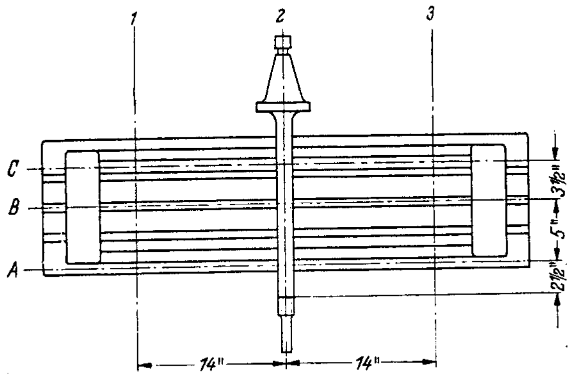


FIG. 88

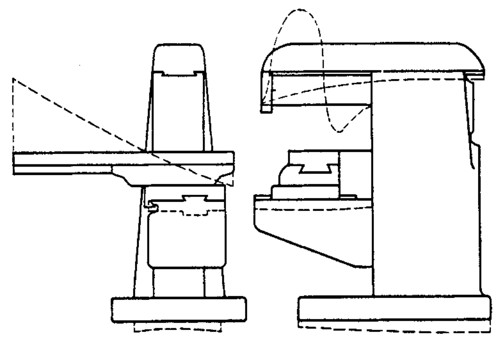
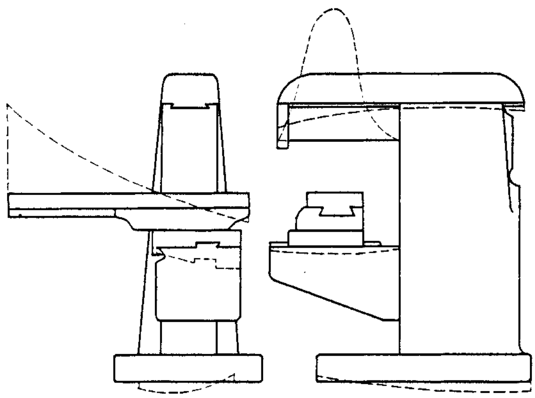
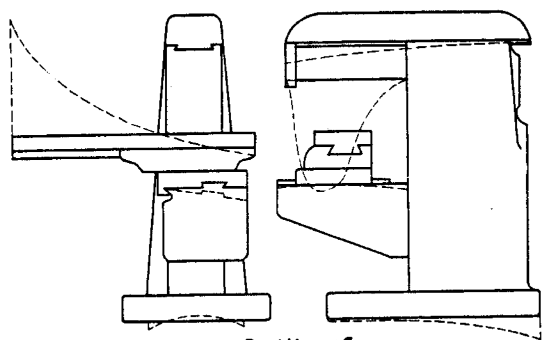


FIG. 89. Position 3A.
Vertical vibration $\omega_0 = 62\ \text{sec}^{-1}$.



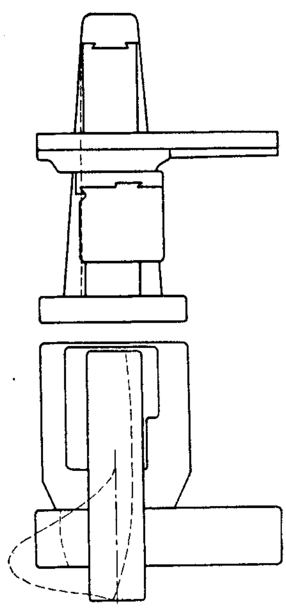
Position 3B
vertical vibration $\omega_0 = 66\ \text{sec}^{-1}$

FIG. 90



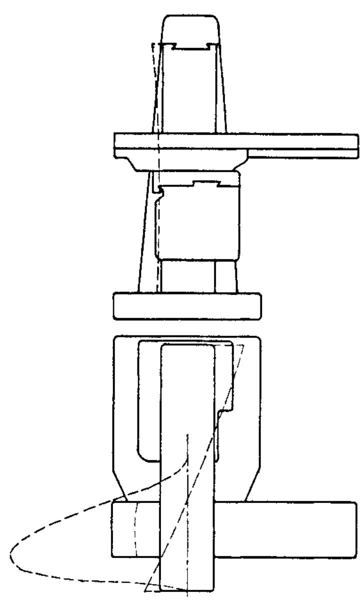
Position 3C
vertical vibration $\omega_0 = 66\ \text{sec}^{-1}$

FIG. 91



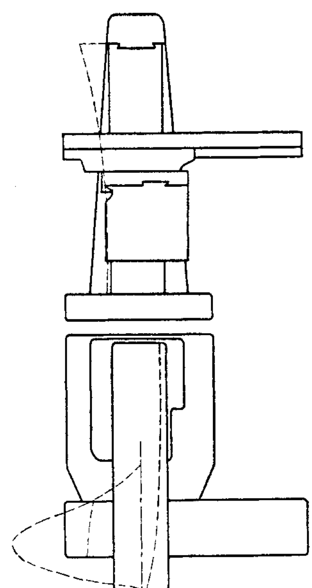
Position 1A
horizontal vibration
$\omega_0 = 62\ \text{sec}^{-1}$

FIG. 92



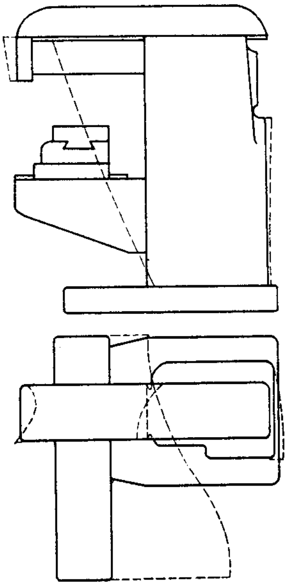
Position 1B
horizontal vibration
$\omega_0 = 76\ \text{sec}^{-1}$

FIG. 93



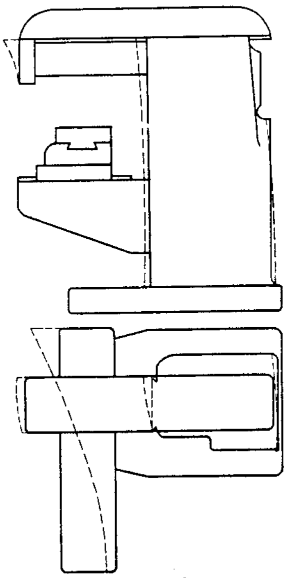
Position 1C
horizontal vibration
$\omega_0 = 70\ \text{sec}^{-1}$

FIG. 94



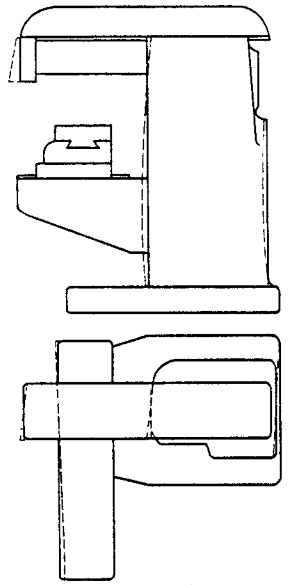
Position 1A
horizontal vibration
$\omega_0 = 76\,sec^{-1}$

FIG. 95



Position 1B
horizontal vibration
$\omega_0 = 42\,sec^{-1}$

FIG. 96



Position 1C
horizontal vibration
$\omega_0 = 95\,sec^{-1}$

FIG. 97

FIGS. 89 to 97. Dynamic bending lines of base, knee, table, overarm and milling arbor of a universal milling machine (from Koenigsberger and Said).

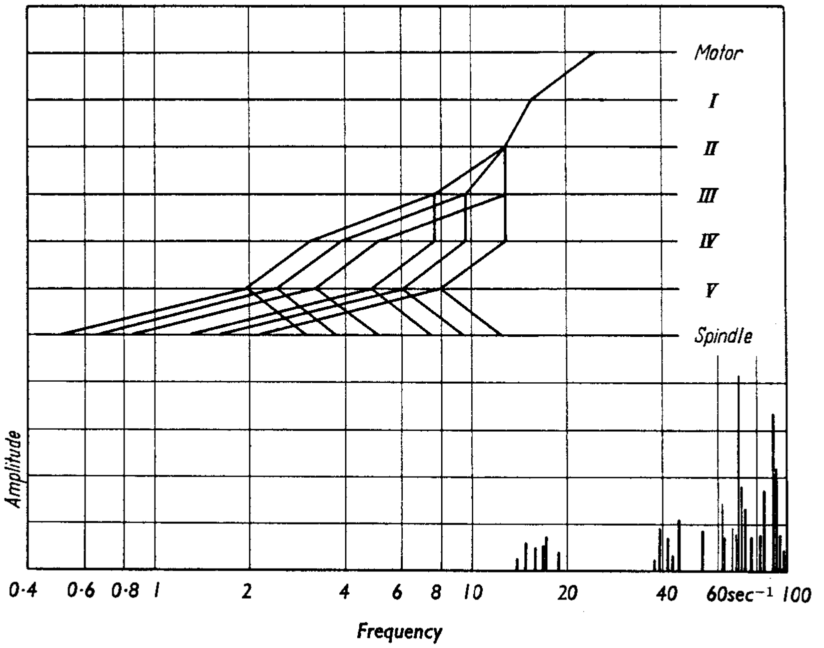


FIG. 98

The waviness of the milled surface is caused by:

(a) The milling arbor (see page 28) and the eccentricity of the milling cutter;

(b) Vibrations of the machine and its elements.

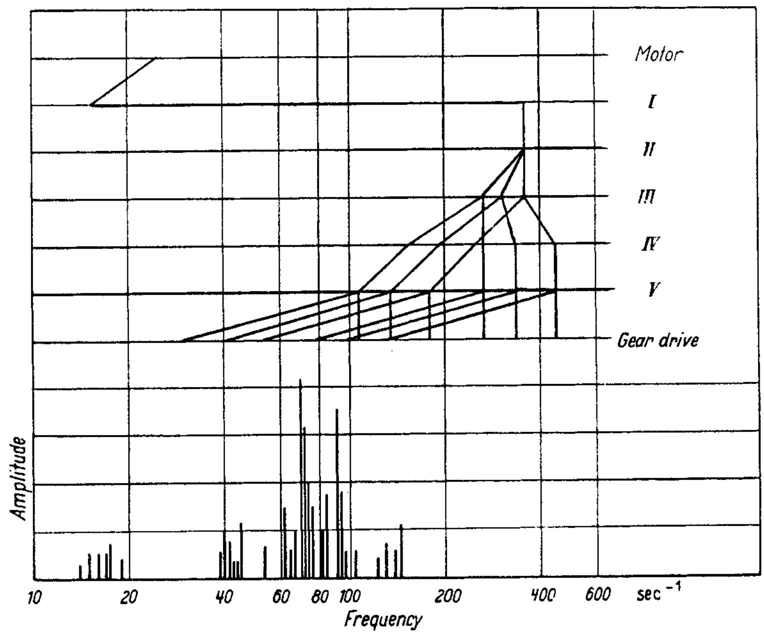


FIG. 99

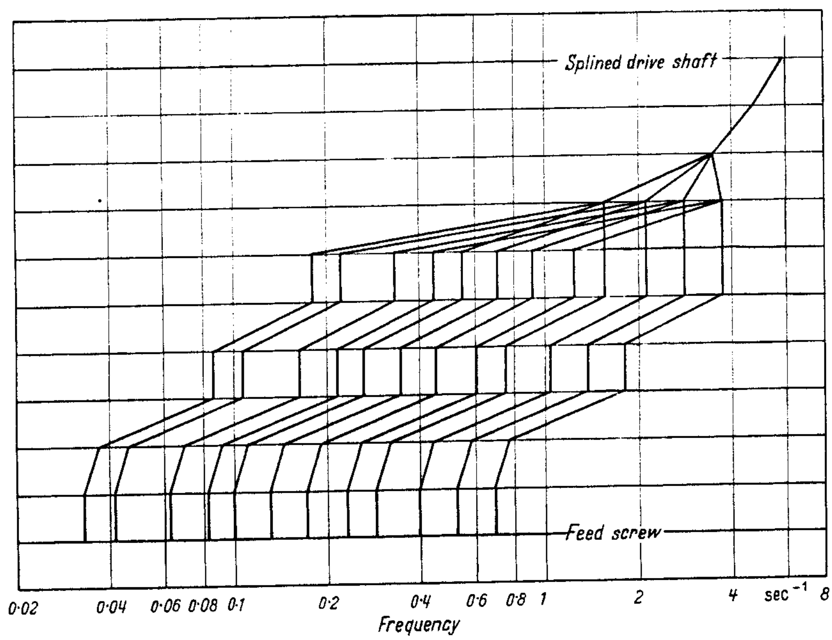


FIG. 100

That waviness of the machined surface which is determined by the geometry of the theoretical path described by the teeth of the milling cutter relative to the workpiece under the given conditions (diameter, eccentricity, number of teeth, number of revs. per minute, depth of cut and feed rate) may be called the "theoretical" waviness. Figure 102 shows a comparison between the actual

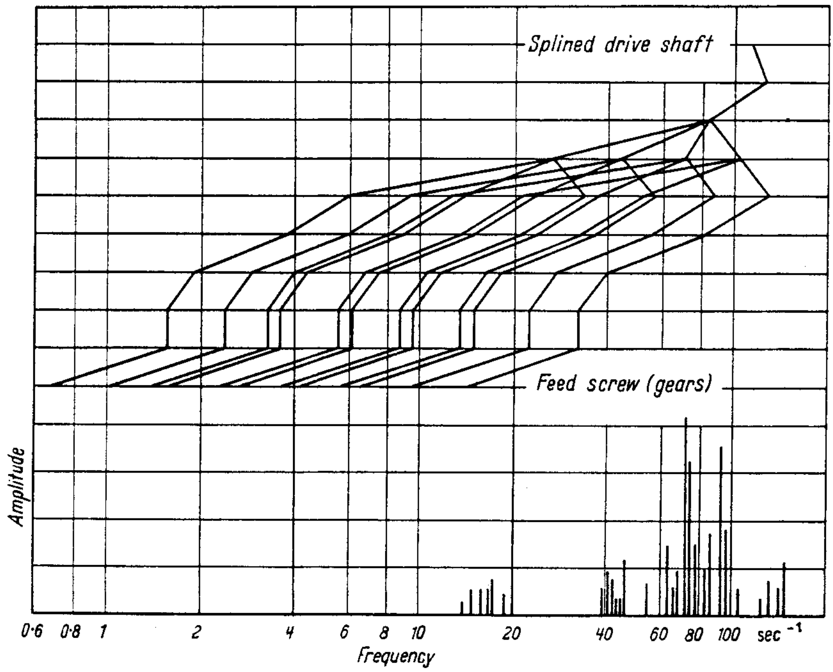


FIG. 101

FIGS. 98 to 101. Frequency graphs (after Kienzle) for the spindle and feed drive of the milling machine (see Figs. 89–97).

waviness and the theoretical one. The theoretical waviness is plotted at only one quarter of the vertical magnification applied for the actual surface. The surfaces machined during the tests showed considerably smaller actual amplitudes than would be expected theoretically, and in most cases it was found that the wavelengths on the machined surfaces were greater than the theoretical ones (Fig. 102). This may be caused by horizontal vibrations between workpiece and cutter. It indicates that vibration phenomena, although not normal to the machined surface, may affect the results considerably.

The dynamic performance of gear drives will depend largely upon the bending and torsional deformations of shafts, gears, etc.[26] From the point of view of the designer it is important to remember that the stiffness of a transmission between input and output decreases with increasing lengths of shafts and increasing distance between bearings, and rises in proportion to the dimensions at right angles to the shaft axes, i.e. shaft diameters and centre distances. Too low torsional stiffness in a drive may give rise to torsional vibrations as well as to inadmissible phase lag between input and output. In the main spindle drives of machine tools, the danger of such

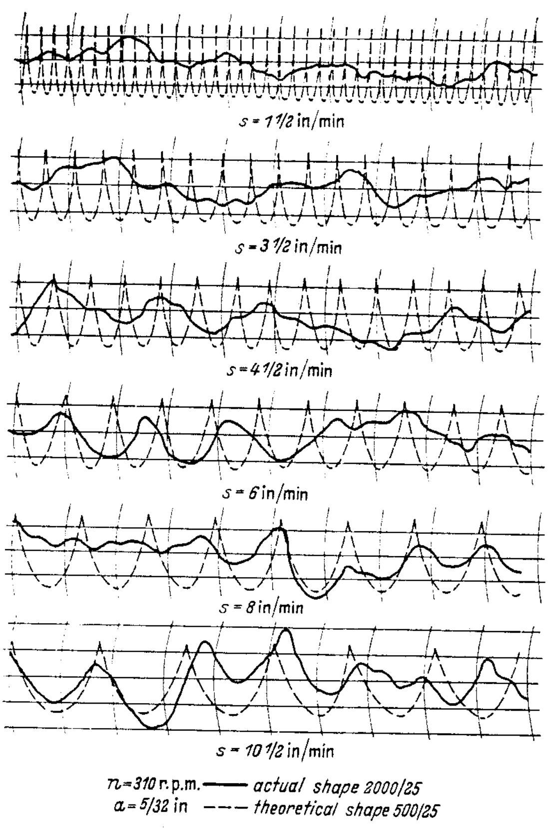
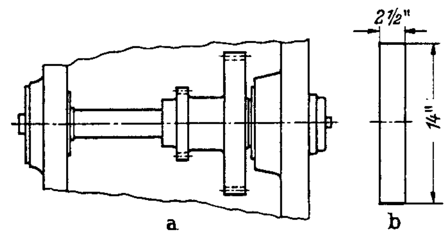


FIG. 102

torsional vibrations can be reduced by the application of masses which act as fly-wheels (see page 57). The effect of such masses can be shown by taking for an example the main spindle of a milling machine (Fig. 103a). The gear block on this spindle is situated immediately behind the main bearings, so that the twist between gear block and milling arbor drive can be kept within small limits. For the purpose of calculating the conditions on the spindle, an equivalent fly-wheel (moment of inertia I) (Fig. 103b) is substituted for the gear block. It may be assumed that the torque acting on the spindle due to the pulsating cutting force varies between T_1 and T_2 and that the angular velocity varies correspondingly between ω_1 and ω_2.



a b

FIG. 103

The kinetic energies of the fly-wheel at angular velocities ω_1 and ω_2 are

$$U_1 = \frac{I}{2}.\omega_1^2$$

and

$$U_2 = \frac{I}{2}.\omega_2^2$$

The change in kinetic energy is thus

$$\Delta U = U_1 - U_2 = \tfrac{1}{2}I(\omega_1^2 - \omega_2^2)$$
$$= \tfrac{1}{2}I(\omega_1 + \omega_2)(\omega_1 - \omega_2$$

The mean angular velocity is

$$\omega_{\text{mean}} = \tfrac{1}{2}(\omega_1 + \omega_2)$$

Suppose

$$\omega_1 - \omega_2 = u\,.\,\omega_{\text{mean}}$$

then

$$\Delta U = I\,.\,u\,.\,\omega_{\text{mean}}^2$$

and

$$u = \frac{\Delta U}{I\,.\,\omega_{\text{mean}}^2}$$

If the milling cutter has n_t teeth then for each entry of a tooth an energy pulsation caused by the variation of the torque will occur, i.e. over an angle $2\pi/n_t$,

$$\Delta U = \tfrac{1}{2}(T_1 - T_2).\frac{2\pi}{n_t}$$

and

$$u = \frac{(T_1 - T_2).\pi}{I\,.\,\omega_{\text{mean}}^2\,.\,n_t}$$

In the case of the spindle (Fig. 103), the following condition may be investigated:

Number of teeth of the milling cutter: $n_t = 6$
Spindle speed: $n = 775$ rev/min
Mean driving power (measured): $N = 4$ kW
Efficiency of the spindle drive: $\eta = 65\%$
Torque variation: 25%
The mean net torque is then:

$$T = 84\,000 \times \frac{4}{775} \times 0{\cdot}65 = 280 \text{ in lb}$$

and the torque variation

$$T_1 - T_2 = 0{\cdot}25 \times T = 70 \text{ in lb.}$$

$$\omega_{\text{mean}} = \frac{2\pi n}{60} = \frac{2\pi \times 775}{60} = 81 \text{ sec}^{-1}$$

$$I = \frac{\gamma}{g} \times b \times \frac{\pi d^4}{32}$$

$$= \frac{0{\cdot}26}{386} \times 2{\cdot}5 \times \frac{\pi \times 38\,400}{32} = 6{\cdot}4 \text{ lb in.sec}^2$$

and the cyclic variation of the angular velocity becomes:

$$u = \frac{70 \times \pi}{6{\cdot}4 \times 81^2 \times 6} \times 100 \approx 0{\cdot}1\,\%$$

Such a spindle-speed variation during milling may be considered permissible. The torque variation can, therefore, be absorbed by the fly-wheel without detrimental effect upon the driving elements, as long as the frequency of torque variation does not coincide with the natural frequency of one of the rotating parts in the drive.

Too low a stiffness in a spindle drive can in part be compensated for by the designer, for instance, by means of a fly-wheel. This is not always possible in a drive for feed or setting movements, especially when high setting or feed accuracy is essential, as they will be for automatic machines. If, for example, the slideways of a machine tool show friction conditions usually defined as "stick-slip" and causing jerking movements (see page 255)[27] this means that the coefficient of static friction between the moving surfaces is greater than the friction coefficient at very low velocity. When, under these conditions, a feed movement is initiated, all parts of the drive are deformed or strained until the output torque of the drive has increased to such an extent that it can overcome the static friction. At this moment, the movement commences, the friction coefficient drops and with it the resistance exerted by the slideway to the feed movement. The energy stored in the strained drive members is freed and the moving part is jerked forward by an amount greater than was originally intended. The higher, therefore, the stiffness of the drive gear, the smaller will be the difference between intended and actual movement of the driven part.

2. STANDARDIZATION OF SPINDLE SPEEDS AND FEED RATES: LAYOUT OF SPEED CHANGE GEARS[28]

The materials of tool and workpiece, the shape of the tool, the type of machining process and the required quality of the surfaces to be produced determine the optimum and most economical speeds for the two machining movements—the cutting and the feed movement (see page 1). Single-purpose machines, which are intended and designed for one single operation, often need only be designed for the one cutting speed and feed rate required for that operation. The designer of multi-purpose machines has, however, to provide a certain speed *range* which covers the requirements of different operations, types and shapes of workpieces and qualities of the surfaces that are to be machined.

The values of the required cutting speeds depend upon technical (cutting properties of the tools, surface finish of the machined surfaces) and economic considerations (minimum tool life between regrinds, grinding costs). The greater the variety of materials used for tools and workpiece, the wider is the required cutting speed range.

The accuracy with which the optimum cutting speed for a particular case is obtainable affects the operating efficiency of the machine. For roughing operations this is often measured by the rate of metal removal and during batch production by the quantity of workpieces which can be machined in a minimum time between regrinds.

Accuracy in the setting of feed rates enables the operator to obtain optimum chip sections, rates of metal removal and operation times. It is also important from the point of view of the surface finish produced by the particular machining operation.

The cutting and feed movements may be either rotary or rectilinear according to the machining process. However, in the majority of cases, the movements of the driving elements are rotary, so that changes in speeds are usually obtained by varying the rev/min of the driving shafts. For this reason, the problem of varying the rev/min is very important.

If the cutting movement is produced by the rotation of a part the cutting speed (v, ft/min) for an operating diameter (d, in.) and at n, rev/min of the workpiece (turning) or of the tool (drilling or milling) is:

$$v = \frac{\pi d n}{12}$$

$$n = \frac{12v}{\pi d}$$

When the specification for the working range of the machine requires that a diameter range of d_{max} to d_{min} and a speed range from v_{max} to v_{min} be covered the maximum obtainable rev/min must then be:

$$n_{max} = \frac{12v_{max}}{\pi d_{min}}$$

and the minimum

$$n_{min} = \frac{12v_{min}}{\pi d_{max}}$$

The so-called "speed-range ratio" of a speed-change gear is then

$$B = \frac{n_{max}}{n_{min}} = \frac{v_{max}}{v_{min}} . \frac{d_{max}}{d_{min}}$$

This rev/min range ratio is, therefore, equal to the product of the cutting speed-range ratio $B_v = v_{max}/v_{min}$ and the diameter-range ratio $B_d = d_{max}/d_{min}$, $B = B_v . B_d$.

For straight-line cutting movements (for instance, planing), the speed-range ratio depends only upon that of the cutting speeds; in the case of rotary cutting movements (turning, drilling and milling), the diameter-range ratio must also be considered. The cutting speed-range ratio for machining workpieces varying from alloy steels to light alloys can be very high. For instance, if carbide tools are used (v_{steel} about 300 ft/min, $v_{light\ alloy}$ about 6000 ft/min), the cutting speed-range ratio becomes $\frac{6000}{300} = 20$, and if high-speed steel tools are also to be used (v_{steel} approximately 60 ft/min), it can rise to $\frac{6000}{60} = 100$. It may be possible to obtain economically a spindle speed-range ratio of 100 by means of a gearbox. If, however, a diameter range of, say, $d_{max} = 2$ in. to $d_{min} = 0{\cdot}2$ in. ($2/0{\cdot}2 = 10$) is also specified (as may be the case in radial drilling machines), the speed-range ratio of $B = 100 \times 10 = 1000$ may become uneconomical, and it will become necessary to reduce the specified working range of the machine.

The ideal solution is the stepless drive because this ensures great accuracy of the spindle speeds. This will be discussed later. In the case of stepped drives, which cover the required spindle speed range with a given number of stepped rev/min, the optimum cutting speed can be obtained only in certain cases, and it is, therefore, necessary to establish an upper and a lower limit for the actual cutting speeds.

If the cutting speeds v are plotted as functions of the diameter d, a straight line $v = \pi dn/12$ appears for each spindle speed n_1, n_2, n_3, n_4, etc. This is shown in Fig. 104 together with constant upper (v_u) and lower (v_l) speed limits. If the spindle rotates at n_1 rev/min, the working diameter must not be greater than d_1 nor smaller than d_2, as otherwise the cutting speed will fall outside the permissible limits v_u and v_l. For a working diameter which is smaller than d_2, it is necessary to provide a higher spindle speed n_2, so that with n_2 the maximum permissible speed v_u is reached for the diameter d_2. At this spindle speed n_2 it is possible to work with diameters ranging between d_2 and d_3, without exceeding the limits of maximum and minimum cutting speeds. It is, therefore, possible to establish the following relations for a number of speeds n_1, n_2, n_3, n_4, etc., and a number of diameters d_1, d_2, d_3, d_4, etc.

$$n_1 = \frac{12v_u}{\pi d_1} = \frac{12v_l}{\pi d_2}$$

$$n_2 = \frac{12v_u}{\pi d_2} = \frac{12v_l}{\pi d_3}$$

$$n_3 \quad \frac{12v_u}{\pi d_3} = \frac{12v_l}{\pi d_4}$$

$$\cdots\cdots\cdots\cdots\cdots\cdots$$

$$n_{n-1} = \frac{12v_u}{\pi d_{n-1}} = \frac{12v_l}{\pi d_n}$$

$$n_n = \frac{12v_u}{\pi d_n} = \frac{12v_l}{\pi d_{n+1}} \quad \text{etc.}$$

The step between two consecutive spindle speeds in a speed range, which makes possible the machining of a certain diameter range at cutting speeds between the established limits v_u and v_l is, therefore, constant:

$$\frac{n_n}{n_{n-1}} = \frac{v_u}{v_l}$$

In other words, the spindle speed range is a geometric progression with a ratio (step)

$$\varphi = \frac{v_u}{v_l}$$

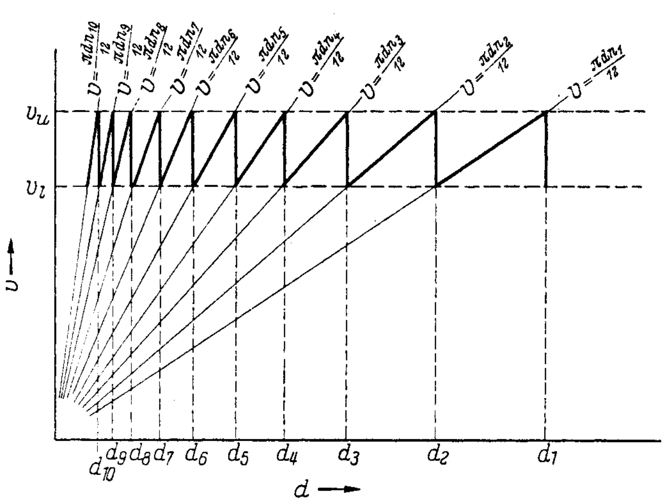


FIG. 104. Saw diagram (geometric progression).

The "saw diagram" (Fig. 104) shows these relations, and from it the increased width of the diameter range in the case of large diameters and low spindle speeds should be especially noticed.

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.

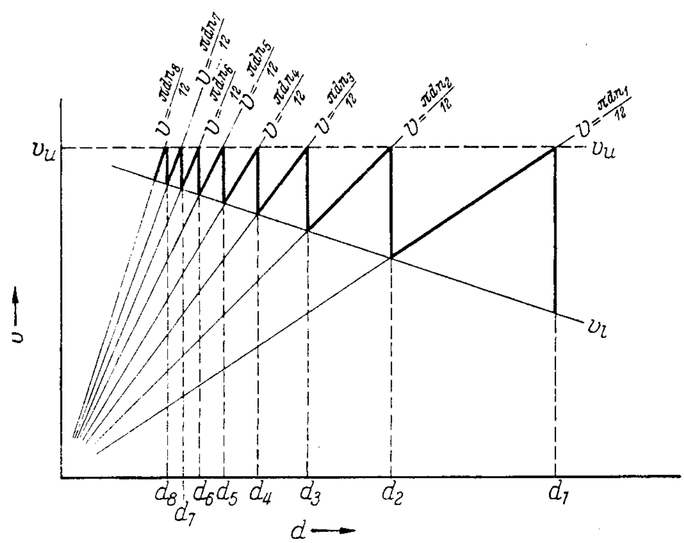


FIG. 105. Saw diagram (arithmetic progression).

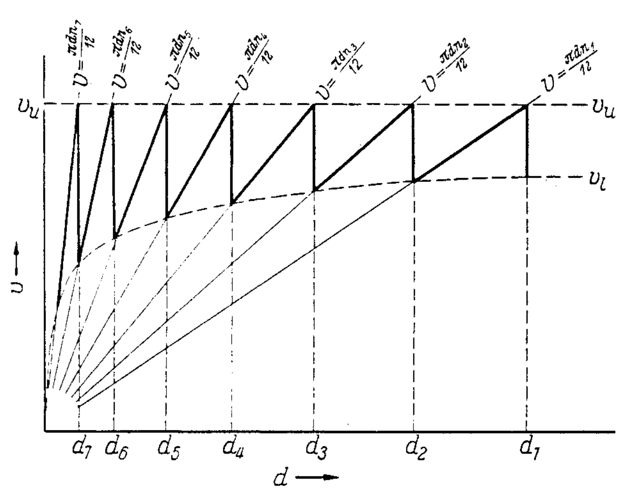


FIG. 106. Saw diagram (logarithmic progression).

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 $\varphi = 1{\cdot}6$		Range R 20/6 $\varphi = 2$		
					(1400)	(2800)			
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			63
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:

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

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.

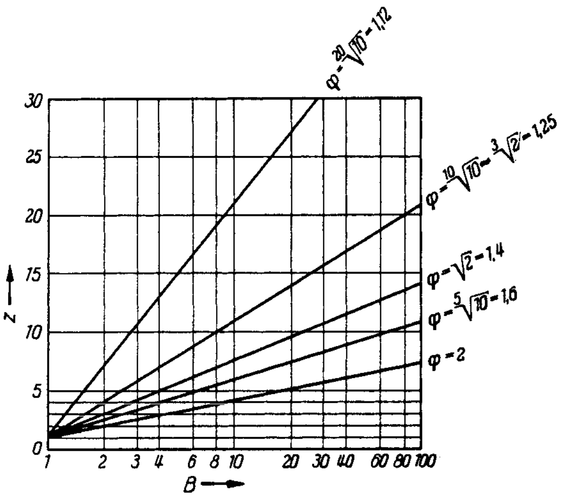


FIG. 107. Relation between speed-range ratio B, number of steps z at standard ratios φ.

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 transmission 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.

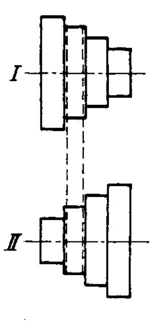


FIG. 108

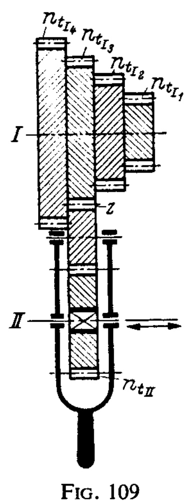


FIG. 109

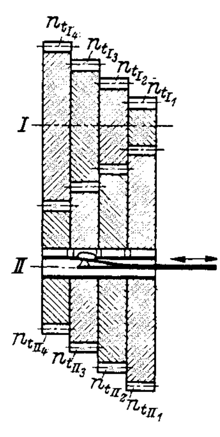


FIG. 110

* 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.

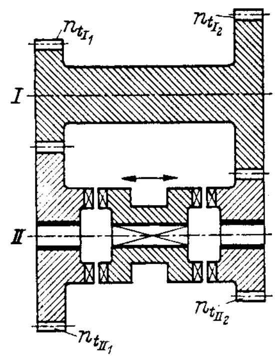


FIG. 111

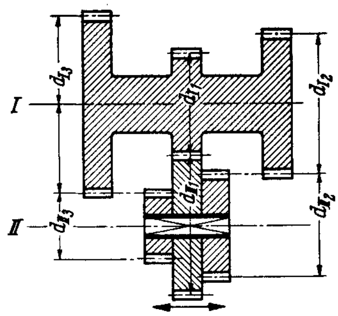


FIG. 112

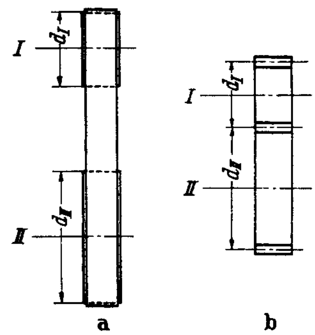


FIG. 113

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:

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

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:

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

(φ_N being a standard step ratio). In other words, the transmission ratios are powers of standard step ratios. In the case of a geometric speed range, the step ratio is:

$$\varphi = \frac{n_{II_n}}{n_{II_{n-1}}} = \frac{u_n}{u_{n-1}}$$

and so

$$u_n = \varphi . u_{n-1}$$

If, therefore, a geometrically-stepped speed range is required, the transmission ratios must also form a geometrically-stepped range with the same step ratios. In the case of standard speed ranges ($\varphi = \varphi_N$), all transmission ratios must also be stepped in accordance with the standard step ratios.

For reasons of space limitations, or when the numbers of gear teeth or the tangential velocities on the pitch circle have to be limited, it may be necessary to keep the transmission ratios between two gears below 2 : 1 and above 1 : 4.

$$u_{\max} = \frac{n_{II_{\max}}}{n_I} = \frac{2}{1}$$

and

$$u_{\min} = \frac{n_{II_{\min}}}{n_I} = \frac{1}{4}$$

This would mean that in the case of a two-axes gearbox, the maximum possible speed-range ratio would be:

$$B_{\max} = \frac{u_{\max}}{u_{\min}} = \frac{n_{II_{\max}}}{n_{II_{\min}}} = 8$$

Moreover, it must be remembered that with the exception of Norton type and draw-key type gearboxes, the numbers of steps, which can be obtained with a two-axes gearbox, is limited.

(2) *Layout of intermediate reduction gears.* It is possible to produce gearboxes which cover larger speed ranges with greater numbers of steps by arranging several two-axes gearboxes in series. In such cases, intermediate transmission can either be completely independent of each other, or the driven gear of one serves as the driving gear for the next part-drive in a composite arrangement (see page 117).

The number of steps in a gearbox with more than two axes is equal to the product of the numbers of steps of the part drives (intermediate transmissions). If, in the case shown in Fig. 114, the gear ratios in the first part-drive (shaft *I* to shaft *II*) are

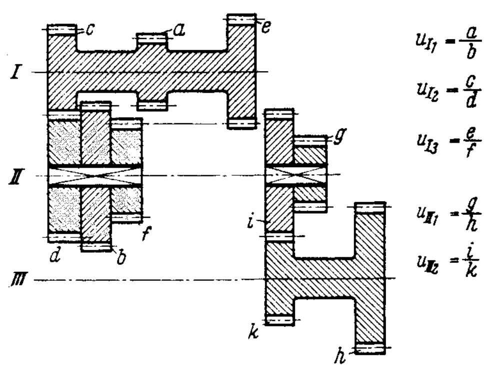


FIG. 114

$$\frac{n_{II_1}}{n_I} = u_{I_1}; \quad \frac{n_{II_2}}{n_I} = u_{I_2}; \quad \frac{n_{II_3}}{n_I} = u_{I_3}$$

and those of the second part-drive (shaft *II* to shaft *III*) are

$$\frac{n_{III_1}}{n_{II_1}} = \frac{n_{III_2}}{n_{II_2}} = \frac{n_{III_3}}{n_{II_3}} = u_{II_1}$$

$$\frac{n_{III_4}}{n_{II_1}} = \frac{n_{III_5}}{n_{II_2}} = \frac{n_{III_6}}{n_{II_3}} = u_{II_2}$$

the total transmission ratios are:

$$e_1 = u_{I_1} . u_{II_1} = \frac{n_{III_1}}{n_I} \qquad e_4 = u_{I_1} . u_{II_2} = \frac{n_{III_4}}{n_I}$$

$$e_2 = u_{I_2} . u_{II_1} = \frac{n_{III_2}}{n_I} \qquad e_5 = u_{I_2} . u_{II_2} = \frac{n_{III_5}}{n_I}$$

$$e_3 = u_{I_3} . u_{II_1} = \frac{n_{III_3}}{n_I} \qquad e_6 = u_{I_3} . u_{II_2} = \frac{n_{III_6}}{n_I}$$

$$e_6 : e_5 : e_4 : e_3 : e_2 : e_1 = n_{III_6} : n_{III_5} : n_{III_4} : n_{III_3} : n_{III_2} : n_{III_1} = \varphi$$

If, therefore, the output speeds

$$n_{III_1} ; n_{III_2} ; n_{III_3} ; n_{III_4} ; n_{III_5} ; n_{III_6}$$

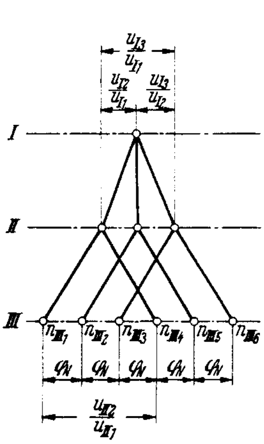


FIG. 115

are parts of a standard speed range with the standard ratio φ_N, the overall transmission ratios between the input and output shafts, e_1, e_2, e_3, e_4, e_5 and e_6 form also a geometric progression with the ratio φ_N.

The layout of a gearbox with more than two axes can be shown graphically in a diagram in which the speeds are plotted horizontally on a logarithmic scale and the shafts are shown as horizontal parallel lines at equal distances from each other (Fig. 115).* With

$$\frac{n_n}{n_{n-1}} = \varphi_N \quad \text{and} \quad \log(n_n) - \log(n_{n-1}) = \log \varphi_N$$

the speed values appear on the logarithmic scale at equal distances ($\log \varphi_N$), and the transmission ratios between two axes are indicated by the horizontal distances between the corresponding speed values ($\log n_{II} - \log n_I = \log u$). If, moreover, the distances between axes *I, II, III*, etc., are equal, the slope of the lines joining the speed values on different axes are an indication of the corresponding gear ratios.

The number of possible layouts for gear drives which produce a certain number of steps is limited, and Germar[28] has covered these possibilities for gear drives producing from four to eighteen steps. For example, in the case of a six-step drive, four layouts are possible (Fig. 116).

In order to show how the various possibilities can be carefully assessed, these four layouts may be considered in greater detail. If the ranges of transmission ratios and, therefore, the differences between the torques which have to be transmitted by the various gears are small, it is possible to design all gears within one particular part drive with equal pitch and equal tooth width. If, on the other hand, certain gears have to transmit considerably higher torques and, therefore, higher tooth loads, these will have to be larger than the others and require more space. In the case of the layout in Fig. 116a, the highest transmission ratios occur in the second part drive, so that only the gears transmitting at the ratio u_{II_1}, and thus producing the three lowest speeds, have to be stronger. In the case of Fig. 116b, the two lowest speeds are transmitted by two sets of gears (transmission ratios u_{II_1} and u_{II_2}). Moreover, in this case, the highest transmission ratio, i.e. the greatest speed change, occurs in the first part drive. In this case, therefore, the gears for reduction u_{I_1} must also be stronger. Similar conditions apply to the cases of Figs. 116c and 116d. A further consideration concerns the numerical values of the transmission ratios (see page 77). These can easily be determined by counting the horizontal distances between various steps, because in each part drive:

$$\frac{u_{\max}}{u_{\min}} = \frac{n_{II\max}}{n_{II_{\min}}}$$

* This layout diagram and the later-mentioned speed diagram have been introduced for the first time by R. Germar, see Reference 28.

and

$$\log\left(\frac{u_{max}}{u_{min}}\right) = \log(n_{II_{max}}) - \log(n_{II_{min}})$$

In the layouts (Figs. 116a and 116c), the largest transmission range lies in the second part drive:

$$\frac{u_{II_2}}{u_{II_1}} = \varphi^3 \quad \text{and} \quad \frac{u_{II_3}}{u_{II_1}} = \varphi^4 \qquad \text{respectively,}$$

whilst in the layouts (Figs. 116b and 116d), it is found in the first part drive

$$\frac{u_{I_3}}{u_{I_1}} = \varphi^4 \quad \text{and} \quad \frac{u_{I_2}}{u_{I_1}} = \varphi^3 \qquad \text{respectively.}$$

In addition, these transmission ranges should not exceed the values mentioned on page 77. Moreover, the arrangements (Figs. 116a and 116c) are also preferable for reasons of tooth strength and space requirements. This means that the theoretically optimum layout is that shown in Fig. 116a, which requires the smallest transmission range.

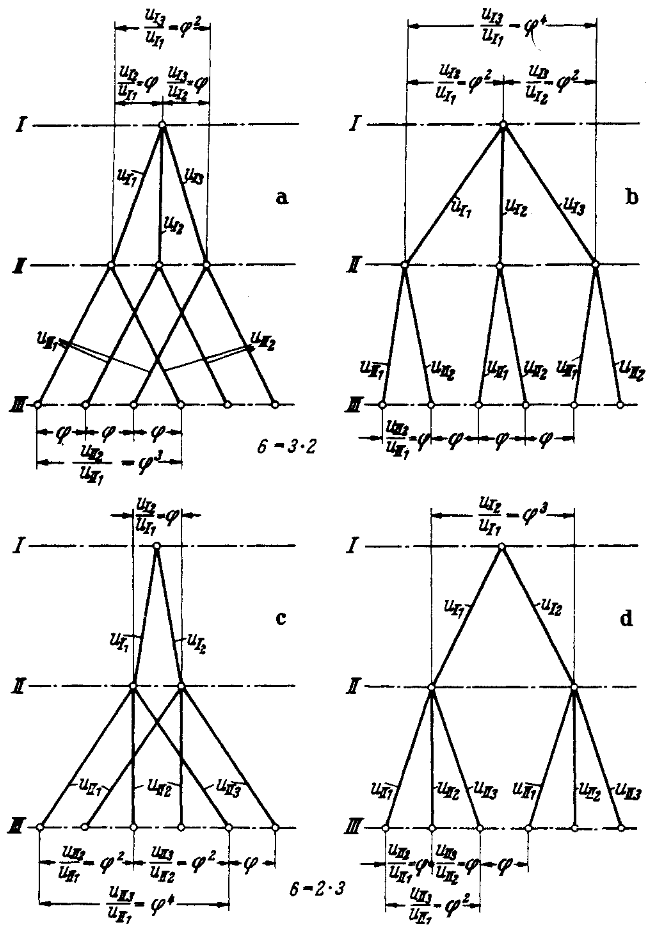


FIG. 116

The layout diagram can also prove very valuable if gearboxes for different working ranges or for different numbers of steps have to be produced with a maximum possible number of identical components. This is illustrated in the following example.

In order to satisfy the requirements of various customers, the spindle drive of a universal milling machine was to be designed in such a manner that the machine could be supplied with either 12 or 18 spindle speeds. In both cases, the speed range had to be approximately 45 to 50. This led to standard step ratios of 1·4 (speed-range ratio for 12 speeds $B_{12} = 45$) and 1·25 (speed-range ratio for 18 speeds $B_{18} = 50$) being chosen. It was also specified that in the part drives of the speed change gearbox (sliding gears), gear blocks with not more than three gears should be used. The layouts shown in Figs. 117 and 118 were available.[28]

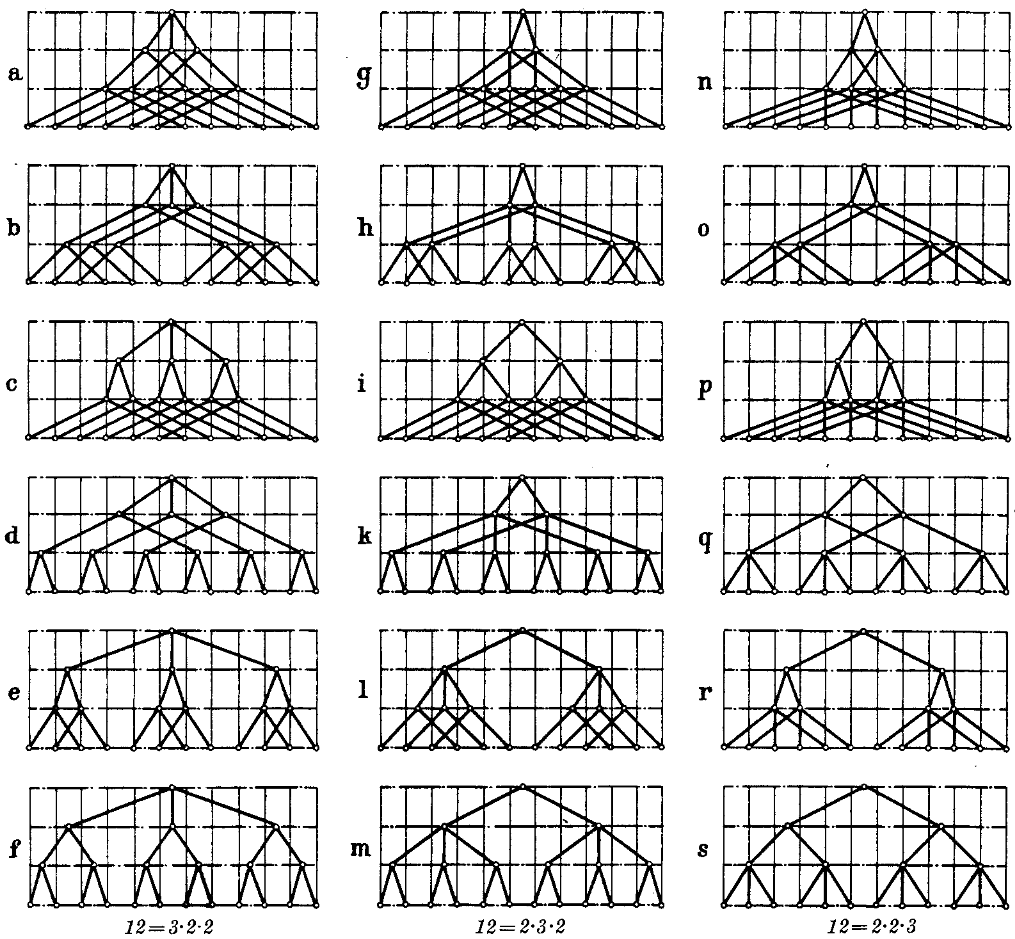


FIG. 117. Layouts for 12-step gear drives (from Germar).

For the reasons discussed in connexion with Figs. 116 (a) to (d), the layouts (Figs. 117g and 118a) showed to advantage. In addition, they were also very suitable from the point of view of converting a 12-speed gearbox to one with 18 speeds or vice versa, as will be shown presently. Figures 119 (a) and (b) show side by side the layouts for a 12- and 18-speed gearbox respectively ($\varphi_{12} = 1{\cdot}4$, $B_{12} = 45$ and $\varphi_{18} = 1{\cdot}25$, $B_{18} = 50$). The largest transmission range ratio between the last two shafts, (*III* and *IV*), is in the case of the 12-speed gearbox φ_{12}^6 (Fig. 119a); the corresponding transmission range ratio between the last two shafts (*III* and *IV*), of the 18-speed gearbox is φ_{18}^9 (Fig. 119b). Moreover, $\varphi_{12}^6 = 1{\cdot}4^6 = (\sqrt{2})^6 = 8$, and $\varphi_{18}^9 = 1{\cdot}25^9 = (\sqrt[3]{2})^9 = 8$, and therefore $\varphi_{12}^6 = \varphi_{18}^9$, so that the part drives between shafts (*III* and *IV*) can be identical for both the 12- and 18-speed gearboxes. The same consideration applies for the part drive between shafts (*II* and *III*) because again

$$\varphi_{12}^2 = \varphi_{18}^3 = 2 \quad \text{and} \quad \varphi_{12}^4 = \varphi_{18}^6 = 4$$

This means that only the transmission ratios between shafts (*I* and *II*) of the gearboxes differ

$$(\varphi_{12} = 1{\cdot}4, \quad \varphi_{18} = 1{\cdot}25 \quad \text{and} \quad \varphi_{18}^2 = 1{\cdot}6),$$

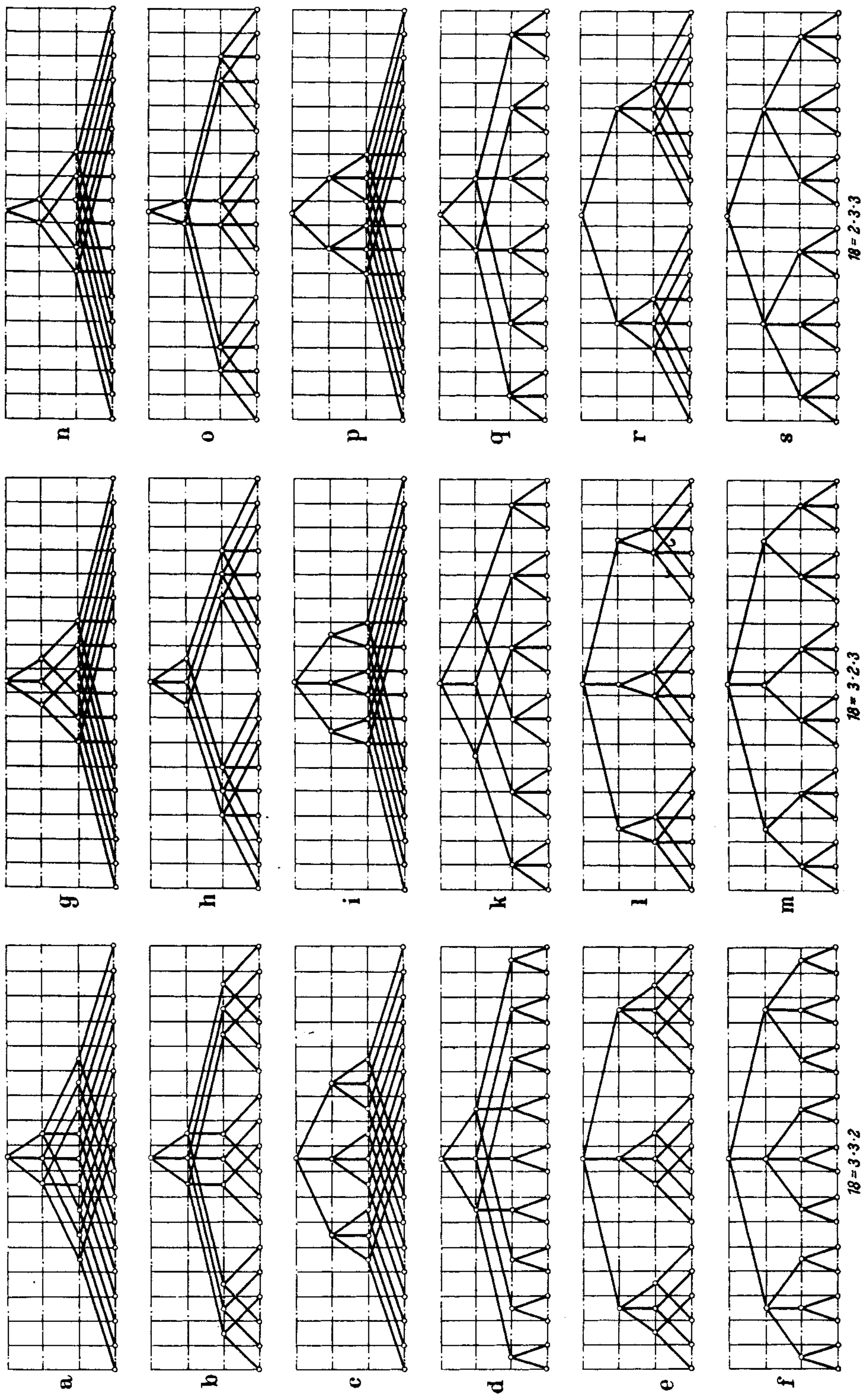


FIG. 118. Layouts for 18-step gear drives (from Germar).

so that the only difference between the two gearboxes must be in the gears between shafts (*I* and *II*). It may be stressed at this point that this very important fact could be found without any calculation of gear ratios, numbers of teeth, etc., and merely by means of carefully considering the layouts of the gear drives in question.

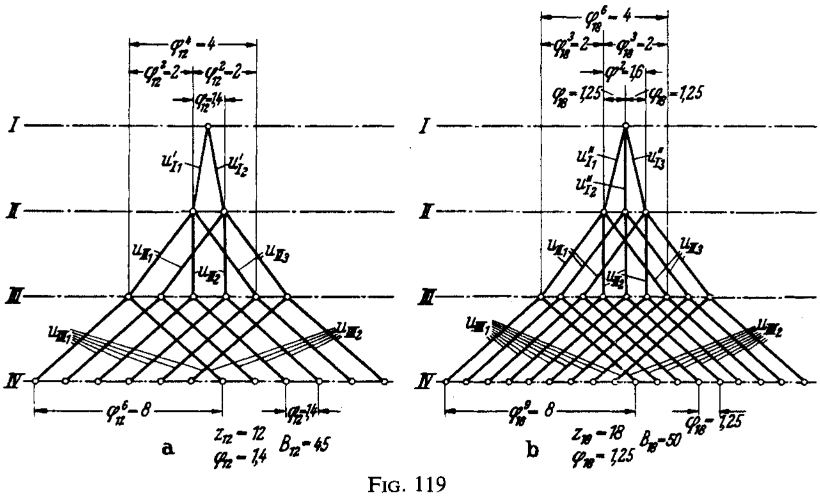


FIG. 119

(3) *Transmission ratios.* The layout diagram shows the relations between gear ratios and steps without giving any information concerning their absolute values. These are represented in the speed diagram,[28] in which the logarithms of the actual speed values are plotted on the horizontal axis of the layout diagram (Fig. 120). In the case of a standard gearbox for a geometric speed progression, the steps, i.e. the spacing between two speeds of one shaft, are the same as in the previously described layout diagram. However, the points indicating the actual speeds of the different shafts are not arranged symmetrically, but are displaced in accordance with the actual speed values. For the same reason, the lines joining the speeds between two axes are no longer arranged symmetrically although, of course, equal gear ratios are again represented by parallel lines and the transmission ranges in a part drive remain unaltered:

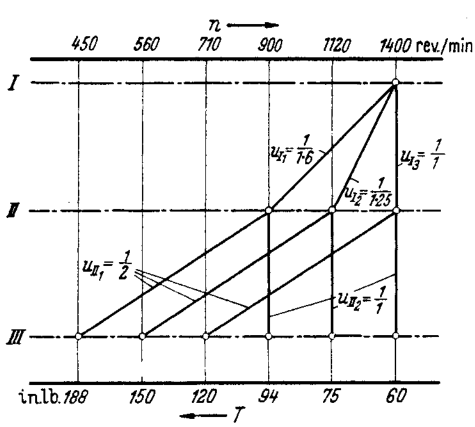


FIG. 120

$$\frac{u_{max}}{u_{min}} = \frac{n_{II_{max}}}{n_{II_{min}}}$$

Whilst, therefore, the layout diagram covers all gearboxes of a given type and arrangement, the speed diagram defines a particular design for the same type of gearbox and the absolute values of its speeds and gear ratios.

As the speed diagram indicates directly the actual magnitudes of the various speeds, it is possible to insert also the values of the torques (T), which the different shafts have to transmit in the case of a given power (N), at different speeds (n), because these torques are inversely proportional to the speed ($T = 84{,}000N/n$; T: in lb, N: kW and n: rev/min) (Fig. 120). In this manner, the basis for the calculation of gears, numbers and size of teeth is given.

As an example, consider the gearbox of the type shown in Fig. 114 (layout diagram Fig. 115), which is designed to produce six speeds from 450–1400 rev/min (step ratio $\varphi = 1{\cdot}25$), on shaft *III*, the input speed of shaft *I* being 1400 rev/min. The maximum reduction ratio is equal to:

$$\frac{u_{II_2}}{u_{II_1}} = \varphi^3 = 2$$

This can be covered by one set of gears. For this reason, it is possible to avoid gear ratios larger than 1/1, and the maximum speed of shaft *III*, which is equal to the speed of input shaft *I*, can be obtained by ratios

$$u_{I_3} = \frac{1}{1} \quad \text{and} \quad u_{II_2} = \frac{1}{1}$$

The other gear ratios are then obtained as follows:

$$u_{I_2} = \frac{u_{I_3}}{\varphi} = \frac{1}{1{\cdot}25}; \qquad u_{I_1} = \frac{u_{I_3}}{\varphi^2} = \frac{1}{1{\cdot}6} \qquad u_{II_1} = \frac{u_{II_2}}{\varphi^3} = \frac{1}{2}.$$

and the speed diagram (Fig. 120) may then be drawn. The standard speeds of the geometric progression (ratio $\varphi = 1{\cdot}25$) can be obtained from Table 11 and inserted directly in the speed diagram.

The torques which the different shafts have to transmit if the power is constant ($N = 1$ kW), and which are inversely proportional to their speeds, are plotted on an axis parallel to that for the speeds.

The foregoing considerations were all based on the one requirement that the speeds of all shafts, or at least those of the output shaft, form part of a range of standard speeds under load. However, for the determination of pulley diameters, gear diameters and numbers of teeth, the following points must also be considered:

(a) With the exception of cone pulley drives, which employ a jockey pulley (Fig. 121) and Norton type gearboxes (Fig. 109), the sum of the diameters of two corresponding pulleys or gears on two shafts must be constant.*

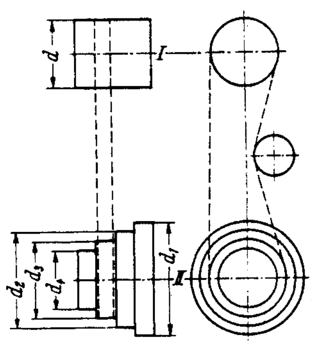


FIG. 121

(b) In the case of all gear transmissions, the gear ratios must be so chosen that they can be obtained with whole numbers of teeth. For this reason, very accurate diameter calculations are possible only for belt transmissions whilst in the case of gear transmissions not only the pitch circle diameters, but also the actual numbers of teeth have to be calculated.

In the case of a belt drive with jockey pulley (Fig. 121):

$$u_1 = \frac{d}{d_1}; \qquad u_2 = \frac{d}{d_2}; \qquad u_3 = \frac{d}{d_3}; \qquad u_4 = \frac{d}{d_4}$$

and, therefore

$$\frac{u_4}{u_3} = \frac{d_3}{d_4}; \qquad \frac{u_3}{u_2} = \frac{d_2}{d_3}; \qquad \frac{u_2}{u_1} = \frac{d_1}{d_2}$$

If a standard geometric progression of speeds with the ratio φ is to be produced:

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

and, therefore

$$\frac{d_1}{d_2} = \frac{d_2}{d_3} = \frac{d_3}{d_4} = \varphi$$

The diameters of a cone pulley working in conjunction with a jockey pulley must be stepped in accordance with a geometric progression the ratio of which is equal to that of the standard speed

* This is not exactly correct in the case of cone pulley drives with two cone pulleys. However, the assumption of a constant sum of the diameters is permissible when the centre distance between the axes exceeds a certain minimum value (see page 84).

range that is to be produced. If a drive employing two cone pulleys (Fig. 122) is to be used, then:

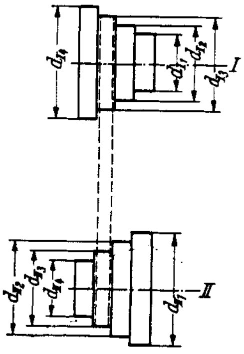


FIG. 122

$$u_1 = \frac{d_{I_1}}{d_{II_1}}; \quad u_2 = \frac{d_{I_2}}{d_{II_2}}; \quad u_3 = \frac{d_{I_3}}{d_{II_3}}; \quad u_4 = \frac{d_{I_4}}{d_{II_4}}$$

or generally

$$u_{n-1} = \frac{d_{I_{n-1}}}{d_{II_{n-1}}}; \quad u_n = \frac{d_{I_n}}{d_{II_n}}$$

If the diameters of the cone pulley on shaft I are stepped in accordance with a ratio φ_I and those of shaft II with a ratio φ_{II}, the following relations exist:

$$\frac{d_{I_n}}{d_{I_{n-1}}} = \varphi_I; \quad \frac{d_{II_{n-1}}}{d_{II_n}} = \varphi_{II}$$

Again in order to obtain a geometrically-stepped speed range of ratio φ

$$\frac{u_n}{u_{n-1}} = \frac{d_{I_n}}{d_{I_{n-1}}} \times \frac{d_{II_{n-1}}}{d_{II_n}} = \varphi$$

$$\varphi_I \times \varphi_{II} = \varphi$$

and if both cone pulleys are stepped by the same ratio ($\varphi_I = \varphi_{II}$)

$$\varphi_I^2 = \varphi_{II}^2 = \varphi; \quad \varphi_I = \varphi_{II} = \sqrt{\varphi}$$

If such cone pulleys are arranged at a centre distance (A) larger than $10(d_{\max} - d_{\min})$, it is possible to make the sum of the diameters of two corresponding cone pulleys constant if one endless belt (i.e. length of belt is constant) is used.

Here

$$d_{I_{n-1}} + d_{II_{n-1}} = d_{I_n} + d_{II_n} = C$$

and with

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

gives

$$d_{I_n} + \frac{d_{I_n}}{u_n} = d_{II_n} \times u_n + d_{II_n} = C$$

$$d_{I_n}\left(1 + \frac{1}{u_n}\right) = d_{II_n}(u_n + 1) = C$$

and finally

$$d_{I_n} = C \times \frac{u_n}{u_n + 1}; \quad d_{II_n} = C \times \frac{1}{u_n + 1}$$

For a transmission using a cone pulley and a jockey pulley the calculation of the diameters is almost identical with the calculation of the numbers of teeth for a Norton-type gearbox (Fig. 109). As all gears of the block on shaft I have to mesh with the intermediate gear (number of teeth n_t), which in turn drives the sliding gear on shaft II (number of teeth n_{tII}), they must have the same pitch. As, moreover, the diameter of a gear is

$$d = \frac{n_t}{p} \quad (p = \text{diametral pitch})$$

$$\frac{d_n}{d_{n-1}} = \frac{n_{t_n}/p}{n_{t_{n-1}}/p} = \frac{n_{t_n}}{n_{t_{n-1}}}$$

The numbers of teeth of all gears on the block on shaft *I*

$$(n_{tI_1}, n_{tI_2}, n_{tI_3}, n_{tI_4})$$

must, therefore, be geometrically stepped with a ratio corresponding to that of the speed range which is to be obtained on shaft *II*.

$$\frac{n_{tI_4}}{n_{tI_3}} = \frac{n_{tI_3}}{n_{tI_2}} = \frac{n_{tI_2}}{n_{tI_1}} = \varphi$$

In the cases of the gearboxes shown in Figs. 110, 111 and 112, the centre distance (A) of all gears, which are in mesh between shafts *I* and *II*, must be constant:

$$A = \frac{d_I + d_{II}}{2} = \frac{(n_{tI}/p) + (n_{tII}/p)}{2} = \frac{1}{2p}(n_{tI} + n_{tII})$$

If all gears in such a transmission have the same pitch, the sum of the number of teeth of all gears which can be put into mesh in one part drive must be constant:

$$n_{tI} + n_{tII} = 2pA = C$$

If the required transmission ratios are expressed in the form of fractions, the numbers of teeth of all gears in a part drive can then be calculated by the method of a common denominator. If two gears produce a ratio a/b, the sum of their numbers of teeth must be divisible by $a + b$.

For example, in the case of the gear transmission (Fig. 111), it is required that:

$$u_1 = \frac{n_{tI_1}}{n_{tII_1}} = \frac{1^*}{1{\cdot}6} = \frac{5}{8}$$

$$u_2 = \frac{n_{tI_2}}{n_{tII_2}} = \frac{1{\cdot}25^*}{1} = \frac{5}{4}$$

The sum of the numbers of teeth $n_{tI_1} + n_{tII_1} = n_{tI_2} + n_{tII_2}$ must be divisible by $5 + 8 = 13$ and by $5 + 4 = 9$. 13 is a relatively large prime number. The common denominator for 13 and 9 is $13 \times 9 = 117$, and the minimum possible sum of the numbers of teeth would be rather large. If one replaces the ratio $1/1{\cdot}6$ by the approximate value $7/11$ the common denominator of $7 + 11$ and $5 + 4$, and with it the minimum sum of the numbers of teeth is 18. The gear ratios are, therefore

$$\frac{n_{tI_1}}{n_{tII_1}} = \frac{7}{11}; \qquad \frac{n_{tI_2}}{n_{tII_2}} = \frac{10}{8}$$

If, for example, the smallest gear (7) should have no less than 21 teeth, all numbers have to be multiplied by 3, so that

$$n_{tI_1} = 21; \qquad n_{tI_2} = 30; \qquad n_{tII_1} = 33; \qquad n_{tII_2} = 24.$$

If it is not possible to arrange for all gears which are in mesh between two shafts to have the same pitch, the sum of the pitch circle diameters becomes

$$\frac{n_{tI_1}}{p_1} + \frac{n_{tII_1}}{p_1} = \frac{n_{tI_2}}{p_2} + \frac{n_{tII_2}}{p_2}$$

and the centre distance between the axes

$$A = \frac{1}{2p_1} \times (n_{tI_1} + n_{tII_1}) = \frac{1}{2p_2} \times (n_{tI_2} + n_{tII_2})$$

$$n_{tI_1} + n_{tII_1} = \frac{p_1}{p_2} \times (n_{tI_2} + n_{tII_2})$$

* Standard transmission ratios.

or

$$n_{tI_2} + n_{tII_2} = \frac{p_2}{p_1} \times (n_{tI_1} + n_{tII_1})$$

If, therefore, the sum of the numbers of teeth has been calculated on the assumption of equal pitch for all gears, it must be corrected for gears with pitch p_2 by multiplying with p_2/p_1 or in the case of gears with pitch p_1 by multiplying with p_1/p_2.

If very small transmission ratios are required in sliding-gear drives, it may be necessary for two gears which do not work together to pass without touching. This means that the difference between the numbers of teeth of two gears on the same shaft must be above a certain minimum. Such a case is shown in Fig. 123. The outside diameter of a gear is

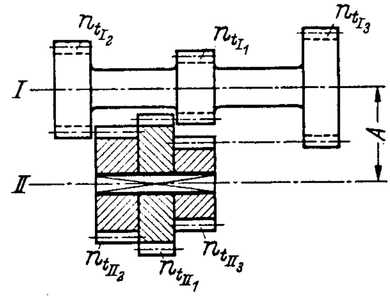


FIG. 123

$$d_{\max} = \frac{n_t + 2}{p}$$

The outside diameters of gears II_2 and I_1 must not touch each other when the gear block on shaft II is shifted. In the case of equal pitch p for all gears the following condition must be satisfied:

$$(n_{tI_1} + 2) + (n_{tII_2} + 2) = n_{tI_1} + n_{tII_2} + 4 \leqq 2pA$$

where

$$2pA = n_{tI_1} + n_{tII_1} = n_{tI_2} + n_{tII_2} = n_{tI_3} + n_{tII_3}$$

Therefore

$$n_{tI_1} + n_{tII_2} + 4 \leqq n_{tI_1} + n_{tII_1}$$
$$\leqq n_{tI_2} + n_{tII_2}$$

The minimum difference between the numbers of teeth of gears I_2 and I_1 or II_1 and II_2 must be:

$$n_{tI_2} - n_{tI_1} = n_{tII_1} - n_{tII_2} = 4$$

If, in addition, it is necessary that:

$$\frac{n_{tI_1}}{n_{tII_1}} = u_1 \quad \text{and} \quad \frac{n_{tI_2}}{n_{tII_2}} = u_2$$

and

$$n_{tI_2} = n_{tI_1} + 4 \quad \text{and} \quad n_{tII_2} = n_{tII_1} - 4$$

Then

$$u_2 = \frac{n_{tI_1} + 4}{n_{tII_1} - 4}$$

$$u_2 = \frac{u_1 . n_{tII_1} + 4}{n_{tII_1} - 4}$$

$$n_{tII_1} = \frac{4(u_2 + 1)}{u_2 - u_1}$$

In this way, it is possible to calculate the numbers of teeth for all gears:

$$n_{tI_1} = u_1 \times n_{tII_1}\,; \qquad n_{tI_2} = n_{tI_1} + 4\,; \qquad n_{tII_2} = n_{tII_1} - 4$$

The calculation of the numbers of teeth for gears in transmissions which are to produce standard speeds can be simplified considerably, because the transmission ratios in these gearboxes are also stepped in accordance with standard ratios (see page 77). If the values of the transmission ratios are always chosen equal to those of standard speed-range steps, it is possible to establish once and for all the pulley or pitch circle diameters or the numbers of gear teeth (which are proportional to the pitch circle diameters) which provide all practically applicable standard transmission ratios (for example, 1 : 2 to 4 : 1 or vice versa). This has been done by Germar and Stephan[31] (see Table 13).

If the numbers of teeth for draw-key gearboxes, clutch operated or sliding gear transmissions with gears of equal pitch have to be determined, i.e. in cases in which the sum of the numbers of teeth of all gears in mesh between two axes has to be constant, tables can be established in which the possible numbers of teeth of pinion and wheel for a given sum of numbers of teeth is laid down for all standard transmission ratios (Table 14).

The use of such tables may be shown with a simple example. It may be necessary to obtain the following transmission ratios between two shafts by means of three sets of gears (see Fig. 123):

$$u_1 = 1 : 1{\cdot}8\,; \qquad u_2 = 1 : 1{\cdot}4\,; \qquad u_3 = 1 : 1{\cdot}12$$

Table 14 shows that the sums of numbers of teeth between 100 and 109, with which the three transmission ratios can be obtained within permissible limits are 102, 106, 108 and 109. The corresponding gear ratios are:

$u_1 = \frac{n_{tI_1}}{n_{tII_1}}$	$u_2 = \frac{n_{tI_2}}{n_{tII_2}}$	$u_3 = \frac{n_{tI_3}}{n_{tII}}$
37 : 65	42 : 60	48 : 54
38 : 68	44 : 62	50 : 56
39 : 69	45 : 63	51 : 57
39 : 70	45 : 64	51 : 58

If, for reasons of space economy or manufacturing cost, the pinion is to be as small as possible (minimum number of teeth 19), the second row could be divided by 2 and would result in the following numbers of teeth:

$$\frac{n_{tI_1}}{n_{tII_1}} = \frac{19}{34}\,; \qquad \frac{n_{tI_2}}{n_{tII_2}} = \frac{22}{31}\,; \qquad \frac{n_{tI_3}}{n_{tII_3}} = \frac{25}{28}$$

(sum of numbers of teeth = 53). In this case, it would be necessary, however, to bear in mind that such gears would be suitable for a draw-key gearbox (Fig. 110), but not for a sliding-gear transmission (Fig. 123). This is because the difference between the numbers of teeth of two adjacent gears

$$(n_{tI_2} - n_{tI_1})$$

or

$$(n_{tII_1} - n_{tII_2})$$

would be only 3 (i.e. less than the 4 required, see page 86), so that gear II_2 (31 teeth) would not pass gear I_1 (19 teeth) without touching it.

As a final example the calculations for the two gearboxes discussed on page 80 and designed for a milling spindle drive with 12 or 18 spindle speeds can now be given. The specification which has to be met when determining the maximum and minimum spindle speeds is:

Maximum cutter diameter d_{max} = 12 in.

Minimum cutter diameter d_{min} = 0·8 in.

Minimum cutting speed of largest milling cutter = v_{min} = 100 ft/min.

Maximum cutting speed of smallest milling cutter = v_{max} = 300 ft/min.

The maximum and minimum spindle speeds are then given by:

$$n_{max} = \frac{12 v_{max}}{\pi d_{min}} = \frac{12 \times 300}{\pi \times 0{\cdot}8} \approx 1400 \text{ rev/min}$$

$$n_{min} = \frac{12 v_{min}}{\pi d_{max}} = \frac{12 \times 100}{\pi \times 12} \simeq 32 \text{ rev/min}$$

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

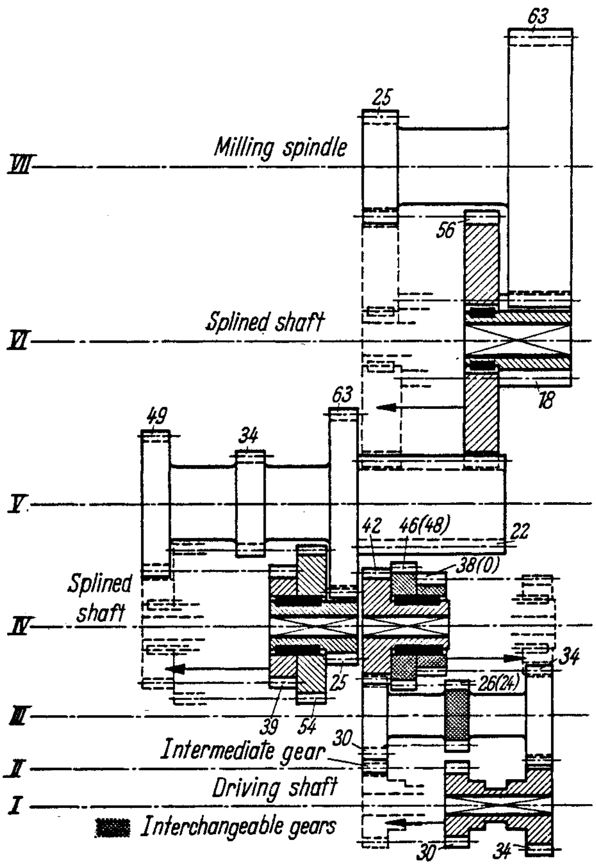


FIG. 124. Milling spindle drive for 12 or 18 spindle speeds.

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 speed diagram for the 18-speed gearbox is shown in Fig. 125 and that for the 12-speed gearbox in Fig. 126.

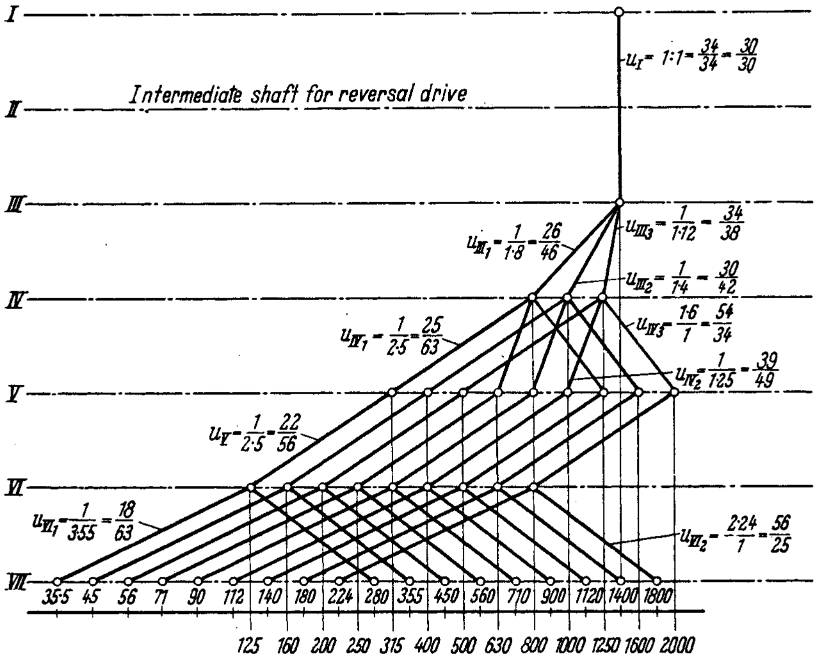


FIG. 125. Speed diagram of the 18-step gear drive (see Fig. 124).

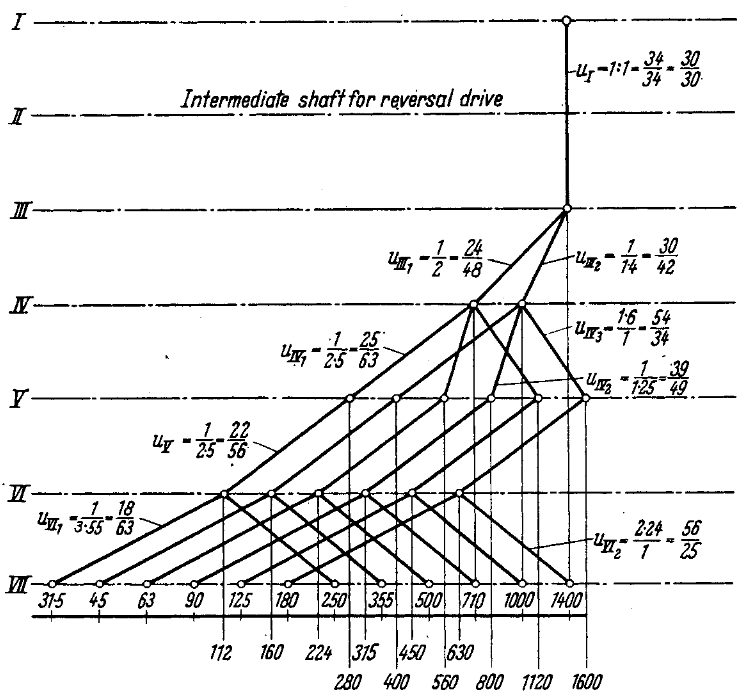


FIG. 126. Speed diagram of the 12-step gear drive (Fig. 124).

Consider first the 18-speed gearbox. In the first part drive (shafts *III* to *IV*) the maximum output speed lies lower by half a step ($\sqrt{\varphi} = \sqrt{1{\cdot}25} = 1{\cdot}12$) than the input speed, so that

$$u_{III_3} = \frac{1}{1{\cdot}12} = \frac{17}{19}$$

$$u_{III_2} = \frac{1}{1{\cdot}12 \times 1{\cdot}25} = \frac{1}{1{\cdot}4} \simeq \frac{5}{7}$$

$$u_{III_1} = \frac{1}{1{\cdot}4 \times 1{\cdot}25} = \frac{1}{1{\cdot}8} \simeq \frac{5}{9} \approx \frac{13}{23}$$

The sum of the numbers of teeth determined by the method of the common denominator is

$$u_{III_3}: \quad 17 + 19 = 36$$

$$u_{III_2}: \quad 5 + 7 \quad = 12$$

$$u_{III_1}: \quad 13 + 23 = 36$$

$$\text{Common denominator} = 36$$

In order to ensure that the minimum number of teeth is 18, the sum of the numbers of teeth must be at least $2 \times 36 = 72$.
Therefore

$$u_{III_3} = 34 : 38$$

$$u_{III_2} = 30 : 42$$

$$u_{III_1} = 26 : 46$$

Consider next the 12-speed gearbox. In the first part drive (shafts *III* to *IV*) (Fig. 126), the input speed is reduced by φ (1·4) and φ^2 (2), so that

$$u_{III_1} = \frac{1}{2}$$

$$u_{III_2} = \frac{1}{1{\cdot}4} = \frac{5}{7}$$

and the numbers of teeth (method of common denominator);

$$u_{III_1}: \quad 1 + 2 = 3$$

$$u_{III_2}: \quad 5 + 7 = 12$$

If the minimum number of teeth is to be 18, the smallest sum of the numbers of teeth must be 60, so that

$$u_{III_1} = 20 : 40 \quad (20 + 40 = 60)$$

$$u_{III_2} = 25 : 35 \quad (25 + 35 = 60)$$

However, in order to make this part drive interchangeable with that for the 18-speed gearbox, (i.e. for an identical distance between axes *III* and *IV*) the sum of the numbers of teeth must be the same in both cases (72), so that

$$u_{III_1} = 24 : 48$$

$$u_{III_2} = 30 : 42$$

The second part drive covers one gear ratio which is larger and two which are smaller than unity.

$$u_{IV_3} = \frac{1{\cdot}25^2}{1} = \frac{1{\cdot}6}{1}; \qquad u_{IV_2} = \frac{1}{1{\cdot}25}; \qquad u_{IV_1} = \frac{1}{1{\cdot}25^4} = \frac{1}{2{\cdot}5}.$$

If the sums of the numbers of teeth are calculated by means of the common denominator method the transmission ratios of the various pairs of gears in mesh are represented by whole numbers, so that periodic transmission errors may result in undesirable vibrations, whilst the numbers of teeth given in the previously mentioned tables (see Tables 13 and 14) represent gear ratios which are not always accurate but still within the permissible tolerances. The ratios between the numbers of teeth of gear pairs in mesh are then not always whole numbers. The numbers of teeth of the second (shafts *IV* to *V*) and third (shafts *VI* to *VII*) part drives have been found by means of the Germar tables.

When the speed reduction ratio between shafts *V* and *VI* is determined, it is necessary to consider that the large driving wheel on shaft *VI* is at the same time the driven wheel of the transmission between shafts *V* and *VI*. The number of teeth (x) of the driven gear for the ratio $u_V = 1 : 2{\cdot}5$, must satisfy the equation:

$$\frac{x}{56} = \frac{1}{2{\cdot}5}$$

$$x = \frac{56}{2{\cdot}5}$$

which is approximately 22.

Finally, it is necessary to check whether the sliding gears on shaft *IV* of the first part drive and the long gear on shaft *V* are clear of each other.

TABLE 15

Gear Drive	Input Speed	Total Transmission Ratio	Output Speed	Standard Speed	Exact Value	Deviation of the Output Speed from the Exact Value
	rev/min.	$u_{III} \cdot u_{IV} \cdot u_V \cdot u_{VI}$	rev/min.	rev/min.	rev/min.	%
		$\frac{26}{46} \cdot \frac{25}{63} \cdot \frac{22}{56} \cdot \frac{18}{63}$	35·2	35·5	35·481	−0·8
		$\frac{30}{42} \cdot \frac{25}{63} \cdot \frac{22}{56} \cdot \frac{18}{63}$	44·6	45	44·668	−0·15
		$\frac{34}{38} \cdot \frac{25}{63} \cdot \frac{22}{56} \cdot \frac{18}{63}$	55·8	56	56·234	−0·8
		$\frac{26}{46} \cdot \frac{39}{49} \cdot \frac{22}{56} \cdot \frac{18}{63}$	71	71	70·795	+0·3
		$\frac{30}{42} \cdot \frac{39}{49} \cdot \frac{22}{56} \cdot \frac{18}{63}$	89·5	90	89·125	+0·42
		$\frac{34}{38} \cdot \frac{39}{49} \cdot \frac{22}{56} \cdot \frac{18}{63}$	112	112	112·20	−0·18
		$\frac{26}{46} \cdot \frac{54}{34} \cdot \frac{22}{56} \cdot \frac{18}{63}$	141	140	141·25	−0·18
18 Steps	1400	$\frac{30}{42} \cdot \frac{54}{34} \cdot \frac{22}{56} \cdot \frac{18}{63}$	178·5	180	177·83	+0·38
		$\frac{34}{38} \cdot \frac{54}{34} \cdot \frac{22}{56} \cdot \frac{18}{63}$	223	224	223·87	−0·4
		$\frac{26}{46} \cdot \frac{25}{63} \cdot \frac{22}{56} \cdot \frac{56}{25}$	276	280	281·84	−2·05*
		$\frac{30}{42} \cdot \frac{25}{63} \cdot \frac{22}{56} \cdot \frac{56}{25}$	350	355	354·81	−1·35
		$\frac{34}{38} \cdot \frac{26}{63} \cdot \frac{22}{56} \cdot \frac{56}{25}$	438	450	446·68	−1·95
		$\frac{26}{46} \cdot \frac{39}{49} \cdot \frac{22}{56} \cdot \frac{56}{25}$	556	560	562·34	−1·15
		$\frac{30}{42} \cdot \frac{39}{49} \cdot \frac{22}{56} \cdot \frac{56}{25}$	702	710	707·95	−0·85
		$\frac{34}{38} \cdot \frac{39}{49} \cdot \frac{22}{56} \cdot \frac{56}{25}$	880	900	891·25	−1·26

TABLE 15 (*continued*)

Gear Drive	Input Speed	Total Transmission Ratio	Output Speed	Standard Speed	Exact Value	Deviation of the Output Speed from the Exact Value
	rev/min	$u_{III} \cdot u_{IV} \cdot u_V \cdot u_{VI}$	rev/min	rev/min	rev/min	%
18 Steps	1400	$\frac{26}{46} \cdot \frac{54}{34} \cdot \frac{22}{56} \cdot \frac{56}{25}$	1105	1120	1122	−1·5
		$\frac{30}{42} \cdot \frac{54}{34} \cdot \frac{22}{56} \cdot \frac{56}{25}$	1400	1400	1412·5	−0·9
		$\frac{34}{38} \cdot \frac{54}{34} \cdot \frac{22}{56} \cdot \frac{56}{25}$	1750	1800	1778·3	−1·6
12 Steps	1400	$\frac{24}{48} \cdot \frac{25}{63} \cdot \frac{22}{56} \cdot \frac{18}{63}$	31·2	31·5	31·623	−0·14
		$\frac{30}{42} \cdot \frac{25}{63} \cdot \frac{22}{56} \cdot \frac{18}{63}$	44·6	45	44·668	−0·15
		$\frac{24}{48} \cdot \frac{39}{49} \cdot \frac{22}{56} \cdot \frac{18}{63}$	62·4	63	63·096	−0·11
		$\frac{30}{42} \cdot \frac{39}{49} \cdot \frac{22}{56} \cdot \frac{18}{63}$	89·5	90	89·125	+0·42
		$\frac{24}{48} \cdot \frac{54}{34} \cdot \frac{22}{56} \cdot \frac{18}{63}$	124·8	125	125·89	−0·87
		$\frac{30}{42} \cdot \frac{54}{34} \cdot \frac{22}{56} \cdot \frac{18}{63}$	178·5	180	177·83	+0·38
		$\frac{24}{48} \cdot \frac{25}{63} \cdot \frac{22}{56} \cdot \frac{56}{25}$	245	250	251·19	−2·4*
		$\frac{30}{42} \cdot \frac{25}{63} \cdot \frac{22}{56} \cdot \frac{56}{25}$	350	355	354·81	−1·35
		$\frac{24}{48} \cdot \frac{39}{49} \cdot \frac{22}{56} \cdot \frac{56}{25}$	491	500	501·19	−2·02*
		$\frac{30}{42} \cdot \frac{39}{49} \cdot \frac{22}{56} \cdot \frac{56}{25}$	702	710	707·95	−0·85
		$\frac{24}{48} \cdot \frac{54}{34} \cdot \frac{22}{56} \cdot \frac{56}{25}$	980	1000	1000	−2
		$\frac{30}{42} \cdot \frac{54}{34} \cdot \frac{22}{56} \cdot \frac{56}{25}$	1400	1400	1412·5	−0·9

* Slightly in excess of the tolerance of ±2%.

In view of the very high torques which the long gear may have to transmit when shaft V runs at low speed (gear ratio $u_{IV_1} = \frac{25}{63}$), it is assumed that the teeth of the long gear are designed with 6 DP, whilst the gears of the first and second part drives are designed with 9 DP. This means that the centre distance (A) between shafts IV and V (second part drive, sum of the number of teeth 88) becomes

$$A = \frac{88}{2 \times 9} = 4{\cdot}889 \text{ in.}$$

The outside diameter of the long gear (22 teeth) is

$$d_{22} = \frac{22 + 2}{6} = 4 \text{ in.}$$

and the outside diameter of the largest driven gear (48 teeth) in the first part drive of the 12-speed gearbox is

$$d_{48} = \frac{48 + 2}{9} = 5{\cdot}556 \text{ in.}$$

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

$$\frac{d_{22} + d_{48}}{2} = \frac{9{\cdot}556}{2} = 4{\cdot}778 \text{ in.}$$

and is, therefore, less than the existing centre distance of 4·889 in.

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 Jaekel[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		High Frequency	Motor Driving the Machine Tool		
Driving Motor	Generator		Number of Poles	Synchronous Speed	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.

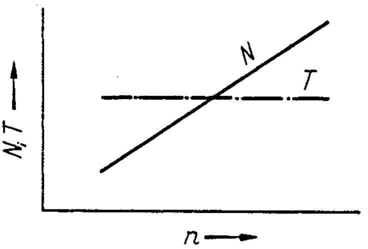


FIG. 127. Speed adjustment of the d.c. shunt motor by changing the rotor voltage.

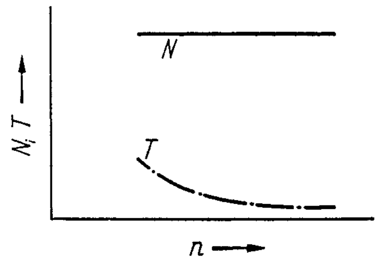


FIG. 128. Speed adjustment of the d.c. shunt motor by changing the field current.

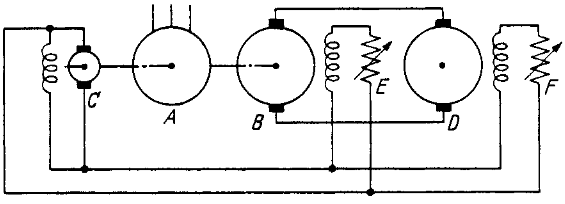


FIG. 129. Circuit of a Ward–Leonard set.

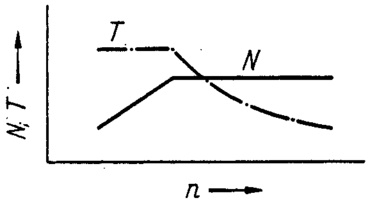


FIG. 130. Speed adjustment of the Ward-Leonard set.

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 motor for each operational movement. Each such motor can be controlled easily by means of levers or push buttons from one or several centralized positions.

A simple example is the rapid traverse driven by a separate motor over a free wheel coupling (Fig. 131a). The rapid traverse movement is here produced by a push button controlled motor *C*, overtaking (free wheel *D*) the ordinary feed drive *A–B–D*. The rapid traverse is disengaged and the ordinary feed movement continued as soon as the rapid traverse motor *C* stops. In view of the unidirectional action of the free wheel *D*, it is, of course, important that the driving shafts for *A*, *B* and *D* run in one direction only, and that the reversing clutch *E* is arranged behind the free wheel. The difficulty is avoided in the design (Fig. 131b)* in which an epicyclic gear is used. The

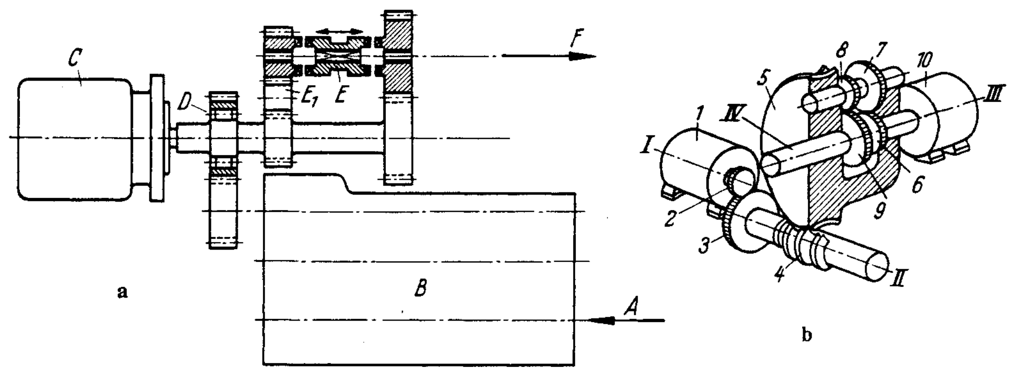


FIG. 131. Rapid traverse drives:
a) Free wheel drive *A*) feed drive; *B*) feed change gear box; *C*) rapid traverse motor; *D*) free wheel device; *E*) reversing gear (E_1—intermediate gear); *F*) drive of feed screw.
b) Epicyclic gear *1*) feed drive motor; *2, 3*) slip gears; *4*) worm; *5*) worm wheel; *6, 7, 8, 9*) planetary gears; *10*) rapid traverse motor; *I*) feed drive motor shaft; *II*) worm shaft; *III*) rapid traverse motor shaft; *IV*) driving shaft for feed screw.

feed motor *1* drives via shaft *I* slip gears *2*, *3* worm *4* and worm wheel *5*. The latter serves at the same time as the arm for the epicyclic gears *6*, *7*, *8* and *9*. The pinion *6* is fixed to shaft *III* of the rapid traverse motor *10* and wheel *9* to shaft *IV* which serves for operating the feed driving elements (screw and nut, pinion and rack, etc.). If the rapid traverse motor is stopped (usually held stationary by means of a brake), the speed n_{IV} of shaft *IV* is:

$$n_{IV} = n_5\left(1 - \frac{n_{t_6}}{n_{t_7}} \cdot \frac{n_{t_8}}{n_{t_9}}\right)$$

n_5 is the speed of the worm wheel which can be set by means of slip gears *2* and *3* and n_{t_6}, n_{t_7}, n_{t_8} and n_{t_9} are the numbers of teeth of gears *6*, *7*, *8* and *9*. When the rapid traverse motor *10* is started, the speed of shaft *III* being n_{III}, the speed of shaft *IV* is approximately:

$$n_{IV(\text{rapid traverse})} \approx n_{III} \cdot \frac{n_{t_6}}{n_{t_7}} \cdot \frac{n_{t_8}}{n_{t_9}}$$

$$\left(n_5 \text{ is small in relation to } n_{III} \cdot \frac{n_{t_6}}{n_{t_7}} \cdot \frac{n_{t_8}}{n_{t_9}}\right).$$

The advantage of the free wheel arrangement (Fig. 131a), which enables the feed drive to be left engaged even while the rapid traverse drive is in action, has been maintained. In addition, the device (Fig. 131b) can also be reversed easily from any direction of either feed or rapid traverse drive to any other direction of either drive if required.

The arrangements (Figs. 131a and b) are very simple because there is no need for interlocking

* Gebrüder Honsberg, Remscheid-Hasten, Germany.

feed and quick traverse motors. Whatever the required working conditions, these motors can be started or stopped independently of each other.

If, however, several motors are applied for driving different and independent movements which must be controlled in a specific interrelated manner, this must be achieved by mechanical or electrical interlocking devices.

A simple mechanical device, which makes faulty operations impossible, consists of all control movements being operated by a single lever. For the control of a milling machine, such a lever could, for instance, operate the following movements in several positions (Fig. 132).

A — rapid traverse of the table from right to left (milling spindle stopped)
B — table feed right to left (milling spindle rotating)
C — milling spindle started
D — zero position (everything switched off)
E — milling spindle started
F — table feed from left to right (milling spindle rotating)
G — rapid traverse of the table from left to right (milling spindle stopped).

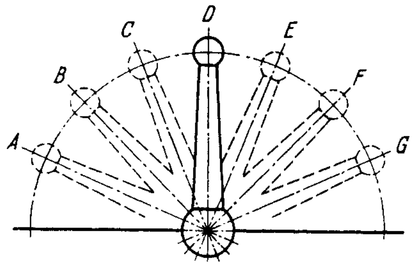


FIG. 132

With this arrangement, it is possible to control the operational movements in a foolproof manner. If one starts from the centre position of the lever (zero position), the table feed in either direction can only be started when the spindle is already running and the rapid traverse cannot be put into operation until the table has already been moved by the feed drive. Some disadvantages of such an arrangement are:

(1) During setting up of the machine, the operator must always start the spindle even if he wishes to move the table quickly from one position to another.

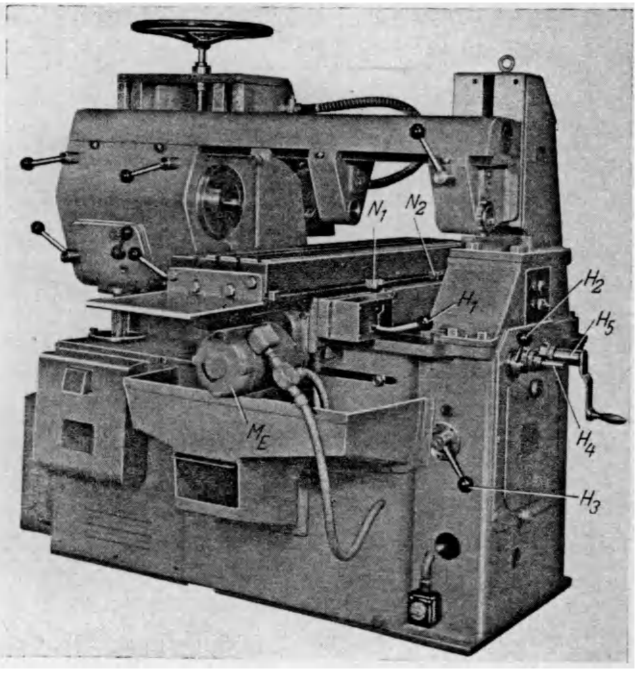


FIG. 133. Production milling machine (welded construction) (Cooke & Ferguson Ltd., Manchester). H_1 = Control lever (*1*, Fig. 139); H_2 = Selector lever for longitudinal or transverse table traverse (on shaft *12*, see Fig. 139); H_3 = Lever for feed drive back gear (*c*, Fig. 137); H_4 = Crank handle shaft for manual longitudinal traverse; H_5 = Cross traverse screw (*g*, Fig. 137); M_E = Rapid traverse motor (*n*, Fig. 137); N_1, N_2 = Adjustable trip dogs for disengaging the longitudinal power feed (via bar *10*, Fig. 139).

(2) If, for instance, the table is to be taken back to its original position, for example, by a rapid traverse movement from left to right after a feed movement from right to left (lever position *B*), the operator has to go through all other lever positions *C*, *D*, *E*, *F* and *G*.

Application to a Multi-purpose Milling Machine

The milling machine (Figs. 133 and 134) is fitted with electro-mechanical equipment in which the operational movements are controlled in the following manner:[37]

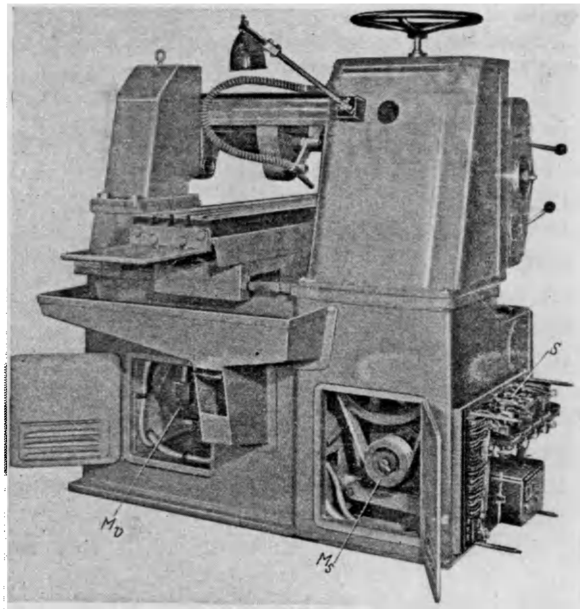


FIG. 134. Rear view of the milling machine (Fig. 133).
M_s = Spindle drive motor; M_v = Feed drive motor (*a*, Fig. 137); S = Control panel (Fig. 53).

(1) Starting (forward or reverse) and stopping the milling spindle by means of manually operated push buttons (Fig. 135).

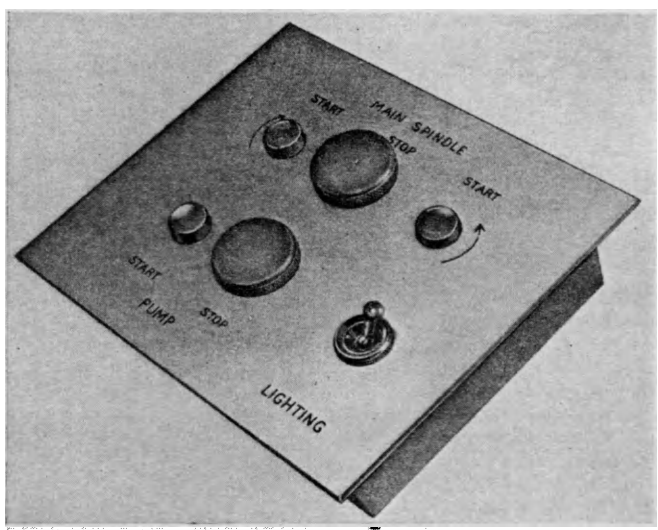


FIG. 135. Push button panel of the milling machine (Fig. 133).

(2) Controlling the various table movements by a single lever guided in a gate (Fig. 136):
A — rapid traverse from right to left
B — rapid traverse from left to right
C — power feed from right to left
D — neutral
E — power feed from left to right.

(3) Electrical interlocking, in which the feed motor can only be started when the spindle motor is already running, in order to prevent the workpiece being fed against a stationary milling cutter, and possible damage to the milling arbor.

(4) Electrical interlocking, through which the spindle motor is stopped, and possibly braked, as soon as the rapid traverse is started (see page 97), so that the milling cutter does not mark the surface, for instance, during a rapid traverse return of the table.

(5) Mechanical interlocking, to allow the feed motor to be started only when the corresponding gears and clutches are engaged and to stop the feed motor as soon as the mechanical power transmission is either interrupted or switched from longitudinal to cross traverse.

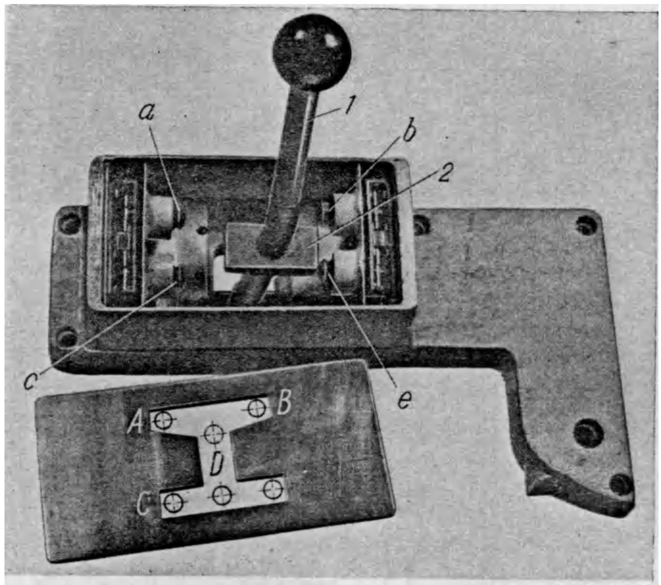


FIG. 136. Feed drive control box of the milling machine (Fig. 133) with cover (gate) removed.

The machine is equipped with four motors. Apart from the motors for the spindle and the coolant pump, a separate motor is provided for the table feed and a flange motor for the rapid traverse of the table (Fig. 137). The circuit diagram (Fig. 138) shows how the various motors are electrically interlocked. The spindle and pump motors (the spindle motor is reversible) can be started and stopped by ordinary push button controlled switches (see Fig. 135). The switches for the feed and rapid traverse motors have no holding contacts, so that these motors run only as long as the corresponding push buttons (*a*, *b*, *c* and *e*) in the control box (Fig. 139) are held down by lever *1*. The control box contains three sets of mechanisms for the following purposes.

(i) Operation of the push buttons for controlling table feed and rapid traverse

(ii) Disengagement of the feed clutch before the rapid traverse motor is started

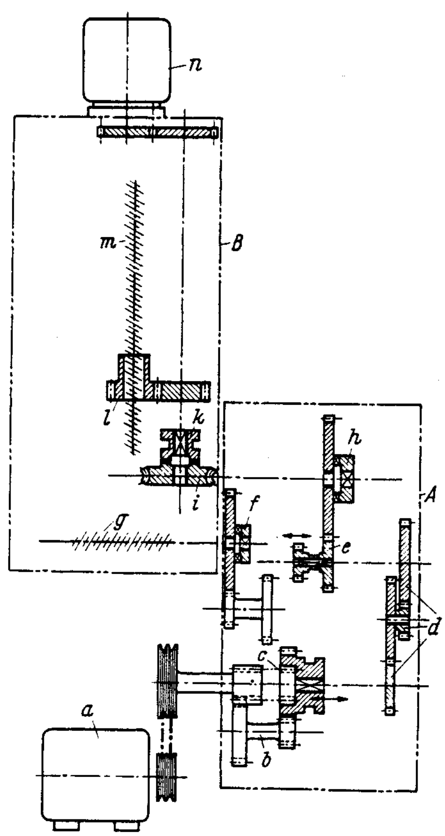


FIG. 137. Feed drive of the milling machine (FIG. 133). *A*—Feed drive gearbox; *B*—Cross slide (saddle); *a*—Feed drive motor; *b*—Back gear; *c*—Internal gear on the sliding gear block; *d*—Slip gears; *e*—Selector gear block for changing from longitudinal to cross traverse; *f*—Overload clutch for the cross traverse drive; *g*—Cross traverse screw; *h*—Overload clutch for the longitudinal traverse drive; *i*—Worm and worm wheel drive for longitudinal traverse; *k*—Clutch for longitudinal traverse drive; *l*—Rotating nut for longitudinal traverse drive; *m*—Fixed longitudinal feed screw; *n*—Rapid traverse motor for longitudinal traverse.

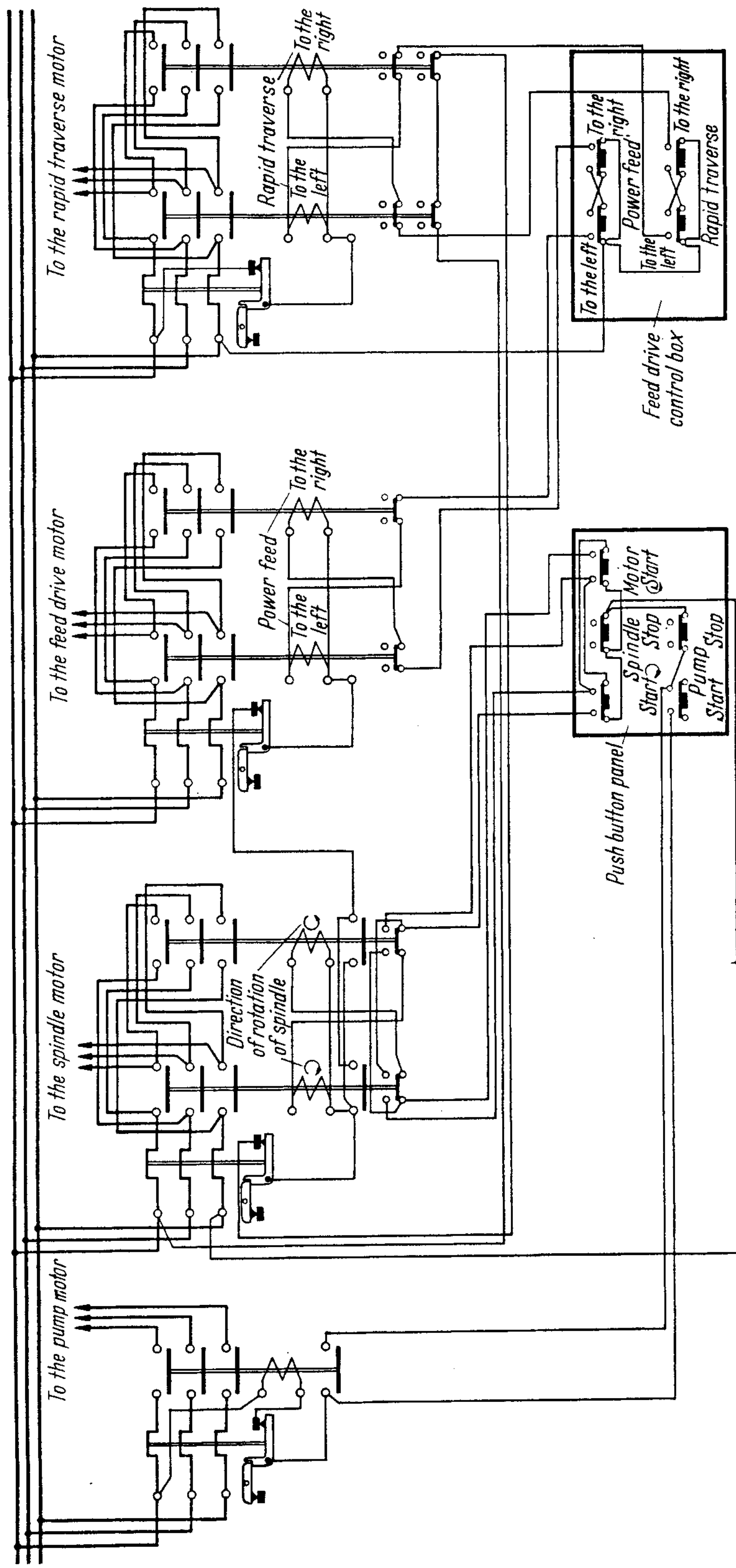


FIG. 138. Circuit diagram for the milling machine (Fig. 133).

(iii) Locking of the control lever so that the feed motor cannot be started before the mechanical driving elements are engaged.

The detailed actions of these mechanisms are as follows:

(i) In accordance with its position in the control box, lever *1* pushes by means of cross piece *2* one of the push buttons (a, b, c, e) respectively. For this purpose, lever *1* can swing around two axes, the horizontal axis *3* and the vertical axis *4*.

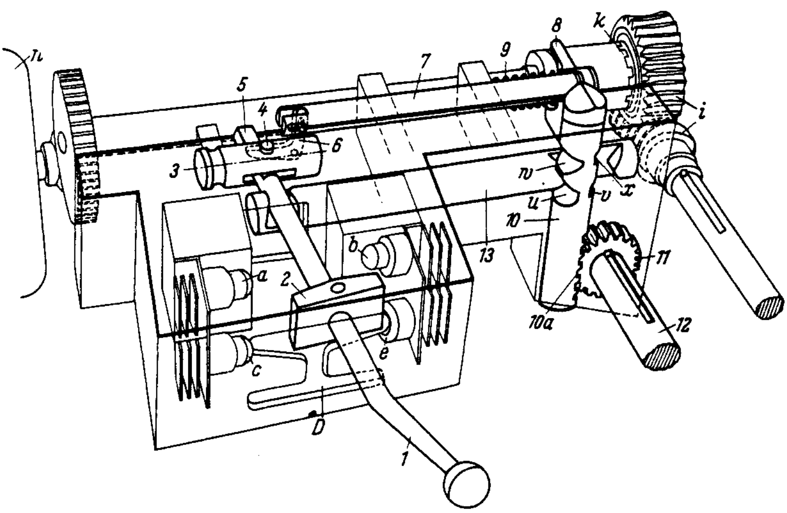


FIG. 139. Single lever control for the power feed and rapid traverse of the milling machine (Fig. 133).

(ii) The worm and worm wheel drive i (Fig. 137) is irreversible. Moreover, the feed movement is reversed by reversing the feed motor a (Fig. 137), so that a free wheel device (see Fig. 131a) cannot be used. Clutch k (Fig. 137) must, therefore be disengaged before the rapid traverse motor n is started. When lever *1* is moved upwards in its centre position, (D, Fig. 136), in order to start the rapid traverse, the tongue *5* (Fig. 139) presses the bell crank lever *6* downwards, so that clutch k is disengaged by means of push rod *7* and fork *8* against the action of spring *9*. As lever *1* is then moved to the left (A, Fig. 136), or to the right (B, Fig. 136), the appropriate push buttons are pressed and the rapid traverse movement to the left or right is initiated. This movement continues as long as lever *1* is held by hand in this position. When the operator releases it, lever *1* drops to the bottom centre position of the gate, spring *9* brings clutch k into engagement and the longitudinal table traverse can be started as soon as the feed motor a (Fig. 137) is started (position C or E of lever *1*, Fig. 136).

(*iii*) Push rod *10* is connected via rack *10a*, pinion *11* and shaft *12* with the control mechanism for the sliding gear block e (Fig. 137), in such a manner that in the top position of rod *10*, the longitudinal feed and in the bottom position the transverse feed is engaged. Lever *1* (Fig. 139) can be moved out of its central bottom position (D) to the left (C) or to the right (E) only when the notches (u, v = top position for longitudinal feed, w, x = bottom position for the transverse movement) in rod *10* allow axial displacement of the interlocking bar *13*. On the other hand, interlocking bar *13* and with it lever *1* are pushed into the central position (and the feed motor is thus stopped because the pressure on the push buttons is relieved) as soon as the adjustable trip dogs N_1 and N_2 (Fig. 133) on the table or on the saddle, push the rod *10* either downwards (during longitudinal traverse) or upwards (during cross traverse) into its middle position.

Longitudinal or cross traverse is selected by a lever (H_2, see Fig. 133), which is keyed to shaft *12*. In order to start the rapid traverse, lever *1* has to be lifted out of the notch in the interlocking bar *13*. As a result, a longitudinal rapid traverse movement can be started independently of the direction of the previously selected feed movement.

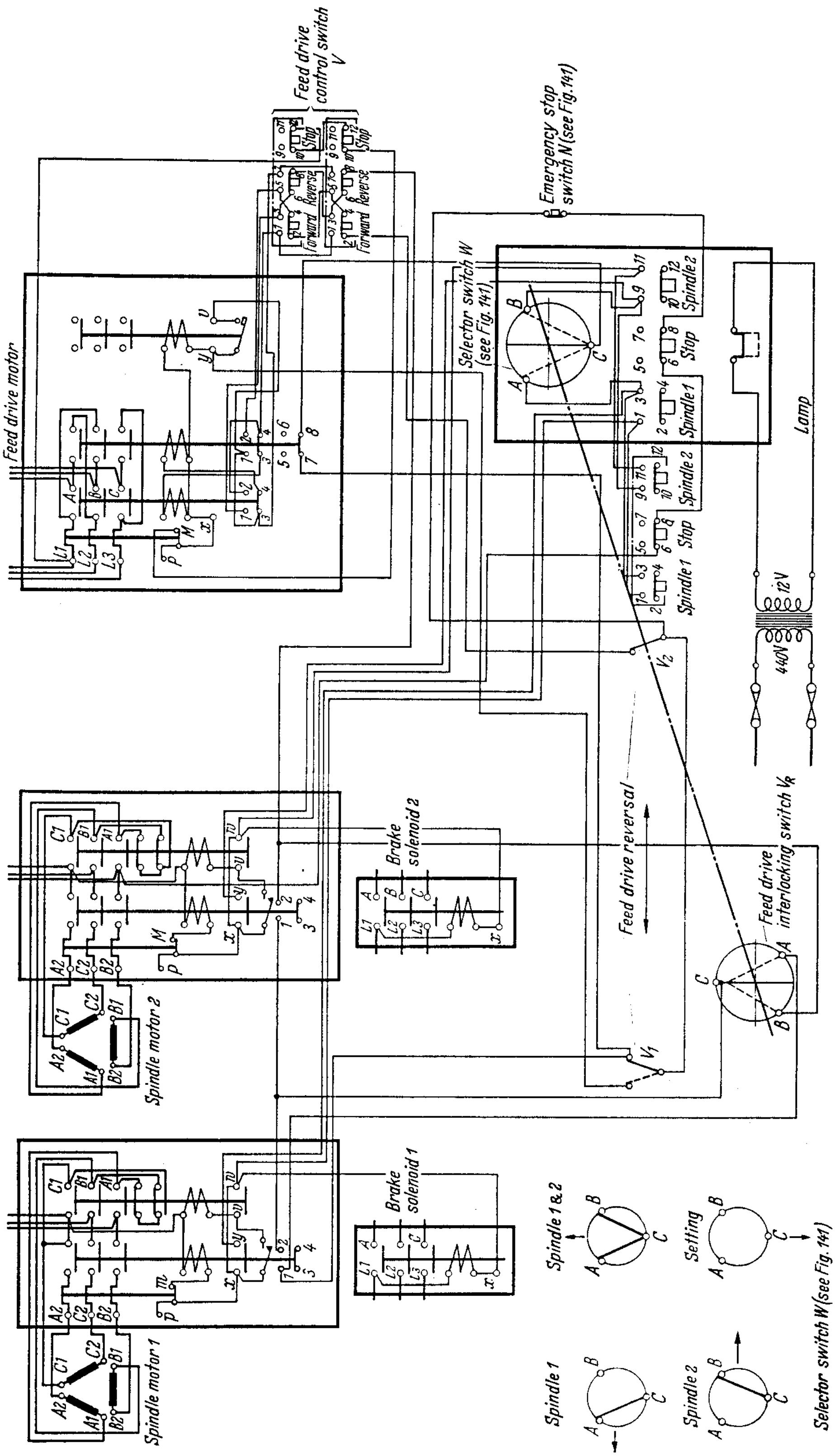


FIG. 140. Control circuit for the semi-automatic control of a two-spindle milling machine (See Figs. 141 and 142).

Application to a Single-purpose Milling Machine

The semi-automatic control circuit (Fig. 140), of the two-spindle milling machine (Figs. 141 and

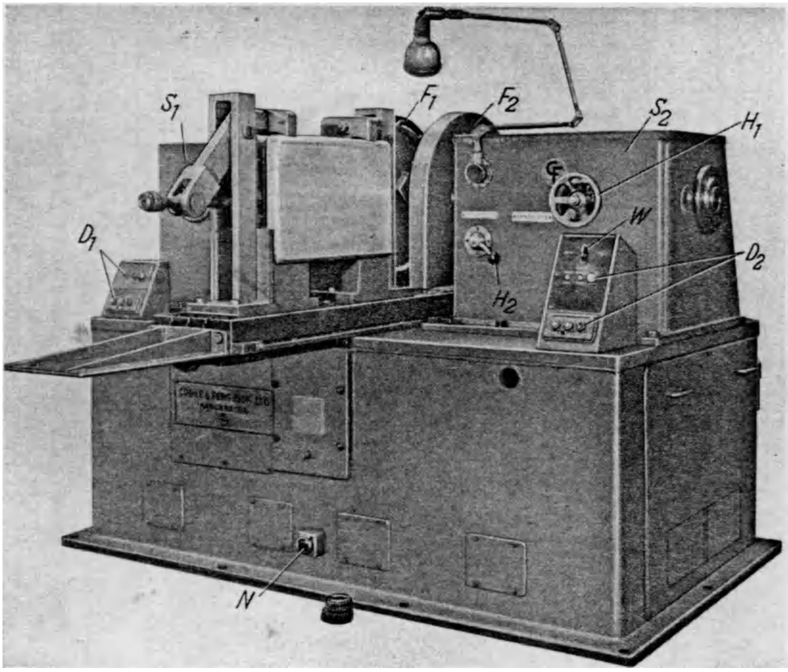


FIG. 141. Two-spindle single-purpose milling machine (Cooke & Ferguson Ltd., Manchester). D_1,D_2 = Push button controls for spindle and feed drives (see Fig. 140); F_1,F_2 = Cutter heads; H_1 = Axial spindle adjustment (spindle *2*); H_2 = Clamping of spindle *2* in axial position; N = Emergency stop switch (see Fig. 140); S_1,S_2 = Headstocks; W = Selector switch (see Fig. 140).

142), with two-spindle motors and one feed motor enables the following operations to be carried out:

(1) Starting (forward or reverse) and stopping of one or both milling spindles by means of push-button control.
(2) Starting (forward or reverse) and stopping the table movement by means of push-button control.
(3) Automatic table reversing at the end of a cutting traverse.
(4) Interlocking as follows:
 (a) The table feed for a cutting traverse can be started only when one or both spindles (selector switch W) are running.
 (b) At the end of a cutting traverse, the table feed is reversed by means of tumbler switch V_1 (Fig. 140) and the spindle motors, together with the brake relief solenoids B_1 and B_2 are switched off, so that the spindle drives are instantly stopped before the return traverse of the table is started. At the end of the return traverse of the table, the feed drive is stopped by means of switch V_2 the machine thus being completely stopped.

The selector switch W (Figs. 140 and 141) serves for operating the machine either with both spindles or with only one. In addition, it is possible to make inoperative the interlocking between the feed motor and the spindle rotation. For the purpose of setting the machine, the operator can then run the table freely either to the right or the left without being in any danger from one of the high-speed spindles rotating at 1700 rev/min, with 24 in. dia. cutter heads.

A selector switch W is provided only on the right-hand side of the machine. Push buttons for all other controls are arranged in front of each headstock (D_1 and D_2, Fig. 141). Once the selector

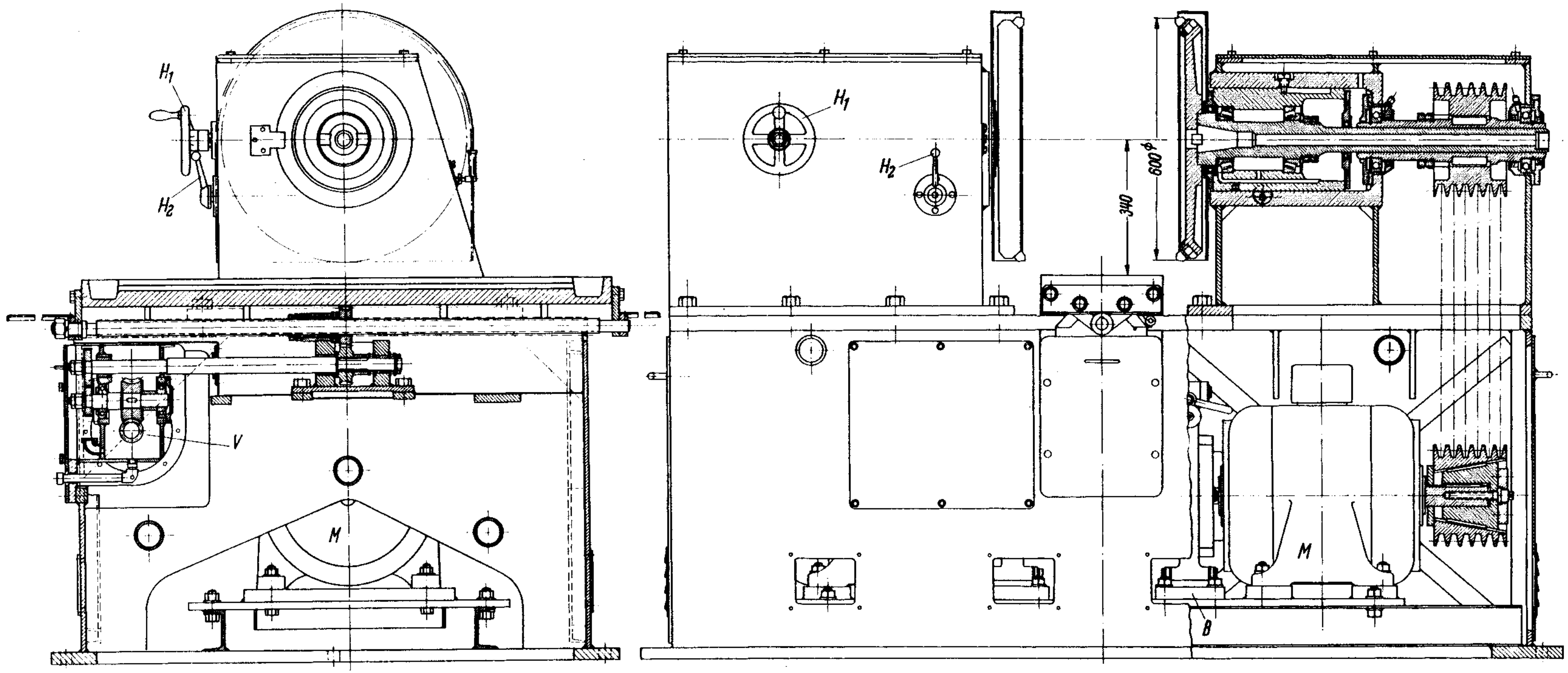


FIG. 142. Two-spindle milling machine (see Fig. 141). *B*) Brake with relief solenoids. Slip gear quadrant; H_1 and H_2, see Fig. 141; *M*) Spindle drive motor (18 kW, 900 rev/min); *V*) feed drive gearbox driven by a flange motor (1·5 kW, 1400 rev/min).

switch is set, therefore, the operator can control the machine either from headstock *1* or headstock *2*.

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 page 97).

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* an *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.

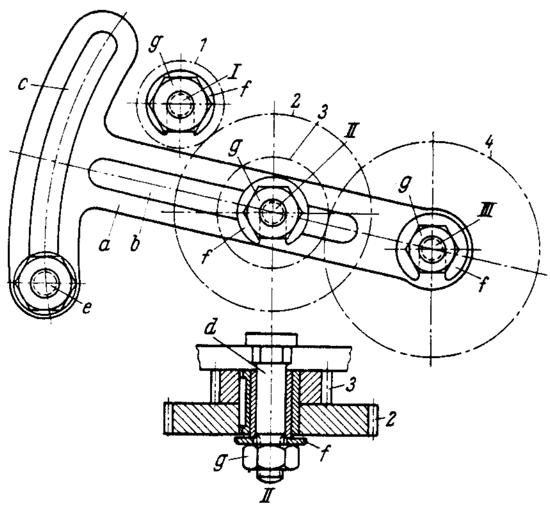


FIG. 143. Slip gear quadrant.

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

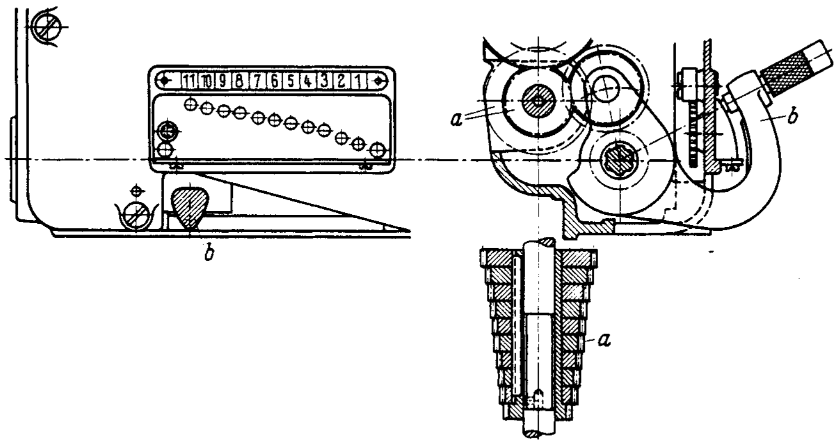


FIG. 144. Norton-type gearbox.

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*.

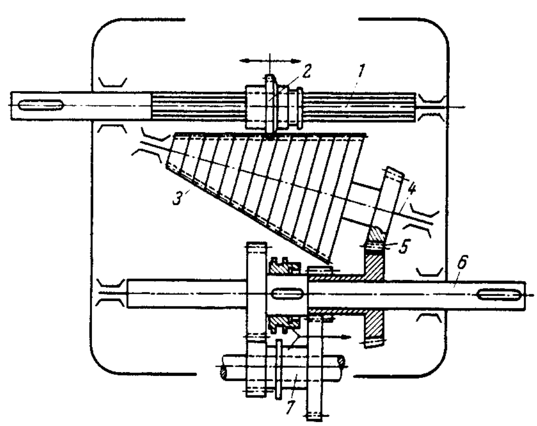


FIG. 145. Sliding gearbox (from *Stanki i Instrument*, December 1958).

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.

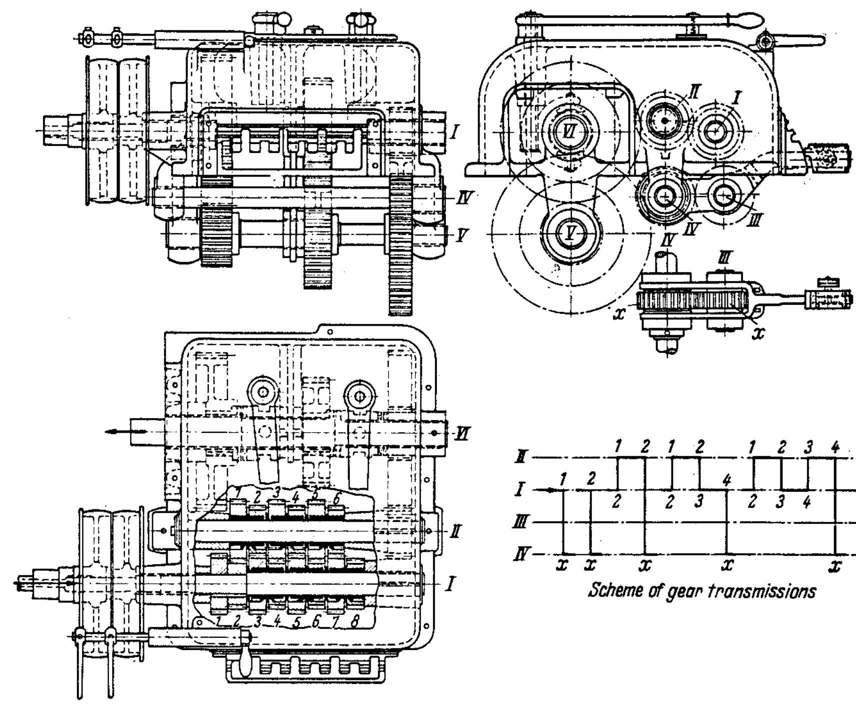


FIG. 146. "Meander" gearbox.

(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:

$$n_{tI_2} + n_{tII_1} = n_{tI_3} + n_{tII_2} = n_{tI_4} + n_{tII_3} = \ldots = \text{const.}$$

Furthermore

$$n_{tI_2} = n_{tI_4} = n_{tI_6} \text{ etc.}$$

and

$$n_{tI_1} = n_{tI_3} = n_{It_5} \text{ etc.}$$

Hence

$$n_{tI_3} = n_{tII_3} = n_{tI_5} = n_{tII_5} \text{ etc.}$$

and

$$n_{tI_2} = n_{tII_2} = n_{tI_4} = n_{tII_4} \text{ etc.}$$

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,

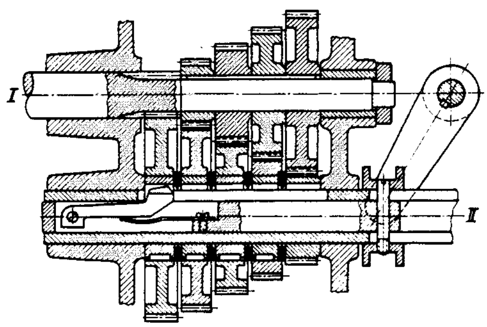


FIG. 147. Draw-key type gearbox.

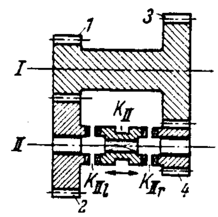


FIG. 148

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.

In the schematic layout of a headstock with preselector gear (Fig. 149), three 2-stepped clutch drives are arranged in series. The three clutches K_{II}, K_{III} and K_{IV} serve for producing $2 \times 2 \times 2 = 8$ speeds (n_1 to n_8) of the main spindle *V*. K_I is the main clutch which connects the gear block (*a*

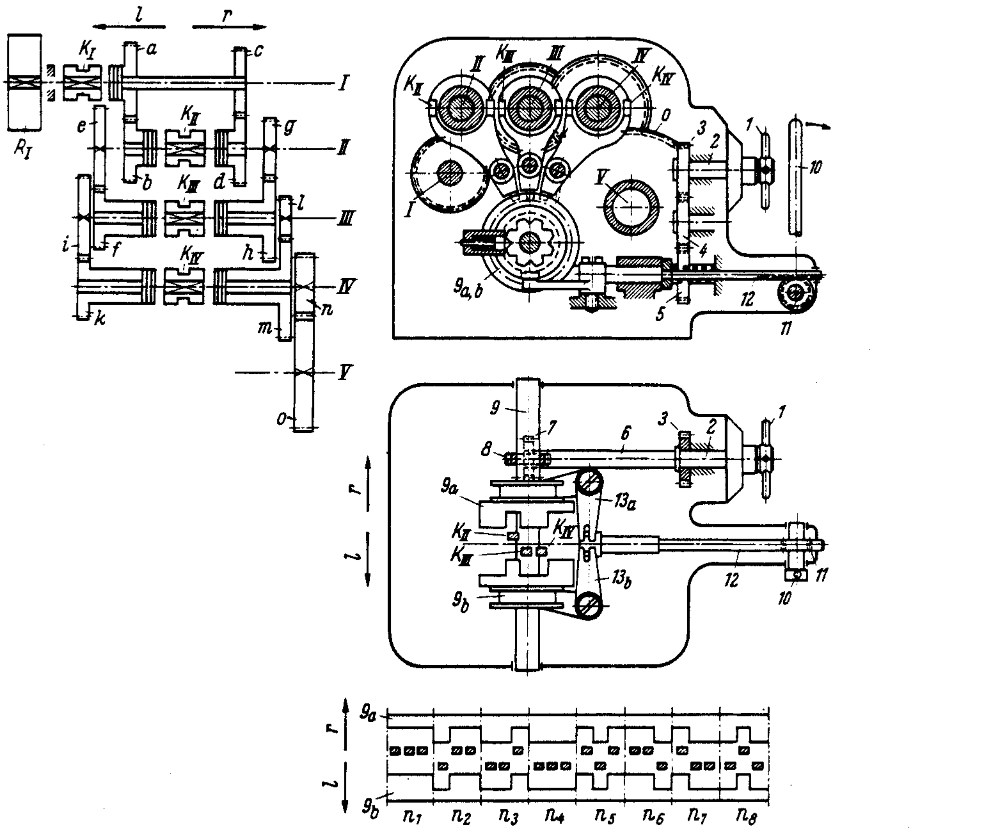


FIG. 149. Preselector gearbox (Alfred Herbert Ltd., Coventry).

and *c*) on shaft *I* with the driving pulley R_I. Gears *e* and *g* are keyed to shaft *II*, gears *i* and *l* to shaft *III*. Gears *b* and *d* are idling on shaft *II*, gears *f* and *h* on shaft *III*, and gears *k* and *m* on shaft *IV*. One of the two gears idling on each shaft can be coupled to the shaft by movement to the right (*r*) or left (*l*) of the clutches K_{II}, K_{III} and K_{IV}, so that the speed diagram (Fig. 150) is obtained. The clutch members are moved by means of forks which, in turn, are either pushed or left in position by the cut-outs or protruding parts respectively of control drums *9a* and *9b* (see development at the right-hand bottom corner of Fig. 149). For example, in order to obtain output speed n_1, all clutches are moved to the right (*r*), for n_2 one clutch is moved to the left (*l*) and two to the right (*r*), etc. In order to prepare the shifting of the sliding forks in accordance with the required subsequent output speed whilst the drive is still in operation and without affecting its work, the control drums *9a* and *9b* are rotated by means of hand-wheel *1*, via shaft *2*, gears *3*, *4* and *5*, shaft *6*, spiral gears *7* and *8* and shaft *9*. They can be located in the required position by a spring loaded index pin in such a manner that the cut-outs or protruding

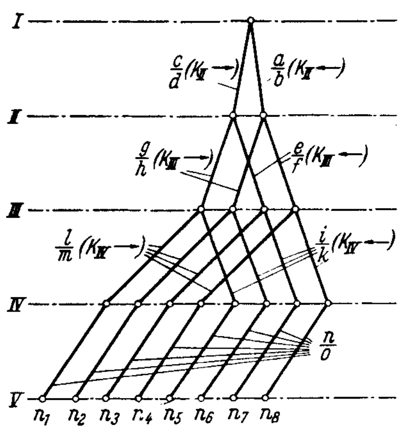


FIG. 150. Speed diagram of the gearbox (Fig. 149).

parts of the drums, which correspond to the required operating position, are brought into a position opposite the sliding forks. If the spindle speed is now to be changed to the preselected value, lever *10* is pulled to the right (arrow). This results in pinion *11*, rack *12* and bell crank levers *13a* and *13b* pushing the two control drums *9a* and *9b* towards each other in such a manner that the cut-outs or protruding parts respectively move the sliding forks for clutches K_{II}, K_{III} and K_{IV} into the required positions and thus connect the gears and shafts which correspond to the preselected transmission ratios. In general, clutch K_I is connected with the preselector mechanism in such a manner that it

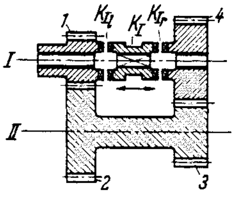


FIG. 151. Back gear drive. (Driving gear *1* is idling on shaft *I*.)

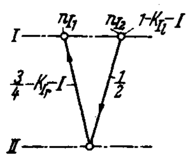


FIG. 152. Speed diagram of the drive (Fig. 151).

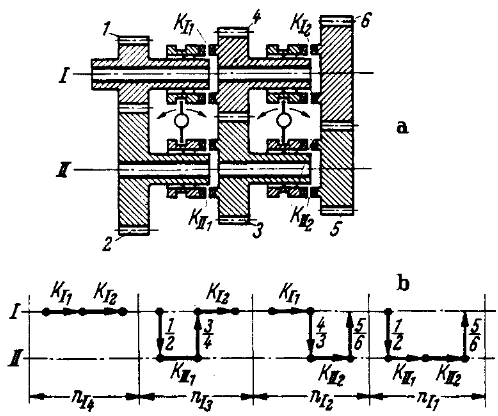


FIG. 153. Ruppert-type gear drive. a) Layout; b) Clutch positions. (Driving gear *1* is idling on shaft *I*.)

allows some slipping during speed changes so that any possible impacts produced by the change from one speed to another are reduced.

Another application of the clutch-type gear is the arrangement in which driving and driven shafts are arranged coaxially so that they can be either coupled directly or driven via one or several reduction gears (back gears).[38] Figure 151 shows such a drive in its simplest form. If clutch K_I is moved to the left (K_{I_l}) the drive goes straight from driving gear *1* to shaft *I*. If the clutch is in its

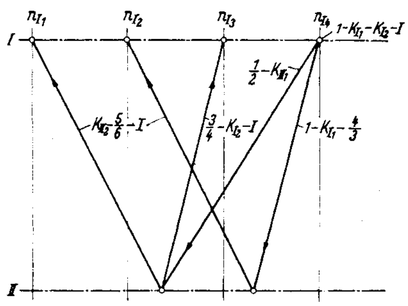
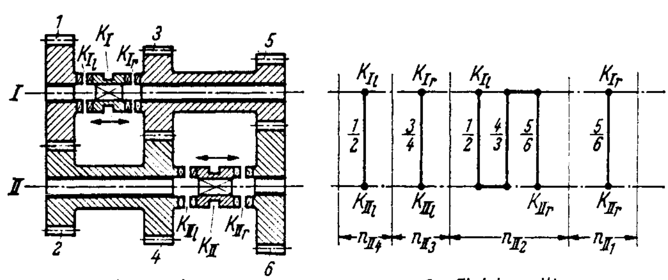


FIG. 154. Speed diagram of the drive (Fig. 153).



a *Lay out* b *Clutch positions*

Fig. 155a and b. *I*—Driving shaft. *II*—Driven shaft.

right position (K_{I_r}) shaft *I* is driven with a reduction ratio $\frac{1}{2} \times \frac{3}{4}$ in accordance with the speed diagram (Fig. 152). Ruppert proposed a further development of this type of gear which is shown in Fig. 153 (speed diagram Fig. 154). A two-axes clutch-type gear with the driving shaft on one and the driven shaft on the other axis, and a back gear of the type just described are arranged in series, and the so-called winding gear drive (Fig. 155, speed diagram Fig. 156) results, in which *I* is the driving *II* the driven shaft. In this design it is impossible, however, to arrange all clutches on the driven shafts, clutch K_I being, for example, on the driving shaft (Fig. 155). In order to reduce the difficulties caused by this arrangement (see page 113), winding gear drives are often equipped with sliding gears (see page 76) (Fig. 157).

Electrically, mechanically or hydraulically operated friction clutches are often applied when speed changes must be possible during running of the gear drive. If, however, positive torque transmission is essential, dog or gear clutches are employed in combination with synchromesh devices.

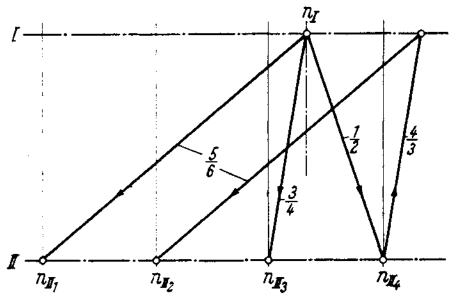


FIG. 156. Speed diagram of the drive (Fig. 155).

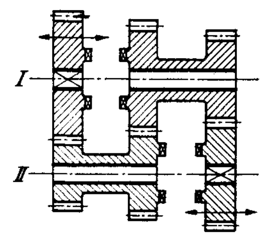


FIG. 157

The main disadvantage of clutch-type gears is their necessary length. This is determined by the length of the sliding parts and that of the bearings on which the idling gears are running on the shafts.[39] As the free length of a shaft between bearings must be kept within certain limits (see page 67), the possible number of clutches which can be arranged on one shaft is also limited. Moreover, the minimum diameter of the idling gears is limited by the diameter of their bearing bushes, and this may result in relatively large diameter gears.

The economic efficiency of a speed change gear design depends, amongst other factors, upon the number of its elements (shafts, clutches, gears). Simple clutch-type gears require one clutch per set of meshing gears. The sliding-gear-type drives, on the other hand, require the same number of gears but no clutches and are, therefore, simpler and more often preferred. Moreover, the moving elements of sliding-gear drives (the sliding gears) may be arranged on either the driving or the driven shaft, as only those gears, which are actually transmitting the torque, are in mesh.

The shortest constructional length is obtained if a narrow sliding-gear block is axially shifted between a wide fixed-gear block (Fig. 158). Unless special precautions are taken the minimum

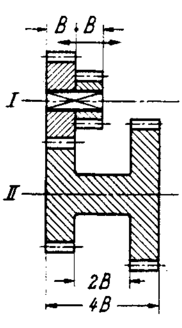


FIG. 158

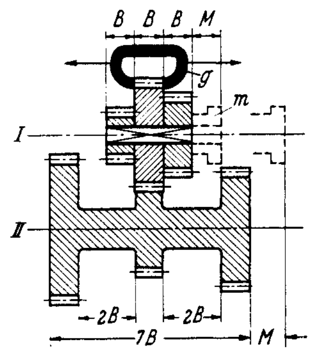


FIG. 159

constructional length is determined by the fact that during the axial movement of the sliding gears, one set of gears must be completely disengaged before the other set begins to come into mesh. The distance between two gears of the fixed block must, therefore, be equal to two gear widths, and the minimum length of a two-stepped sliding-gear drive is, therefore, four (Fig. 158), and that of a three-stepped sliding-gear drive seven gear widths (Fig. 159). A circular groove for the application of the gear-shifting fork m (Fig. 159) would increase the length of the sliding-gear block and with it the required minimum length of the drive. For this reason, sliding forks g which act on the faces of one gear are often employed (Fig. 159).

In the case of three-step drives with constant input speed the sliding-gear block must be in its middle position when either the maximum or the minimum output speed is to be obtained, as otherwise it could not be moved to either side. The shortest arrangement of the sliding-gear block does not, therefore, provide a speed range which is either increasing or decreasing as the gear block is shifted from one extreme position to the other. If a movement of the gear block in one direction is to result in a corresponding increase or decrease of the output speed the minimum length of the gear drive has to be increased to nine gear widths (Figs. 160a and 160b), because the axial space

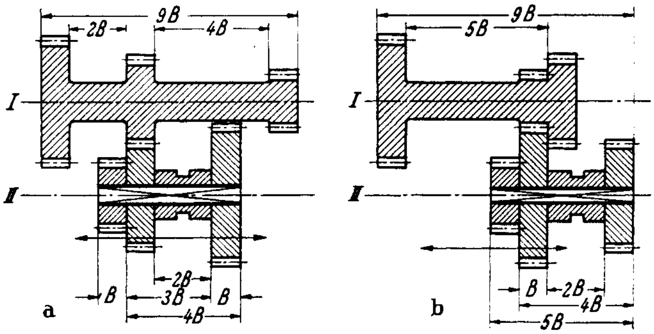


FIG. 160

between the two largest sliding gears must again be at least two gear widths. Such a "logical" arrangement is desirable not only from the point of view of ease of operation, but also because it facilitates the changing from one speed to the next, the difference in circumferential speeds of the gears being a minimum.

If several sliding-gear drives are arranged in series, and if all sliding gears are located on one shaft (Fig. 161a), only one splined shaft is required; on the other hand, the minimum total length is smaller in the arrangement shown in Fig. 161b.

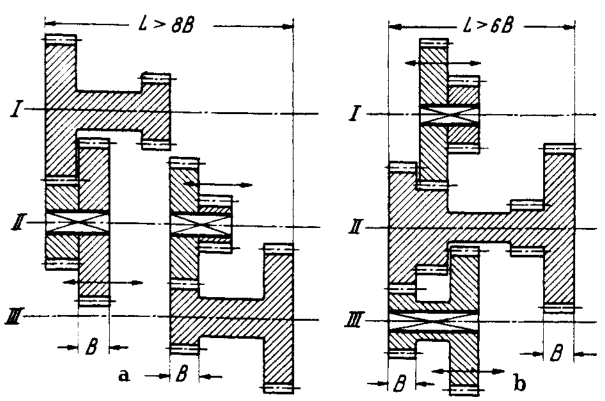


FIG. 161

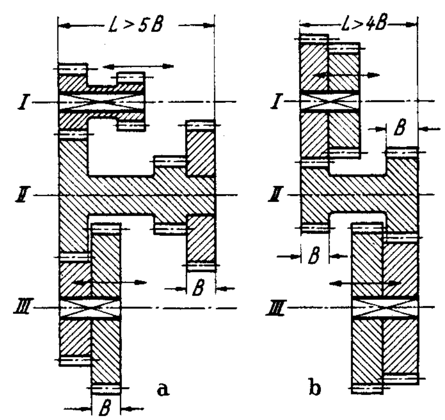


FIG. 162

It is possible to reduce the necessary number of gears in a speed change gearbox which is based on several part-drives arranged in series, if the driven gears of one part-drive serve as driving gears of the next part-drive. A drive with only one gear, which has this dual function, is called a single composite gear (Fig. 162a). A drive with two dual-function gears (driving and driven) is called a double composite gear (Fig. 162b) etc.[40] From the point of view of economy, the use of as many composite gears as possible is advantageous, as the total number of gears is reduced. It is, however, difficult with such arrangements to obtain geometrically-stepped speed ranges, and it has been shown that triple composite gears cannot produce a clean geometrically-stepped speed range.[41] There are only few possible transmission ratios for double composite gears, and these have been worked out by Germar.[28] Apart from space requirements, the number of shafts and gears which are necessary for obtaining a certain number of output speeds is important both from the points of view of simplicity and of economy. Table 17, which is valid for non-composite gear drives, shows that a 9-speed drive requires the same number of gears as an 8-speed drive, and that a 12-speed drive does not require more gears than a 10-speed drive. Similarly, an 18-speed gearbox needs no more gears than a 16-speed gearbox. This is the reason why designers prefer 9-, 12- and 18-speed gearboxes.

TABLE 17

Number of Steps Produced by the Gearbox	Possibilities of Dividing into Part Drives			Minimum Number of Gears in Non-Composite Layouts
4	2·2			8
6	3·2	2·3		10
8	2·2·2			
	4·2	2·4		12
9	3·3			
10	5·2	2·5		
12	3·2·2	2·3·2	2·2·3	14
	4·3	3·4		
	6·2	2·6		
15	5·3	3·5		
16	2·2·2·2			
	4·2·2	2·4·2	2·2·4	16
	4·4			
18	3·3·2	3·2·3	2·3·3	
	6·3	3·6		18

The number of shafts required depends upon the numbers of gears which can be arranged in one sliding block and on the number of sliding-gear blocks which can be arranged in parallel. In addition, the available space must be considered, because the axial or radial dimensions of the drive may be limited (see page 67). The possibility of modifying a 12-speed into an 18-speed gearbox by means of adding a single set of gears has been discussed on page 80.

The preceding considerations refer to the kinematic conditions and the geometric arrangements of the various elements. Although the strength calculations for clutches, shafts and gears of machine tools do not differ in principle from those generally applied, the working conditions assumed in such calculations vary in accordance with their applications.

The loads may be constant or pulsating, the output shafts may have to transmit either constant power or constant torque over the full speed range. If constant power transmission is required, the torque increases with decreasing output speed (see Fig. 128). The strength calculations for all elements of the drive (shafts, clutches, gears) have to be carried out for the minimum speed n_{min} when the maximum torque is $T_{max} = 7000\, N^{(kW)}/n_{min}$ (ft lb). In this case, the whole driving mechanism can be protected against overload by a device which switches the motor off as soon as the maximum power is exceeded. 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:

$$N_{max} = \frac{T^{(\text{ft}\cdot\text{lb})} \times n_{max}}{7000} \quad [\text{kW}]$$

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.

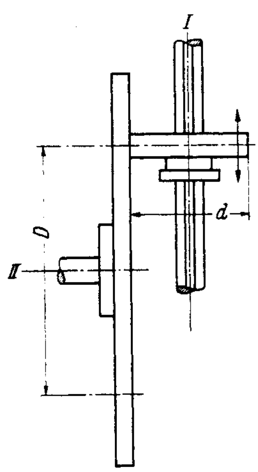


FIG. 163

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.

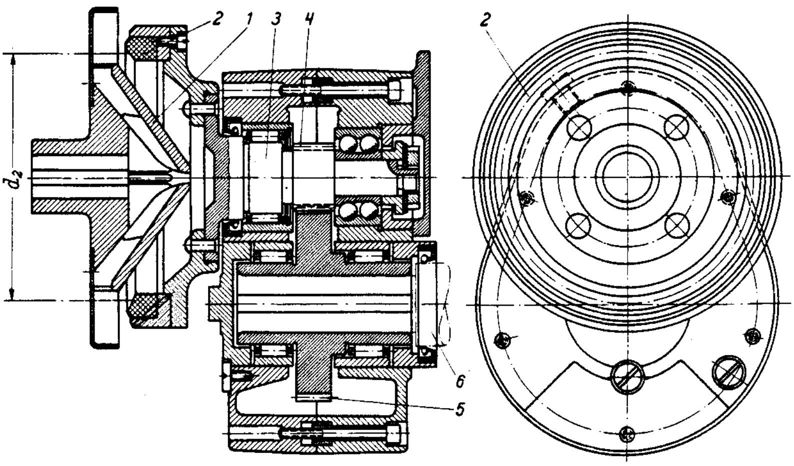


FIG. 164. Friction drive (William Prym).

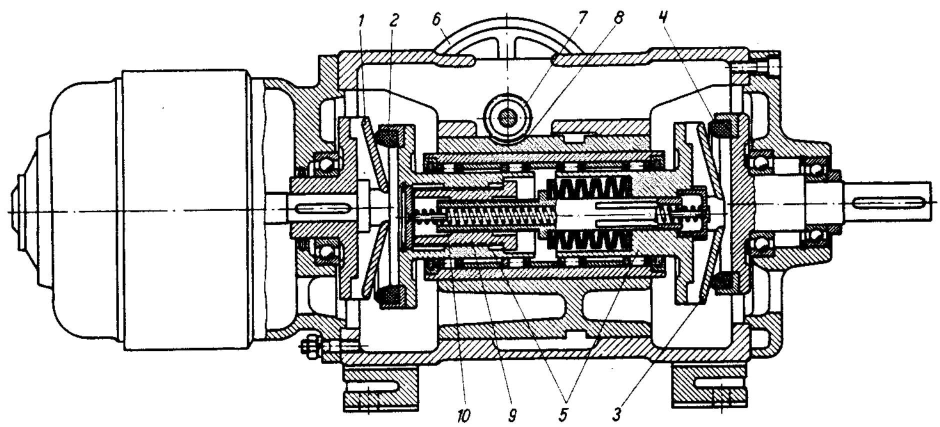


FIG. 165. Friction drive with increased speed-range ratio (William Prym).

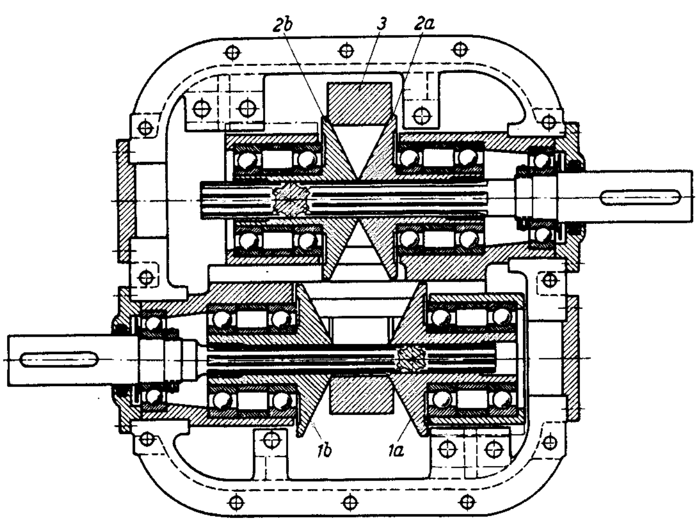


FIG. 166. Heynau drive.

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.

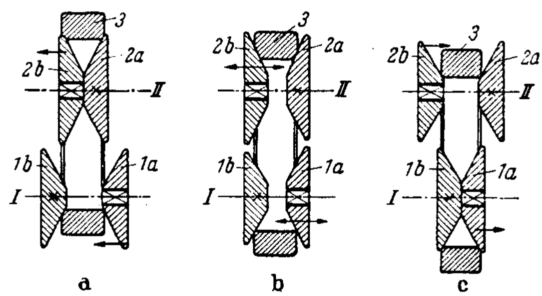


FIG. 167. Working principle of the Heynau drive. *I*—Driving shaft; *II*—Driven shaft.

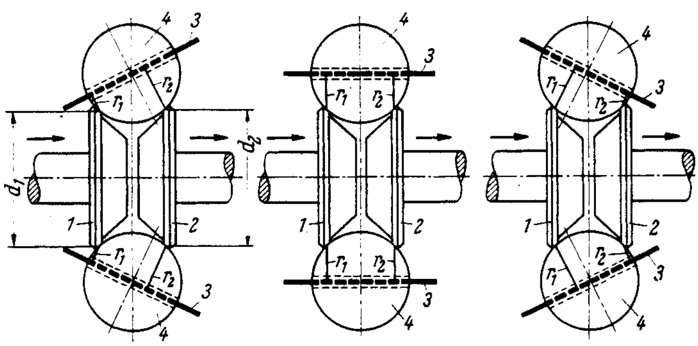


FIG. 168. Working principle of the Wülfel–Kopp Tourator.

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 $d_1 = d_2$

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

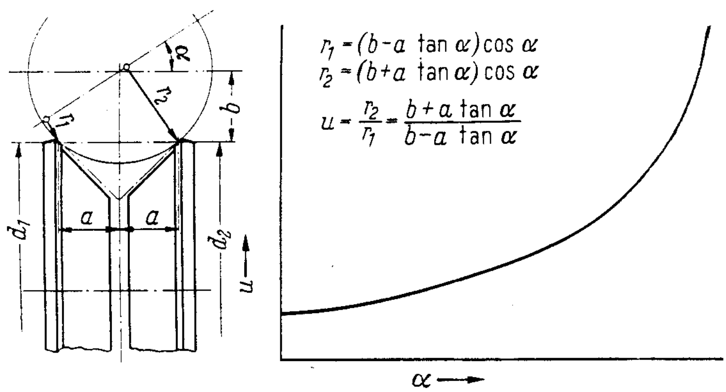


FIG. 169

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

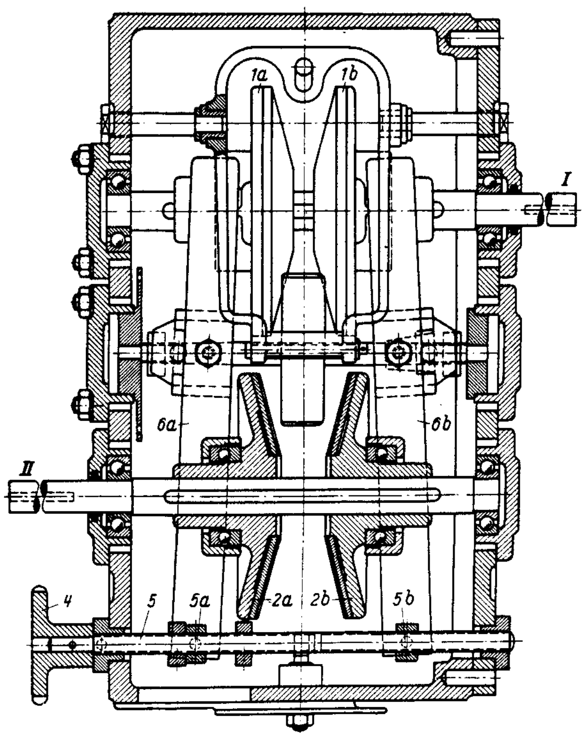


FIG. 170. P.I.V. drive: *I*—Driving shaft (input); *II*—Driven shaft (output).

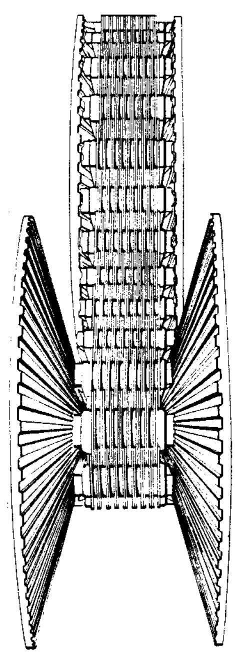


FIG. 171

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*

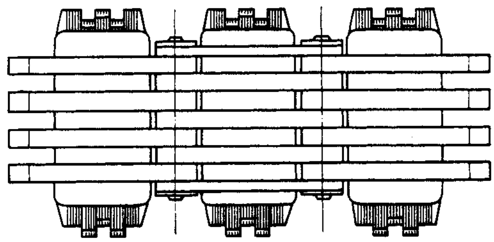


FIG. 172

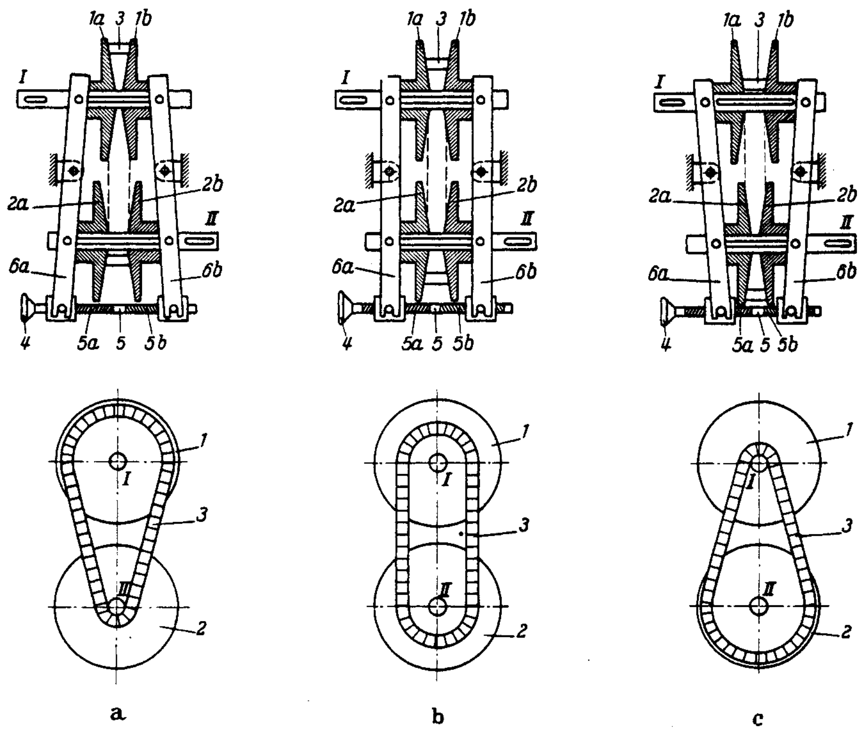


FIG. 173. Working principle of the P.I.V. drive. a—Speed increase; b—1:1 transmission; c—Speed reduction.

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

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

where d_1 and d_2 are the variable effective diameters of the chain wheels on shafts I and II.

With

$$d_{1_{max}} = d_{2_{max}} = d_{max}$$

and

$$d_{1_{min}} = d_{2_{min}} = d_{min}$$

$$n_{II_{min}} = \frac{d_{min}}{d_{max}} \times n_I$$

$$n_{II_{max}} = \frac{d_{max}}{d_{min}} \times n_I$$

The speed-range ratio, therefore, becomes:

$$B = \frac{n_{II_{max}}}{n_{II_{min}}} = \left(\frac{d_{max}}{d_{min}}\right)^2$$

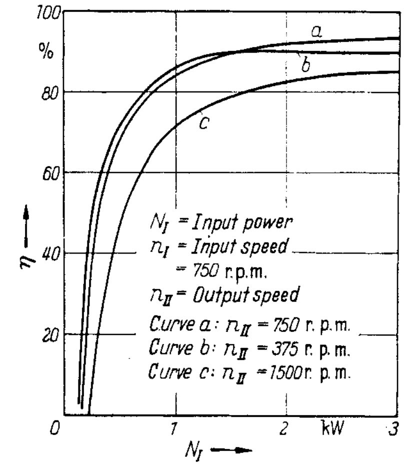


FIG. 174. Typical efficiency curves of a P.I.V. drive.

For a given maximum chain load P, the torque which can be transmitted by the constant speed input shaft decreases with decreasing effective diameter d_1. As a result, the power which can be transmitted decreases in proportion to the output speed. The torque transmitted by the input shaft (I) is:

$$T_I = \frac{P \times d_1}{2}$$

Hence

$$T_{I_{max}} = \frac{P}{2} \times d_{max}$$

$$T_{I_{min}} = \frac{P}{2} \times d_{min}$$

and

$$\frac{T_{I_{max}}}{T_{I_{min}}} = \frac{d_{max}}{d_{min}}$$

In the case of a constant speed input shaft the ratio between powers and torques transmitted is equal.

$$\frac{N_{max}}{N_{min}} = \frac{T_{I_{max}}}{T_{I_{min}}} = \frac{d_{max}}{d_{min}} = \sqrt{B}$$

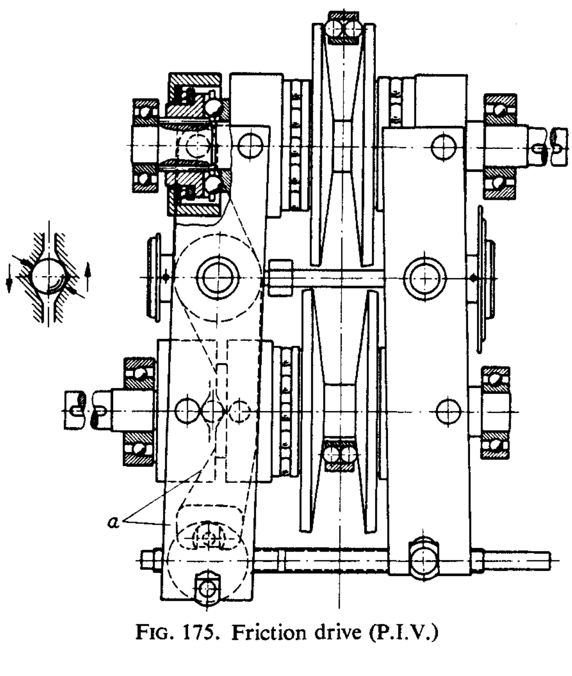


FIG. 175. Friction drive (P.I.V.)

Although in the design of this P.I.V. drive it has been possible to combine the advantages of a stepless drive with the requirements of positive load transmission, it is interesting to note that the manufacturers have also developed a stepless speed-change mechanism having a speed-range ratio up to 10 in which the load transmission is by friction (Fig. 175). In this device the cones have no teeth but smooth, hardened and ground surfaces. The pack of lamellae in each chain link is replaced by two glass-hard steel rollers which, on entry into the chain wheel, are compressed against the cone surfaces under the effect of the chain load, thus transmitting the torque by friction. As the chain links come out of the chain wheel, the wedge action on the rollers is released (Fig. 176). In an alternative design, the axial pressure of the cones against the chain can be adjusted in accordance with the torque via sloping faces on scissor levers a (Fig. 175).

The positively acting P.I.V. drive can transmit up to 40 kW, the friction type design up to 15 kW.

For very high output speeds compressed air can be employed, e.g. by arranging a compressed air turbine on the driven shaft, an example being found in an internal grinding spindle running at speeds up to 120,000 rev/min. Such an arrangement can be well-balanced and the exhaust air serves not only for cooling the bearings, but also for preventing the entry of foreign matter. The output speed can be varied by means of a throttle valve.

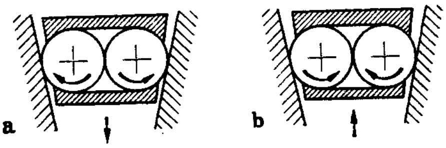


FIG. 176. Working principle of the drive (Fig. 175). a—"Wedging" of the incoming chain link; b—Releasing of the outgoing chain link.

*Hydraulic drives.** In hydraulic drives, oil is generally used as the energy-transmitting medium. The oil, which is put into motion and under pressure by means of a pump, drives a hydraulic motor. It is important to realize that, in most cases, only static energy is transmitted because the velocity of the oil stream in hydraulic drives normally used in machine tools is relatively low so that the kinetic energy of the oil is of little effect. Even if the oil velocity reaches the relatively high value of 32 ft/sec, the velocity height of the oil column is only

$$\frac{v^2}{2g} = \frac{1024}{2 \times 32} = 16 \text{ ft}$$

On the other hand, with an oil pressure of $p = 700$ lb/in^2, and a specific weight of the oil of $\gamma = 0{\cdot}032$ lb/in^3

$$\frac{p}{\gamma} = \frac{700}{0{\cdot}032 \times 12} = 1830 \text{ ft}$$

The velocity height is, therefore, only about 0·9 per cent of the pressure height.

The advantages of hydraulic drives are in their relatively small dimensions at high loading capacity, the possibility of simple and smooth reversing, of stepless speed variation which can be adjusted without difficulties even under load, their quiet running, together with the possibility of programme control and overload protection by means of suitable valve designs.

Disadvantages are the cost of producing the accurate parts which have to be assembled with relatively tight fits and the heating caused by throttling phenomena. The latter may not only affect the accuracy of the machine but also the viscosity of the oil, so that positive transmission of movement between elements connected by hydraulic drives and absolute accuracy of working speeds are not easily obtainable. Mention may also be made of the possible slip due to leakage losses which contribute towards reducing the efficiency of such drives. Moreover, the compressibility of the oil, however small, and the elasticity of the oil carrying elements, especially long pipelines, may seriously affect the stiffness of the load transmission and with it the vibration properties of the machine and its elements.

The hydraulic oil must have good lubricating properties and a flat viscosity–temperature curve. It must not be subject to ageing effects.

The compressibility depends upon temperature and pressure and is

$$\frac{V_1 - V_2}{V_1} = \beta \times (p_2 - p_1)$$

where V_1 is the initial volume, V_2 the compressed volume, p_1 the initial pressure, p_2 the final pressure and β a compressibility coefficient which has a value of 0·0000058 at 122°F (50°C) and 700 lb/in^2.

If the oil is free from air inclusions, the value of β depends only upon the oil temperature and the oil pressure. The effect of both pressure and temperature on β is, however, small compared with the effect which air inclusions have upon the compressibility (see Fig. 177).

* For a detailed study of this subject see ref. 44.

The elastic shortening of an oil column, in a system of length L and cross-section A under a pressure difference $p_2 - p_1 = p$, can be calculated as follows:

$$V_1 = A.L$$

$$V_2 = A(L - \Delta L)$$

$$\frac{\Delta L}{L} = \beta \times p$$

$$\Delta L = \beta \times p \times L$$

The oil volume in the pipelines must not be neglected in such calculations.

Dürr and Wachter[44] suggest the following permissible oil velocities:

Suction lines	300 ft/min
Drains	400 ft/min
Pressure lines up to 350 lb/in^2	600 ft/min
Pressure lines up to 700 lb/in^2	800 ft/min

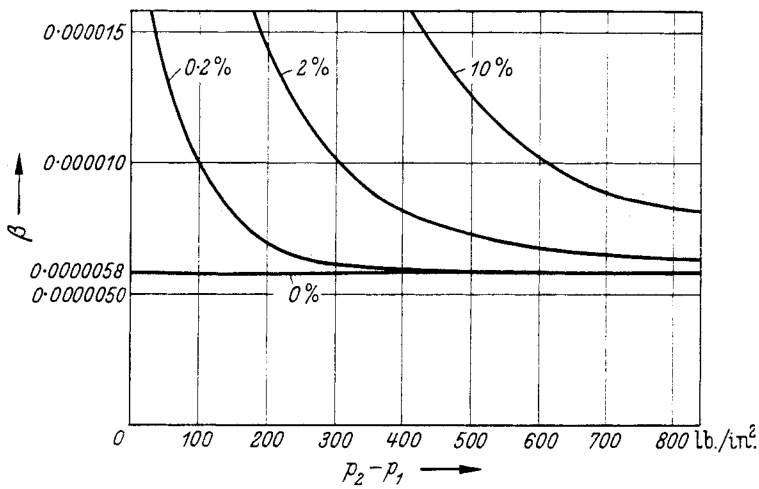


FIG. 177. Effect of air inclusion on the compressibility of oil.

In very short channels or bores and in pressure valves, oil velocities may be higher on occasions. They must, however, be avoided at all costs in long pressure, suction and drain lines. If the oil velocity in a suction line is too high, the resulting pressure drop may lead to air inclusion in the oil which, apart from being detrimental to the functioning of the hydraulic drive, has a greater effect upon the compressibility of the oil than pressure and temperature. Figure 177 shows values for β at 122°F (50°C) as a function of the pressure, for the cases of 0 per cent (ideal case), 0·2 per cent, 2·0 per cent and 10 per cent air inclusion.[45] With increasing pressure, the effect of the air inclusion decreases. However, at 150 lb/in^2, the small air inclusion of 0·2 per cent increases the compressibility coefficient β by over 40 per cent!

In order to ensure that the hydraulic devices work satisfactorily, means for not only cooling and filtering the oil, but also for the removal of any possible air inclusions are most important.

The main elements of hydraulic drives, apart from the oil and the pipe connexions, are the pump, the motor and the control devices.

Pumps

The pump is usually driven at constant speed by an electric motor. It takes the oil from a tank and delivers it through the pipelines and control devices to the hydraulic motor. The net power N

of the pump is determined by the volume it delivers (Q, in^3/min) and the working pressure (p, lb/in^2):

$$N = \frac{Q \times p}{530{,}000} \quad \text{(kW)}$$

The pumps which are used in machine tools can be divided into two groups:

1. *Constant delivery pumps*

Gear pumps (Fig. 178) are designed for pressures up to 1500 lb/in^2 and at times more. The delivery of these pumps depends upon their rotational speed and upon the size and pitch of the gear teeth. Each tooth displaces the volume of oil which is contained in the corresponding gap between two teeth in the meshing gear, and is equal to the volume of the tooth measured between its own outside diameter and that of the opposing gear wheel.

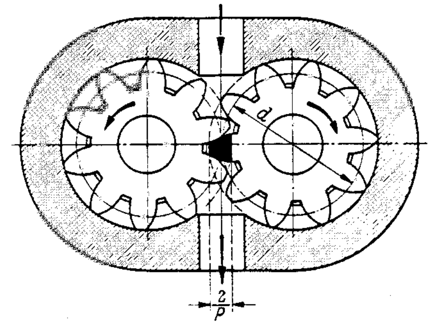


FIG. 178. Gear pump.

During one revolution, each gear wheel with a pitch circle diameter d, diametral pitch P and width b, delivers a volume equal to half that of a ring with outside diameter $d + 1/P$, inside diameter $d - 1/P$ and width b. During one revolution the two meshing gears deliver, therefore a volume

$$V = 2 \times \tfrac{1}{2}\pi d \times \frac{2}{P} \times b = 2\pi \times \frac{db}{P}$$

If each gear rotates at n rev/min, the volume delivered per minute is

$$Q = 2\pi \times \frac{db}{P} \times n$$

If the number of teeth of each gear is n_t, and with $d = n_t/P$

$$Q = 2\pi \times \frac{n_t \times b \times n}{P^2} \quad [\text{in}^3/\text{min}]$$

If the gears run with practically no backlash and are well-fitted in the bore and between the faces of the pump body, leakage losses can be kept within reasonable limits. As the gap between two meshing teeth varies during their engagement, the resulting small amount of high pressure oil will make the pump run hard and rough. This can be prevented by relieving channels in the faces of the housing, through which the high pressure oil can escape into the pressure chamber. For high working pressures (up to 1500 lb/in^2), pumps with a large number of teeth, small pitch and high rotational speeds are employed. At lower pressures, lower rotational speeds result in quieter running conditions. In this case, it may be necessary, however, to use larger teeth in order to deliver the required oil volume per unit time. Quieter running conditions can also be obtained by the use of helical gears or double helical gears, the latter avoiding axial tooth pressure components. With helical gears, it is important, however, to ensure that the helix angle does not become so large that one gap between two teeth connects both the suction and the pressure chamber of the pump!

The volumetric efficiency of a gear pump depends upon the pressure and the accuracy of manufacture since this affects the leakage losses. Figure 179 shows typical delivery curves of gear pumps.

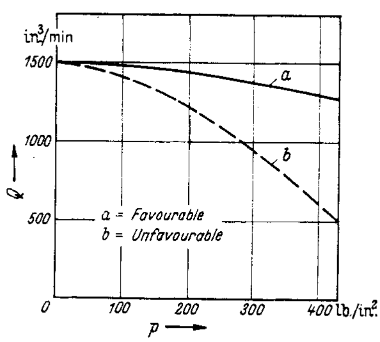


FIG. 179. Typical delivery characteristics of gear pumps.

A development of the gear pump principle is the screw conveyor pump, in which two, three or five right- and left-hand threaded shafts work in the same manner as the gears in a gear pump.

A variation of the delivery is usually only possible by letting a surplus of oil escape through a suitable valve, or by throttling the pressure line and letting oil escape through an overload valve. Such "loss adjustment" is, however, permissible only for cases in which the total power is small and the resulting power losses can be tolerated (see page 157).

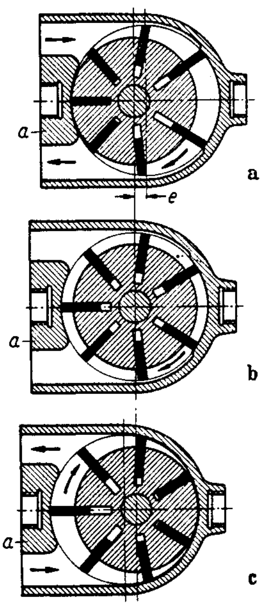


FIG. 180. Vane pump.

2. *Variable delivery pumps*

Vane pumps. The pumping action results from the vanes making reciprocating movements in the radial slots of an eccentrically supported rotor. The stroke of the vanes is proportional to the eccentricity between the rotor and the bore in the pump body. For a given rotational speed and direction the volume and direction of delivery is thus determined by the eccentricity of the rotor (Fig. 180). The axis of the rotor is generally fixed and that of the pump body displaced for varying the delivery. When the rotor and the bore in the pump body are concentric (Fig. 180b), the spaces between the rotor, the pump body and the vanes remain constant during rotation and the pump does not deliver. If the concentricity is as shown in Fig. 180a, the spaces between the rotor, the pump body and the vanes increase during rotation at the top half of the pump, and decrease correspondingly at the bottom half. This means that suction occurs at the top and delivery at the bottom. If the eccentricity lies in the opposite direction (Fig. 180c), delivery occurs in the opposite direction also. The volume delivered depends upon the stroke of the vanes, i.e. on the eccentricity *e*. It can be varied smoothly from a maximum in one direction through zero to a maximum in the opposite direction. The suction and pressure chambers are separated by the partition *a* in the pump body. Figure 181* shows the design of a vane pump. The eccentricity *e* is varied by moving the pump body to the left or to the right in relation to the fixed axis of the driving shaft *b* in the housing *c*. In order to keep

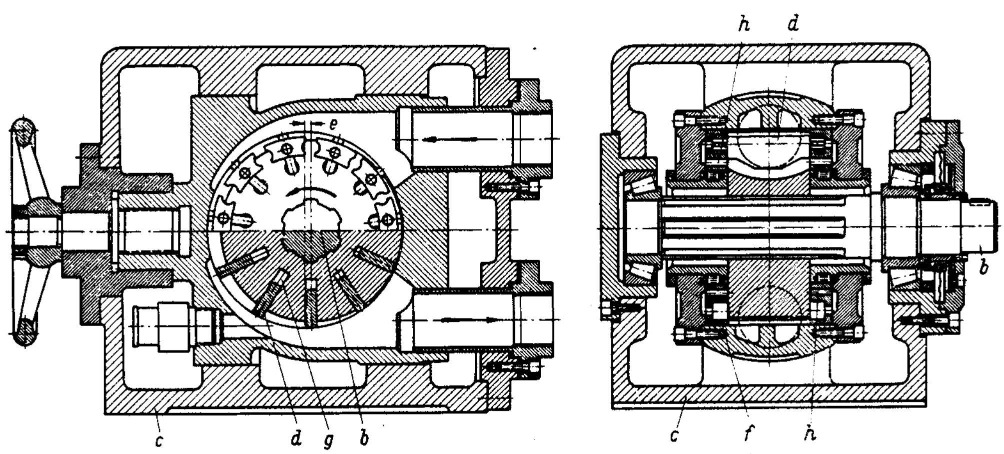


FIG. 181. Forst–Enor vane pump.

mechanical losses low the friction between the vanes *d* and the bore in the pump body which controls their radial movement, is reduced by the provision of rings *f*. The spaces *g* underneath the vanes *d*, which vary during each revolution between a maximum and a minimum, are relieved by holes *h*.

* From Dürr & Wachter, ref. 44.

As the spaces g are connected alternately with the suction and the pressure chamber, an additional pumping effect results. The volume thereby delivered compensates for the reduction in volume caused by the thickness of the vanes (see page 130).

The friction can also be reduced if the oil is supplied from the centre of rotation of the pump. In this case, the whole body of the pump, which is eccentrically adjustable, rotates with the rotor around a fixed shaft a, (Fig. 182). The oil is both drawn into the pump and delivered through

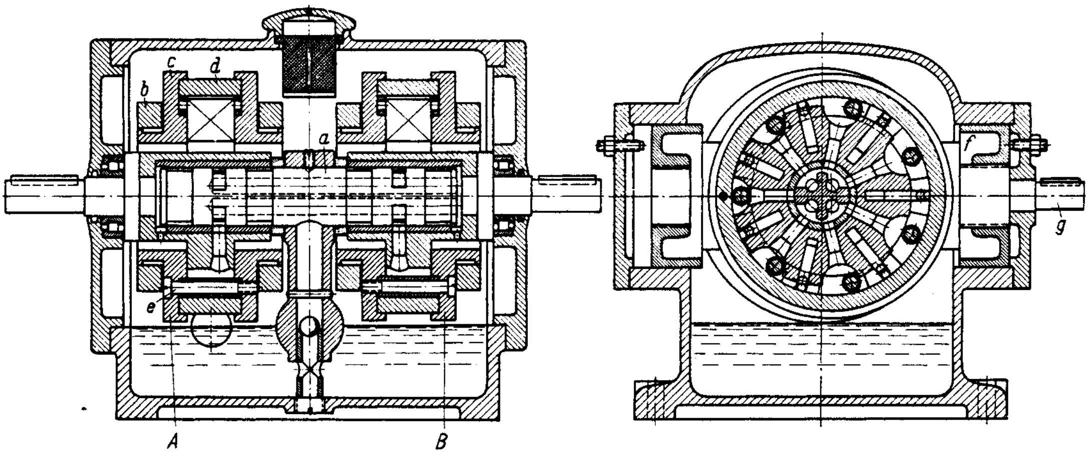


FIG. 182. Vane pump and motor (from, SCHLESINGER, *Die Werkzeugmaschinen, loc. cit.*). *A*—Pump; *B*—Motor; *a*—Fixed supporting shaft; *b*—Movable carrier; *c*—Pump cover; *d*—Cylindrical pump body; *e*—Fixing screw; *f*—Piston; *g*—Adjusting screw.

bores in shaft a. The pump body consists of a main annular part d and two covers c, held together by screws e. It runs in needle bearings which are located in supports b and rotates together with the driving shaft, so that the relative circumferential velocity between vanes and pump body, and with it the friction, are small. This arrangement also makes possible the reduction of play between faces which have to form oil-tight joints. In these pumps, the spaces underneath the vanes have to be connected with the external oil chambers and it is necessary to consider in the calculation the amount of loss in delivery which is determined by the thickness of the vanes.

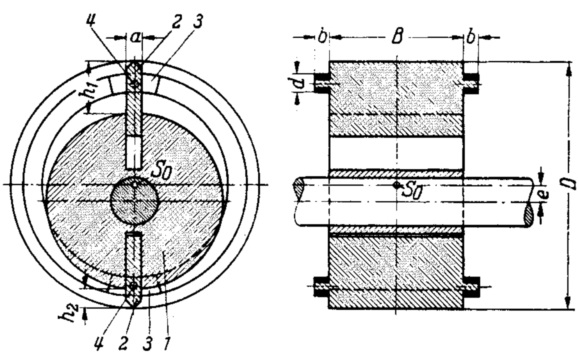


FIG. 183

The volume delivered per minute Q of a vane pump can be calculated as the product of the rotating area A and the path which its centroid describes during one revolution (Fig. 183). The area A is equal to the projection of the rotor *1* and the vanes *2*, plus the projection of the guide pieces *3* and their rollers *4*.

If the rotor makes n rev/min, the path per minute s of the centroid S_0 moving at radius e is:

$$s = 2\pi en$$

and

$$\begin{aligned} Q &= A \times 2\pi en \\ &= (BD + 4bd) \times 2\pi en \\ &= \pi en \times (2BD + 8bd) \end{aligned}$$

If the volume underneath the vanes does not replace the volume lost due to the thickness of the vanes, (see page 129), the reduction in volume delivered for each vane due to the thickness of the vanes is:

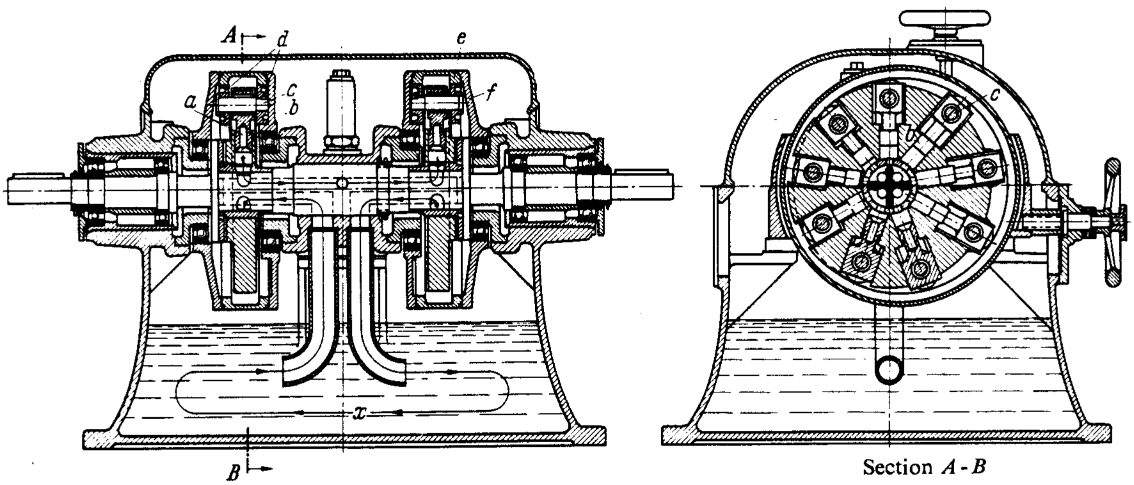
$$Q' = aB(h_1 - h_2)$$

(see Fig. 183) which is exactly the volume below each vane. If there are z vanes and if $h_1 - h_2 = 2e$, the reduction in volume per minute is

$$Q_R = 2enzaB$$

and the actual volume delivered by the pump is

$$Q = \pi en(2BD + 8bd) - 2enzaB$$



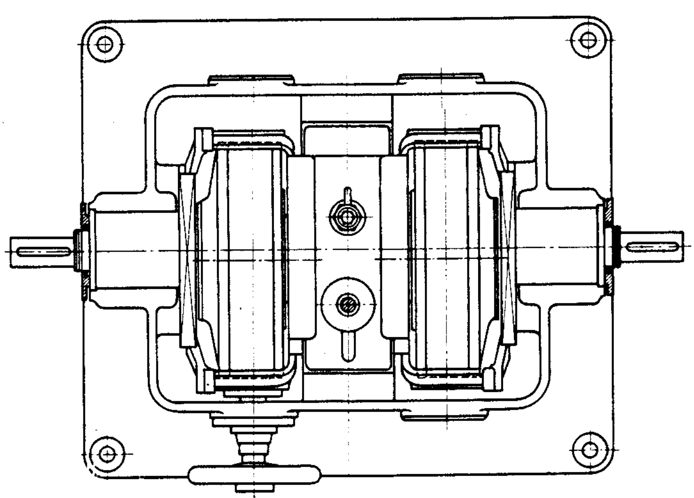


FIG. 184. Lauf-Thoma (radial piston) pump and motor unit (from G. SCHLESINGER, *Die Werkzeugmaschinen*, *loc. cit.*). *a*—Rotor; *b*—Piston; *c*—Gudgeon pin; *d*—Roller; *e*—Guide for roller *d*; *f*—Movable pump body; *x*—Oil circuit.

Piston pumps. The manufacture of the cylindrical sliding faces of pistons and cylinders in a piston pump is easier than that of the rectangular vane sections, which must move with minimum play in the slots of the rotor. As a result, leakage losses in piston pumps are usually smaller, even if the pressures employed are higher. The pistons can be arranged radially or axially.

Figure 184 shows a radial arrangement (combined pump and motor unit) with the oil supply and delivery through the centre of the pump. The radial cylinder bores in the rotor *a* guide the pistons *b*, the movements of which are controlled by the rollers *d*. These rollers are located on the gudgeon pins *c* and move in the guiding slots *e* of the pump body *f*. The stroke of the pistons and with it the delivery of the pump can be varied by changing the eccentricity between rotor and pump body. Another pump which uses a radial piston arrangement and which is particularly successful, because of its simple and compact design, is the Oil Gear pump.[44]

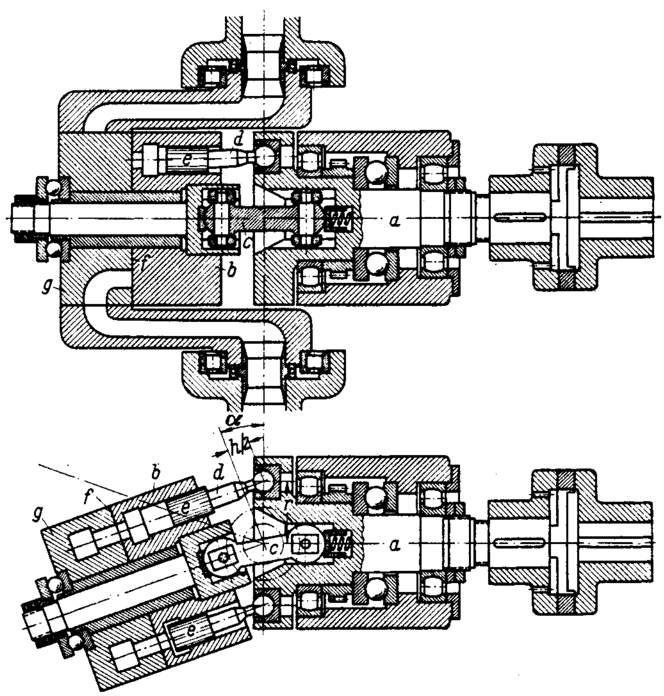


FIG. 185. Jahns-Thoma (axial piston) pump (from G. SCHLESINGER, *Die Werkzeugmaschinen, loc. cit.*). *a*—Driver; *b*—Pump body; *c*—Universal joint; *d*—Piston rod; *e*—Piston; *f*—Thrust face; *g*—Tilting frame; *h*—Piston stroke; α—Tilting angle.

In an axial piston arrangement, the piston stroke can be varied either by changing the angular position of a swashplate, or by tilting the pump body which houses the cylinders. In the pump shown in Fig. 185, the driver *a* transmits the rotational movement to the body *b* via the universal joint *c*. The movement of the pistons *e* is controlled by the piston rods *d*, the ends of which are held in spherical seatings in the driver *a*. The face of the pump body *b* fits tightly against a corresponding face *f* of the frame *g*. Ports are arranged in face *f* in such a manner as to act as distributors which guide the oil from the inlet side into the cylinders and hence to the outlet on the other side. The frame *g* can be tilted (α) around journals through which pass the oil inlet and outlet.

If a piston pump is equipped with *z* pistons, each piston having a diameter *d* and a stroke *h* and if the driver and the pump body rotate at *n* rev/min, the volume delivered per minute is

$$Q = \frac{\pi d^2}{4} \times h \times z \times n$$

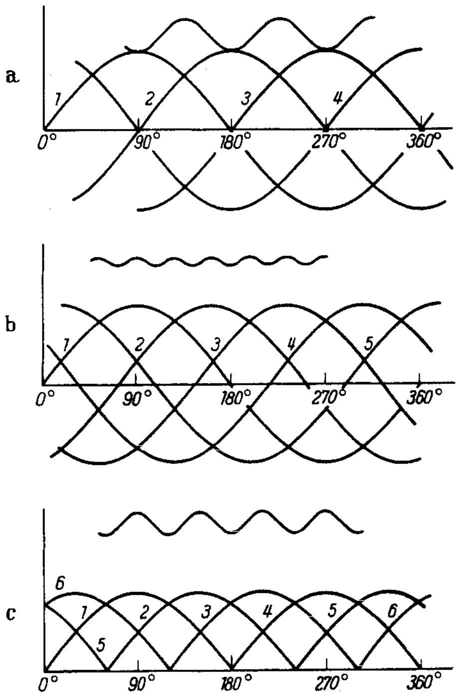


FIG. 186. Pulsations of oil delivery. *a*—4 vanes or pistons; *b*—5 vanes or pistons; *c*—6 vanes or pistons.

If the pistons are arranged radially and the eccentricity is e, then $h = 2e$ and

$$Q = \frac{\pi}{2} \times e \times n \times z \times d^2$$

If the pistons are arranged axially (see Fig. 185), then

$$h = 2r \times \sin \alpha$$

and

$$Q = \frac{\pi}{2} \times r \times \sin \alpha \times n \times z \times d^2$$

In both vane and piston pumps, delivery pulsations are unavoidable. They are less pronounced in pumps with an odd number of vanes or pistons compared with those obtained from pumps with an even number of pistons, even if in the latter case the actual number of pistons is larger. Figure 186 shows the superimposed volumes delivered by the separate pistons, the variation being in accordance with a sine function. It will be seen that the pulsation of the 5-piston arrangement is of smaller amplitude than that of the 6-piston arrangement.

The power output of all pumps depends upon the product of the volume delivered per unit time and the pressure. In other words, the same powers can be produced by pumps with low delivery and high pressure and by those with large delivery and low pressure. If pressures are kept low, mechanical stresses are also low and the problems of making oil-tight joints are less difficult to solve. However, energy losses, space requirements and the detrimental effect of air inclusions, are considerably smaller in high pressure systems (see Fig. 177).

Gear pumps are used for pressures up to 1500 lb/in^2, vane pumps for pressures up to 350 lb/in^2, and piston pumps for pressures up to 2000 lb/in^2, and even higher.

In order to determine the driving power required for a pump, the pump efficiency (η) must be known, because

$$N_{\text{drive}} = \frac{Q \times p}{530000 \times \eta}$$

For pumps which are at present on the market, efficiencies varying between 60 and 85 per cent can be assumed.

Motors

All pumps described in the previous paragraphs can be made to work as hydraulic motors by reversing their operation. In this case, the rotational speed depends upon the oil volume supplied per unit time. Gear pumps are rarely used as hydraulic motors, because at low speeds and increasing oil temperatures, the slip would reach considerable magnitude. On the other hand, hydraulic motors of the vane or piston type are often combined with pumps of the same design in order to form hydraulic driving units with variable output speeds (see Figs. 182 and 184). Instead of the delivery Q_1 of the pump, the volume absorbed Q_2 by the motor has to be considered. The calculation is identical with that already described for pumps (see page 129). If the efficiency of a hydraulic motor were 100 per cent, the power at the output shaft would be:

$$N_2 = \frac{Q_2 \times p_2}{530\,000} \text{ kW} \qquad (Q_2 \text{ in}^3/\text{min}, \ p_2 \text{ lb/in}^2)$$

At an output speed of n_2, the torque would be:

$$T = \frac{530\,000}{2\pi n_2} \times N_2 = \frac{Q_2 . p_2}{2\pi n_2} \qquad (N_2 \text{ kW}, T \text{ in/lb}, n_2 \text{ rev/min})$$

In the case of vane motors supplied from the outside diameter, the oil volume absorbed is

$$Q_2 = \pi e_2 n_2 \times (2B_2D_2 + 8b_2d_2)$$

(see page 130) and the torque, therefore:

$$T_2 = \frac{\pi e_2 n_2 \times (2B_2D_2 + 8b_2d_2) . p_2}{2\pi n_2}$$

$$= (B_2D_2 + 4b_2d_2) \times e_2 p_2$$

Similarly, in the case of radially arranged piston motors:

$$Q_2 = \frac{\pi}{2} \times e_2 \times n_2 \times z_2 \times d_2^2$$

$$T_2 = \frac{\pi \times e_2 \times n_2 \times z_2 \times d_2^2 \times p_2}{2 \times 2 \times \pi n_2}$$

$$= \tfrac{1}{4} . e_2 \times z_2 \times d_2^2 \times p_2$$

For axially arranged piston motors:

$$Q_2 = \frac{\pi}{2} r_2 \times \sin \alpha_2 \times n_2 \times z_2 \times d_2^2$$

$$T_2 = \frac{\pi r_2 \times \sin \alpha_2 \times n_2 \times z_2 \times d_2^2 \times p_2}{2 \times 2 \times \pi n_2}$$

$$= \tfrac{1}{4} r_2 \times \sin \alpha_2 \times z_2 \times d_2^2 \times p_2$$

If no losses occurred in the pump or the motor, the power input and the power output would be equal. As the pressure drop is negligible in the very short connexion between pump and motor units generally used in machine tool drives $p_1 = p_2 = p$, the volume to be delivered by the pump would then have to be equal to the volume absorbed by the motor $Q_1 = Q_2$. However, it is necessary to consider both friction and leakage losses. The former reduce the output torque of the motor, the latter cause a drop in the output speed. If the mechanical efficiency of the motor, determined by the friction losses, is η_{m_2}, the net output power would be

$$N_{2\,\text{net}} = Q_2 \times p \times \eta_{m_2}$$

In order to obtain the required output power $N_2 = Q_2 \times p$, it would be necessary either to increase the volume absorbed by the pump from Q_2 to $Q_2' = Q_2/\eta_{m_2}$ or to increase the pressure from p to $p' = p/\eta_{m_2}$. In the former case an increase in the volume delivered by the pump would not be sufficient because the power increase would be obtained merely by a higher output speed, assuming that the dimensions of the motor remain unaltered. In order to obtain the required power output without changing the output speed, the volume absorbed by the motor must be raised by increasing the motor dimensions, or in other words, by increasing the torque at equal output speed. In this case, the required volume which the pump has to deliver would be

$$Q_1 = \frac{Q_2}{\eta_{m_2}}$$

and to this must be added the volume of oil required to compensate for leakage losses Q_s. In other words, the pump has to deliver a volume

$$Q_1 = \frac{Q_2}{\eta_{m_2}} + Q_s$$

In addition, if the mechanical efficiency of the pump is η_{m_1} the power required for driving the pump is

$$N_1 = \frac{Q_1 \times p}{\eta_{m_1}} = \frac{(Q_2 + \eta_{m_2} \times Q_s) \times p}{\eta_{m_1} \times \eta_{m_2}}$$

With a power output of the motor

$$N_2 = \frac{Q_2}{\eta_{m_2}} \times p \times \eta_{m_2} = Q_2 \times p$$

the total efficiency of the pump-motor unit is

$$\eta_{\text{total}} = \frac{N_2}{N_1} = \frac{Q_2 \times p \times \eta_{m_1} \times \eta_{m_2}}{(Q_2 + \eta_{m_2} \times Q_s) \times p}$$

$$= \frac{\eta_{m_1} \times \eta_{m_2}}{1 + \eta_{m_2} \times Q_s/Q_2}$$

If the volume delivered by the pump or absorbed by the motor respectively are:

$$Q_1 = c_1 \times e_1 \times n_1 = \frac{Q_2}{\eta_{m_2}} + Q_s$$

and

$$\frac{Q_2}{\eta_{m_2}} = c_2 \times e_2 \times n_2$$

c_1 and c_2 being constants which depend upon the design of the drive, these may be equated as follows:

$$c_1 \times e_1 \times n_1 - Q_s = c_2 \times e_2 \times n_2$$

The output speed of the motor is then given by

$$n_2 = \frac{c_1 e_1 n_1 - Q_s}{c_2 e_2} \quad \text{[rev/min]}$$

For reasons of simplicity, it can be assumed that Q_s depends entirely upon the oil pressure. At constant oil pressure, the output speed of the motor depends only upon the eccentricities e_1 and e_2 and is proportional to the eccentricity of the pump (Fig. 187a) and inversely proportional to

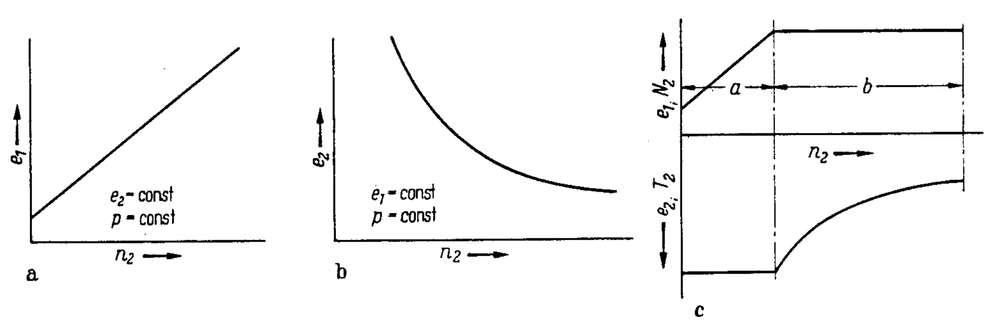


FIG. 187. Speed variation by changing a) Pump eccentricity; b) Motor eccentricity; c) a = Pump, b = Motor eccentricity.

the eccentricity of the motor (Fig. 187b). If the eccentricity of the *pump* alone is variable, the output power is:

$$N_2 = \frac{Q_2}{\eta_{m_2}} \times p \times \eta_{m_2}$$

$$\frac{Q_2}{\eta_{m_2}} = Q_1 - Q_s = c_1 \times e_1 \times n_1 - Q_s$$

$$N_2 = (c_1 \times e_1 \times n_1 - Q_s) \times p \times \eta_{m_2}$$

In other words, when the eccentricity of the pump alone is variable, the output power is proportional to the output speed and the torque is constant.

If, on the other hand, the eccentricity of the *motor* is variable, the output power is

$$N_2 = c_2 \times e_2 \times n_2 \times p \times \eta_{m_2}$$

and the output speed is

$$n_2 = \frac{1}{e_2} \times \frac{c_1 \times e_1 \times n_1 - Q_s}{c_2}$$

In other words, if the pump eccentricity remains constant, $e_2 n_2$ is also constant and the output power is

$$N_2 = c_2 \times \text{const} \times p \times \eta_{m_2}$$

When only the eccentricity of the motor is variable, the output power remains constant over the whole range of output speeds and the torque decreases with increasing output speeds.

The widest range of output speeds of a pump-motor unit is obtained in a combined system in which the eccentricities of both pump and motor can be varied. In this case, it is often advisable to use the variation of the pump eccentricity for the low output speeds at which high torques are also required, whilst the variation of the motor eccentricity covers the high speeds at which constant output power is desirable. These relationships are shown in Fig. 187c.

Figure 188 shows the characteristics of a vane pump-motor unit in which the eccentricity of both pump and motor can be varied simultaneously. With increasing output speed, the pump adjustment provides an increase in the output power. The effect of the motor adjustment is a decrease of the output torque.

An interesting point is the effect of the oil pressure upon the total efficiency. At low pressure, leakage losses are relatively small whilst the mechanical efficiency increases with higher pressure. At high speeds and for pressures above 140 lb/in^2, these two influences are approximately in balance, so that the total efficiency remains almost constant. However, at low output speeds the leakage losses have an overriding effect, so that the total efficiency drops with increasing pressure and is lower at 140 lb/in^2 than at 85 lb/in^2.

Two limitations of the system in which the eccentricity of the motor is varied must be mentioned.

1. The output speed of the motor shaft is inversely proportional to the eccentricity e_2. This means that for extremely small eccentricities, the output speed would approach infinity. However, even for the case of very small eccentricities the motor would stall when the torque, which drops with decreasing eccentricity, becomes too small to overcome the friction resistance. In this case, the volume of oil which the motor could absorb being zero, the volume delivered by the pump could not be taken up by the motor and damage would result. For this reason the minimum eccentricity of the motor must be limited, unless an overload valve is provided (see Fig. 184).

2. Arising out of 1, it follows that the variation of the pump eccentricity is preferable, unless the requirement of a very wide speed range demands both pump and motor adjustment. For reversing the output speed, the reversal of the pump eccentricity is again generally preferred, as otherwise the delivery of the pump must be stopped or the oil delivered by the pump switched to the tank prior to the motor eccentricity passing through zero during change from rotation in one direction to that in the opposite direction.

(b) *Rectilinear Reciprocating Drives*

These mechanisms serve for transforming a rotational movement of the input shaft into the rectilinear movements required for the cutting action or for feeding or setting the machine element which carries the workpiece or the tool.

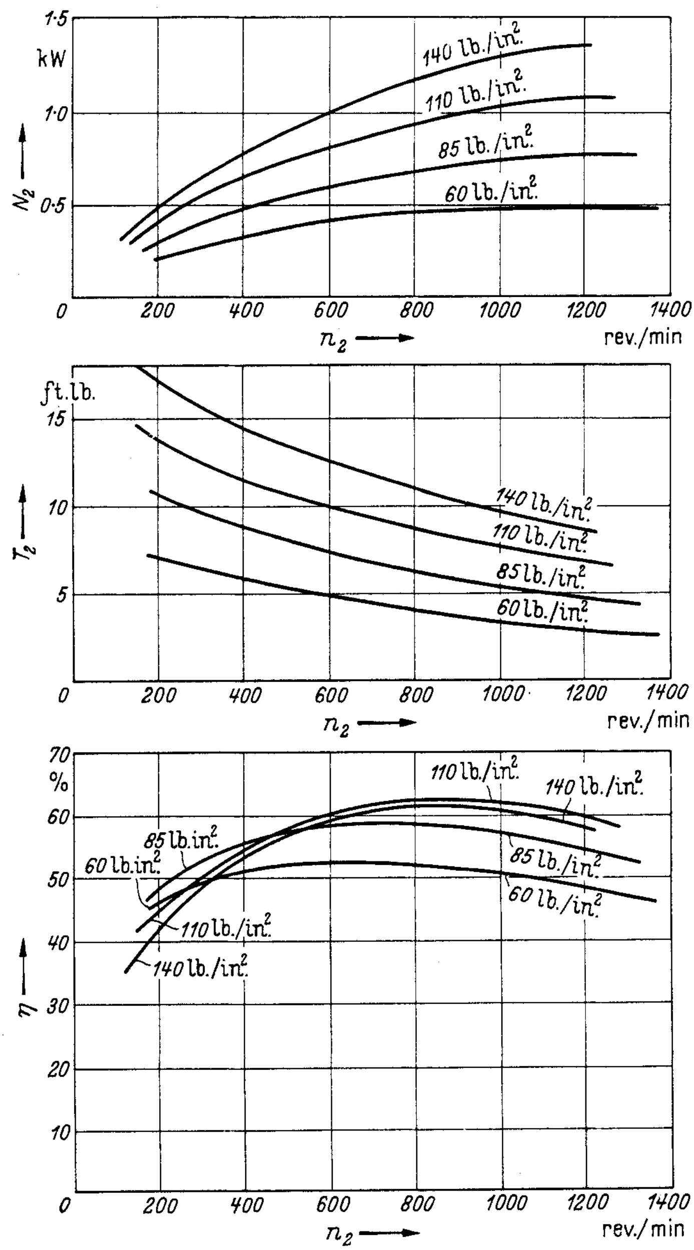


FIG. 188. (From G. SCHLESINGER, *Die Werkzeugmaschinen, loc. cit.*).

The duration of a rotational movement, once it is started in a certain direction, may be limited by the requirements of reversing this direction, but not by the design of the machine. In contrast to this a rectilinear movement is limited in stroke and duration by the dimensions of the machine parts. It must be possible, therefore, to start and stop these movements repeatedly and if necessary to reverse them.

When short-time movements are required, such as in the feed movement of planing and shaping machines, or the depth setting of grinding machines, it must be possible to stop after brief intervals and to re-start again in the same direction. If more continuous operations are required, e.g. in the cutting drive of planing and shaping machines, the movement must be reversed at the end of each working stroke and the moving part returned to its starting position, whereupon after again reversing the next working stroke can be started. With few exceptions (for instance, the feed movement during grinding) only one direction of a rectilinear movement represents an actual working stroke. In order to obtain the highest possible efficiency of the drive, it follows that the reversing action and the return stroke must be executed in the shortest possible time and at a minimum cost of energy. The driving devices must be so designed as to control and transmit the resulting inertia forces in such a manner that the operations are carried out smoothly and efficiently.

The mechanisms used for such drives can be separated into mechanical and hydraulic devices:

(a) *Mechanical devices* are usually driven and controlled by the rotating output shafts of gear-boxes and other mechanisms described in the previous chapter. The reversing action is either an

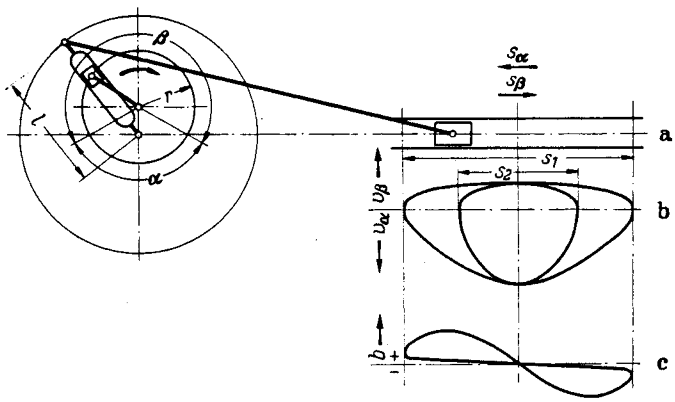


FIG. 189

inherent feature of the mechanism and occurs without reversing the driving shaft, or it is obtained by reversing the direction of rotation of the input shaft. Devices with positive reversing and uni-directional rotation of the input shaft are (i) crank drives (crank or eccentric with a connecting rod, quick-return mechanism, etc.), and (ii) cam-operated drives (cam and ram or cam and lever). If the input shaft on a crank drive rotates at uniform speed, the output speed, i.e. the speed of the reciprocating ram, is non-uniform. In the case of metal-cutting machine tools, simple crank drives are rarely used for operating the principal movements, because the forward and reverse velocities of the ram are approximately equal in each stroke, the forward and reverse strokes each taking place during approximately half a revolution of the crank shaft. In such cases, the time and energy losses during the return stroke are excessive. Crank drives are, however, used in the control mechanisms of some automatic lathes (see page 183). For the principal drives of machines with relatively short working strokes (e.g. the cutting drive for shaping and slotting machines) rotating or oscillating linkages (quick-return mechanisms) have been applied.

The Whitworth quick-return mechanism (Fig. 189a) is more efficient than the ordinary crank drive because the rotating crank, to which the ram is linked through the connecting rod, revolves with non-uniform angular velocity and thus evens out the velocity variation of the driven part. In other words, the velocity curve of the ram is flattened and more uniform than the sinusoidal one in the simple crank drive. The radius r of the driving crank and the angles covered by the rotating driving shaft during the forward ($= \beta$) and reverse ($= \alpha$) movement, are independent of the stroke s, which can be changed by varying the length l. The ratio v_β/v_α between forward and reverse speed remains, therefore, constant. Both the velocity curve (Fig. 189b) and the acceleration curve (Fig. 189c) have no discontinuities; the reversing action is, therefore, smooth. However, the actual speed of the rectilinear ram movement changes with varying ram stroke, even if the speed of the input shaft is constant. In order to obtain a required ram speed for any length of stroke, a speed change device for the driving shaft must be provided.

The length of stroke of the crank and slotted lever quick return mechanism (Fig. 190a) can be accurately adjusted by varying the radius of the driving crank. However, with decreasing stroke the angular ratio α/β and, therefore, the ratio between return and forward speeds, decreases; in other words, it becomes less favourable. Figure 190b shows the velocity curves for two strokes s_1 and s_2, the input speed of the driving shaft being set so that the maximum speed during the forward stroke is the same. The reversing action is again smooth (Fig. 190c).

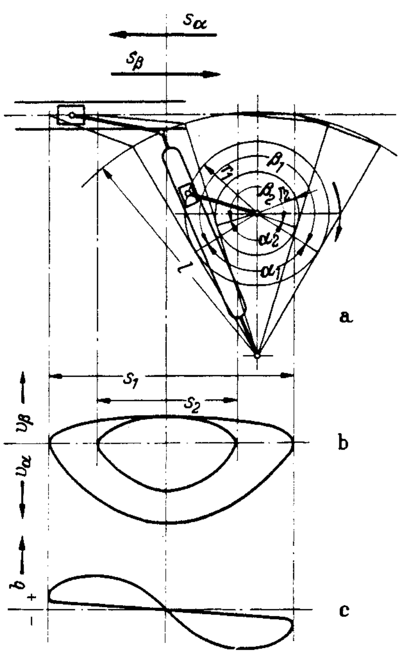


FIG. 190

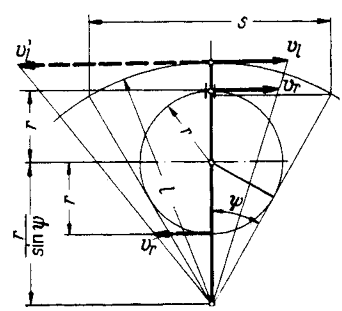


FIG. 191

The maximum forward speed v can be determined (Fig. 191) by assuming a sufficiently long connecting rod, resulting in $v_{\beta_{max}} = v_l$. In this case, the ratio between the ram speed and the circumferential velocity of the crank is:

$$\frac{v_l}{v_r} = \frac{l}{(r/\sin\psi) + r}$$

If the length of stroke is s:

$$\sin\psi = \frac{s}{2l}$$

and

$$\frac{v_l}{v_r} = \frac{l}{(2lr/s) + r} = \frac{ls}{2r(l + \frac{1}{2}s)}$$

$$v_l = v_r \times \frac{ls}{2r(l + \frac{1}{2}s)}$$

and with $v_r = 2\pi rn$ (where n = number of rev/min of the crank shaft)

$$v_l = \pi n \times \frac{ls}{(l + \frac{1}{2}s)}$$

The maximum return speed of the ram is (see page 138 and Fig. 191):

$$v_l' = v_r \times \frac{l}{(r/\sin\psi) - r}$$

$$v_l' = v_r \times \frac{ls}{2r(l - \frac{1}{2}s)}$$

and

$$\frac{v_l'}{v_l} = \frac{l + \frac{1}{2}s}{l - \frac{1}{2}s}$$

$$v_l' = v_l \times \frac{2l + s}{2l - s}$$

The rotational speed n of the crank corresponding to a ram speed v_l is:

$$n = v_l \times \frac{l + \frac{1}{2}s}{\pi ls}$$

$$= \frac{v_l}{2\pi} \times \frac{(2l + s)}{ls}$$

For a given length l of the slotted link and for a specified working range (maximum stroke s_{max}, minimum stroke s_{min}, maximum forward speed $v_{l_{max}}$ and minimum forward speed $v_{l_{min}}$):

$$n_{max} = \frac{v_{l_{max}}}{2\pi} \times \frac{(2l + s_{min})}{ls_{min}}$$

$$n_{min} = \frac{v_{l_{min}}}{2\pi} \times \frac{(2l + s_{max})}{ls_{max}}$$

The speed-range ratio is then:

$$B = \frac{n_{max}}{n_{min}} = \frac{v_{l_{max}}}{v_{l_{min}}} \times \frac{(2l + s_{min})}{(2l + s_{max})} \times \frac{s_{max}}{s_{min}}$$

Under the assumption that the ram speed is approximately constant over the full stroke, the net power at the ram is

$$N' = \frac{P \times v_l}{530\,000} \quad \text{(kW)}$$

(P = working resistance acting on the ram, lb, and v_l the ram speed, ft/min). Hence, if the mechanical efficiency is η, the driving power required is:

$$N = \frac{N'}{\eta} = \frac{P \times v_l}{530\,000\eta}$$

$$= \frac{P \times \pi \times n \times l \times s}{530\,000 \times \eta \times (l + \frac{1}{2}s)}$$

If the power of the drive is known the obtainable force at the ram can be calculated:

$$P = \frac{530\,000}{\pi} \times \eta \times \frac{N}{n} \times \left(\frac{1}{s} + \frac{1}{2l}\right)$$

For given values of the length l of the slotted link, driving power N and efficiency η, the force P, which the ram can exert, depends only upon the stroke of the ram and the rotating speed of the driving shaft.

If the friction forces arising at the crank pins, the shafts and the bearings and between the sliding part and the slotted link are neglected, the force P acting with link z on the lever can be reduced to a corresponding force P_{red} acting on the crank pin (distance x from the fulcrum of the oscillating lever):

$$P_{red} = P . \frac{l}{x}$$

It is possible to resolve the force P_{red} into two components, of which one P_K acts at right angles to the guiding surface of the slotted link (friction neglected), and the other P_L acts in the direction of this guiding surface.

P_K represents, therefore, the load on the crank pin, P_L the load on the bearing L for the slotted link. The latter is stressed in bending by P_K and in tension or compression by P_L. These relations are shown for four different positions (*1*, *2*, *3* and *4*) in Fig. 192. The force P_K is transmitted from

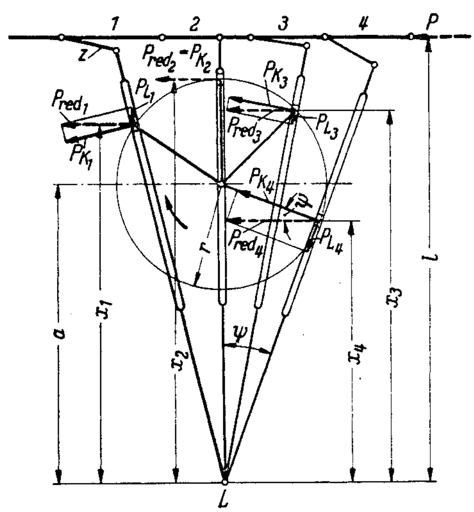


FIG. 192

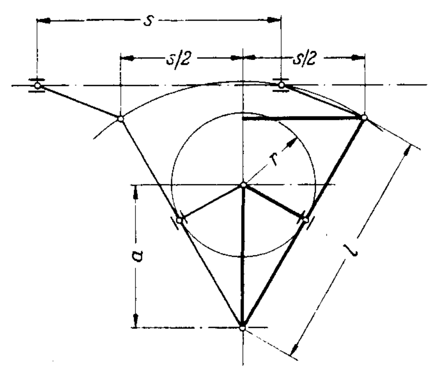


FIG. 193

the sliding part to the crank pin and can in turn be resolved into a radial force acting in the direction of the crank and a tangential force acting at 90° to the radial force. The tangential force determines the power at the crank and reaches its maximum in the centre (position *2*). At the reversing point (position *4*), it becomes zero. Component P_K reaches its minimum in the centre (position *2*) and its maximum at the reversing point (position *4*). In the latter position, P_K also represents the maximum radial force acting on the crank. Component P_L also reaches its maximum when at position *4*. It is, therefore, necessary to check the strength of the crank, the slotted link and the fulcrum bearing "L" for the loads P_{K_4} and P_{L_4}. These can be calculated as follows:

$$P_{K_4} = P_{red_4} \times \cos \psi$$

$$P_{red_4} = P \times \frac{l}{x_4} \quad \text{and} \quad \cos \psi = \frac{x_4}{\sqrt{(a^2 - r^2)}}$$

Hence

$$P_{K_4} = P \times \frac{l}{x_4} \times \frac{x_4}{\sqrt{(a^2 - r^2)}}$$

$$P_{K_4} = P \times \frac{l}{\sqrt{(a^2 - r^2)}}$$

In order to obtain the required stroke s, the crank radius r is adjustable. The relationship between s and r is (Fig. 193):

$$\frac{s/2}{r} = \frac{l}{a}$$

$$r = \frac{a \times s}{2l}$$

$$P_{K_4} = P \times \frac{l}{\sqrt{[a^2 - (a^2 s^2/4l^2)]}}$$

$$= P \times \frac{l}{a\sqrt{[1 - (s/2l)^2]}}$$

and

$$P_{L_4} = P_{K_4} \times \frac{r}{\sqrt{(a^2 - r^2)}}$$

But

$$P_{K_4} = P \times \frac{l}{\sqrt{(a^2 - r^2)}} \quad \text{and} \quad r = \frac{a \times s}{2l}$$

Hence

$$P_{L_4} = P \times \frac{s}{2a[1 - (s/2l)^2]}$$

If the working resistance P acting on the ram is constant, and the dimensions a and l are fixed by the design, the loads on the various elements of the machine increase with increasing stroke s (Fig. 194). The strength calculations have to be carried out, therefore, for the case of maximum stroke.

Apart from the static loads previously mentioned the dynamic forces, which can be determined as the products of mass and acceleration, have to be considered. The masses of the moving parts depend upon their shapes and the materials chosen, whilst the accelerations can be obtained from a diagram such as Fig. 190c.

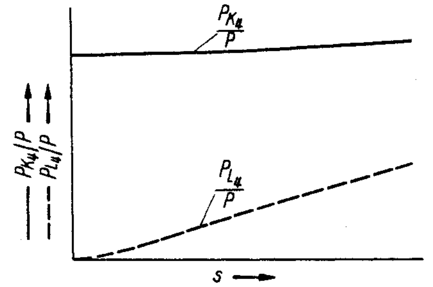


FIG. 194

Cam operated devices are used for setting and feed movements (relieving lathe, automatic lathe), and also for the operation of control elements. Their main advantage lies in the fact that the velocity of the driven element can be made completely independent of the design of the driving mechanism and that the shape of the curve can be designed to suit the required working conditions. This means, however, that for changing working conditions the shape of the curve must also be changed. In other words, it must be possible to remove and replace such cams, unless special devices are incorporated. If, e.g. the stroke and speed of the moving part can be varied by changing the length of levers or the speed of the cam, the working requirements can be satisfied without changing the cams.

The velocity and acceleration conditions of the cam-operated parts must be investigated and carefully checked by the designer and not only the working forces acting on the cam (friction and cutting forces), but also the accelerating forces have to be considered.*

A combination of a cam and rack is used in the drive of the turret slide for automatic lathes (Fig. 195) and a cam and crank linkage combination can be found in a small grinding machine (Fig. 196).

* Methods for calculating cam drives can be found in books concerned with theory of machines, kinematics, etc.

The following mechanical drives are usually employed when the required stroke of the ram is too long to be covered by the devices mentioned before, when the ram velocity has to be constant and when, moreover, the optimum values of forward and reverse speed must be obtained:

(i) Helical drives (screw and nut, worm and rack);

(ii) Pinion and rack, or wheel segment and rack.

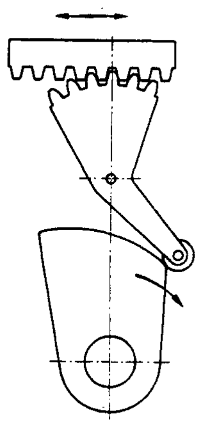


FIG. 195

When these devices are used, the driving shaft must be reversed when the movement has to be changed from forward to reverse or vice versa. The reversing process is, therefore, a separate operation and is not executed by the kinematics of the device employed. The conditions during the reversing action of the rotating input shaft have been studied in detail by Schlesinger* and the following considerations are taken mainly from his book.

The energy required for the reversing action can be determined by considering the conditions during the engagement of a clutch (Fig. 197). A pinion *1* drives a wheel *2* at constant angular velocity ω_2. This wheel is idling on shaft *I* and carries the female taper of a friction clutch *K*. The clutch member carrying the male taper slides on shaft *I* and can transmit the torque to shaft *I* via a key. Shaft *I* also carries a mass with the moment of inertia *J*. Before shaft *I* is coupled to the wheel *2* by means of the clutch it rotates at a velocity ω_1. The energy which must be transmitted by wheel *2* from the moment when the clutch becomes engaged and up to the moment

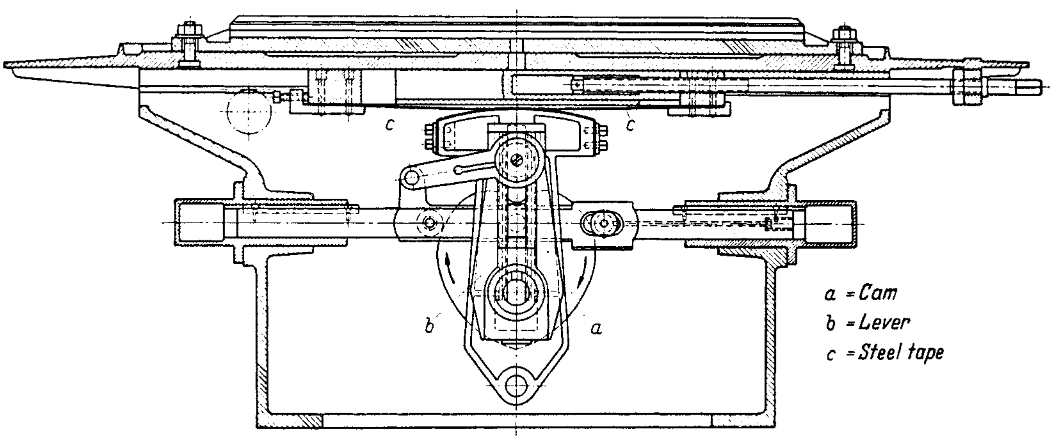


FIG. 196. Feed drive of a small grinding machine (Beling & Lübke, Berlin, 1935) (from G. SCHLESINGER, *Die Werkzeugmaschinen, loc. cit.*).

when shaft *I* has reached the angular velocity ω_2 of wheel *2* can be calculated. The clutch *K* transmits a torque *T* which serves for accelerating the mass with the moment of inertia *J*.

$$T = J \times \frac{d\omega}{dt} \tag{1}$$

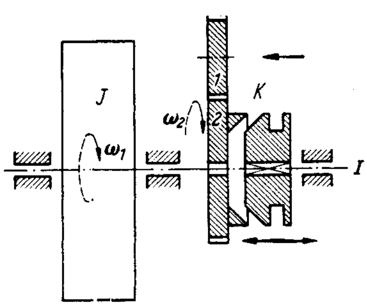


FIG. 197

If the wheel *2* maintains its angular velocity during the full period of clutch engagement, the drive, in order to exert the torque *T*, must transmit the power *N*.

$$N = T \times \omega_2 = J\omega_2 \times \frac{d\omega}{dt} \tag{2}$$

$$Ndt = J \times \omega_2 \times d\omega \tag{3}$$

$$\int Ndt = J\omega_2 \times \int_{\omega_1}^{\omega_2} d\omega \tag{4}$$

* *Die Werkzeugmaschinen*, Part 1, Ref. 10.

The product $N\,dt$ represents the work A which the wheel provides in order to change the angular velocity from ω_1 to ω_2.

$$A = J\omega_2 \times (\omega_2 - \omega_1) \tag{5}$$

This work consists of the kinetic energy W_k, and a remainder is used up as frictional heat A_W between the coupling faces of the clutch.

The kinetic energy of the masses rotating at an angular velocity ω_1 is

$$W_{k_1} = \frac{J}{2} \times \omega_1^2 \tag{6}$$

if the angular velocity is ω_2 the kinetic energy is

$$W_{k_2} = \frac{J}{2} \times \omega_2^2 \tag{7}$$

In other words, the accelerated masses have increased their kinetic energy by

$$W_k = W_{k_2} - W_{k_1} = \frac{J}{2}(\omega_2^2 - \omega_1^2) \tag{8}$$

The remainder of the work done is then transformed by friction into heat, and is

$$A_W = A - W_k$$

$$A_W = \frac{J}{2}(\omega_2 - \omega_1)^2 \tag{9}$$

The energy balance of the coupling process—on the credit side the work transmitted by the gear and on the debit side the frictional energy transformed into heat and the increase of kinetic energy—then becomes

$$A = A_W + W_k \tag{10}$$

$$J\omega_2 \times (\omega_2 - \omega_1) = \frac{J}{2}(\omega_2 - \omega_1)^2 + \frac{J}{2}(\omega_2^2 - \omega_1^2) \tag{11}$$

The distribution of the work in accordance with equations (5), (8) and (9) can be shown diagrammatically as follows. The three parts in the energy equation are proportional to the moment of inertia (J). The balance equation can, therefore, be established for $J = 1$:

$$\omega_2(\omega_2 - \omega_1) = \tfrac{1}{2}(\omega_2 - \omega_1)^2 + \tfrac{1}{2}(\omega_2^2 - \omega_1^2) \tag{12}$$

The energy introduced into the coupling process can, therefore, be represented by a rectangle, one side of which is of the length ω_2 and the other of the length $(\omega_2 - \omega_1)$, (Fig. 198). Draw the square $OCBD$, the sides of which are $OC = OD = \omega_2$, then, moving from the left-hand corner O to the right by an amount $\omega_1 = OE$, draw a line EF parallel to the vertical side of the square OD through the point E. The resulting rectangle $EFBC$ to the right is proportional to the work done

$$A = \omega_2(\omega_2 - \omega_1) \tag{5a}$$

By drawing the diagonal OB in Fig. 198 the work A is divided into the kinetic energy W_k and the frictional heat A_W. The lower trapezoid $EGBC$ represents the increase in kinetic energy W_k, being obtained by subtracting triangle

$$OEG = \frac{\omega_1^2}{2} = W_{k_1} \tag{6a}$$

from triangle

$$OCB = \frac{\omega_2^2}{2} = W_{k_2} \tag{7a}$$

The triangle GBF is a measure of the frictional heat, being half the square over the length $FB = (\omega_2 - \omega_1)$. It is, therefore:

$$A_w = \frac{(\omega_2 - \omega_1)^2}{2} \tag{10a}$$

Of particular importance is the process of engaging the clutch and starting the rotation of the shaft from zero speed. In this case the initial velocity of the shaft to be started is

$$\omega_1 = 0$$

and the energy balance of the coupling process from eqn. (11) is

$$J \times \omega_2^2 = \frac{J}{2} \times \omega_2^2 + \frac{J}{2} \times \omega_2^2 \tag{13}$$

In this equation, the right-hand side consists of two equal parts, in other words, the frictional energy is equal to the kinetic energy. This appears in the diagram as shown in Fig. 199. When a

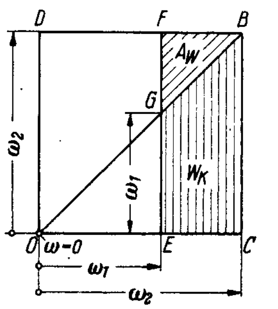


FIG. 198

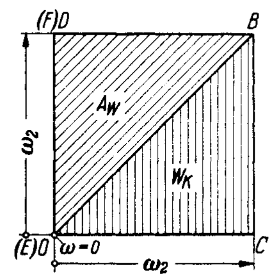


FIG. 199

stationary shaft is started by engaging a clutch, therefore, the frictional energy and the kinetic energy provided by the drive are equal. This fact is of general validity and is equally applicable to the starting of a motor by direct connexion to the mains supply. In this case the losses appear in the form of heat produced by the current in the rotor and its resistance.

If a shaft has to be reversed then ω_1 is negative, and the energy balance eqn. (11) reads as follows:

$$J\omega_2 \times (\omega_2 + \omega_1) = \frac{J}{2} \times (\omega_2 + \omega_1)^2 + \frac{J}{2} \times (\omega_2^2 - \omega_1^2) \tag{14}$$

The two parts of this equation which represent the kinetic energies are

$$W_{k_1} = \frac{J}{2} \times \omega_1^2 \tag{6}$$

(kinetic energy of the masses rotating in one direction)

$$W_{k_2} = \frac{J}{2} \times \omega_2^2 \tag{7}$$

(kinetic energy of the masses rotating in the opposite direction), so that eqn. (14) can be re-written as

$$A = A_W + W_{k_2} - W_{k_1}$$

$$A + W_{k_1} = A_W + W_{k_2} \tag{15}$$

During the reversing action the following amount of energy is thus supplied (left side of eqn. 15)

i. by the drive:

$$A = J \times \omega_2(\omega_2 + \omega_1) \tag{16}$$

ii. by the kinetic energy of the rotating masses prior to being reversed:

$$W_{k_1} = \frac{J}{2} \times \omega_1^2 \tag{6}$$

This energy is transformed (right-hand side of eqn. 15)

i. into frictional heat in the clutch

$$A_W = \frac{J}{2}(\omega_2 + \omega_1)^2 \tag{17}$$

ii. into kinetic energy of the masses rotating in the opposite direction after they have been reversed:

$$W_{k_2} = \frac{J}{2} \times \omega_2^2 \tag{7}$$

The two sides of the energy balance eqn. (15) can be represented by the areas shown above each other in the sketch (Fig. 200a). The lower area (left-hand side of eqn. 15) represents the energy A

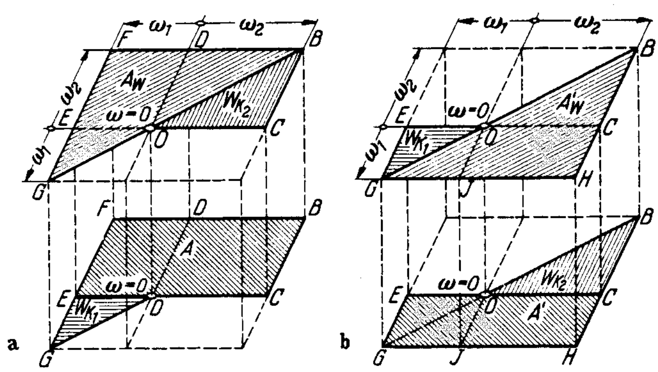


FIG. 200

supplied by the drive and the kinetic energy W_{k_1} provided by the rotating masses prior to being reversed. The upper area (right-hand side of eqn. 15) represents the heat energy A_W and the kinetic energy W_{k_2} of the rotating masses after being reversed.

The next sequence in the reversing action is a little different because the angular velocity prior to reversing is ω_2 (instead of ω_1) and after reversing ω_1 (instead of ω_2). The energy balance eqn. (14) is, therefore:

$$J\omega_1 \times (\omega_1 + \omega_2) = \frac{J}{2} \times (\omega_1 + \omega_2)^2 + \frac{J}{2} \times \omega_1^2 - \frac{J}{2} \times \omega_2^2 \tag{18}$$

$$A' = A'_W + W_{k_1} - W_{k_2}$$

$$A' + W_{k_2} = A'_W + W_{k_1} \tag{19}$$

The work A' which the drive provides for reversing is the only value which has not been contained in eqn. (14), because A'_W is equal to A_W.

It is again possible to draw a diagram (Fig. 200b). As in Fig. 200a, the lower area is for the left hand side of the equation, i.e. rectangle $EGHC$ representing the drive energy:

$$A' = J\omega_1 \times (\omega_1 + \omega_2) \tag{20}$$

and triangle OCB representing the kinetic energy W_{k_2} of the masses prior to reversing. The upper area represents the right hand side of the equation, triangle GHB being the heat energy A'_W lost in the clutch, and triangle GOE the kinetic energy W_{k_1} of the rotating masses.

If both reversing actions are considered as one complete cycle, the addition of the two eqns. (14) and (18) gives

$$J\omega_2(\omega_2 + \omega_1) = \frac{J}{2}(\omega_2 + \omega_1)^2 + \frac{J}{2}\omega_2^2 - \frac{J}{2}\omega_1^2 \tag{14}$$

$$J\omega_1(\omega_1 + \omega_2) = \frac{J}{2}(\omega_1 + \omega_2)^2 + \frac{J}{2}\omega_1^2 - \frac{J}{2}\omega_2^2 \tag{18}$$

$$\text{i.e.} \qquad A_{\text{total}} = J(\omega_1 + \omega_2)^2 \tag{21}$$

In other words, $A_{\text{total}} = A_{W_{\text{total}}}$ and $W_{k_{\text{total}}} = 0$!

This result shows that the total energy, which has to be provided by the drive during one complete reversing cycle from a forward rotation to the reverse rotation and back again, serves exclusively for making up the heat energy losses in the clutch.

The clutch is the only energy absorber. Theoretically the moving masses store the energy and return it, so that the energy absorbed by the clutch is provided in part by the drive, and in part by the stored kinetic energy of the rotating masses. Both are, however, eventually transformed into heat by the clutch.

This is particularly clear from a combination of Figs. 200a and 200b. The triangle OCB of the upper area in Fig. 200a represents the kinetic energy which is stored in the rotating masses. It is provided by the corresponding part of rectangle $ECBF$, representing the drive, in the lower part of the illustration. In Fig. 200b this triangle OCB representing the same kinetic energy appears in the lower area (energy supplied) and this provides part of the heat energy GHB in the upper area of this figure (energy absorption).

Similarly, triangle GOE represents the kinetic energy during rotation in the opposite direction. In Fig. 200b this triangle appears in the upper area, and is provided by the corresponding part of the drive energy $GHCE$ in the lower area. It also appears in the lower area of Fig. 200a, and provides the corresponding part of frictional heat energy GBF which appears in the upper area.

During each reversing process of the clutch the same amount of energy is transformed into heat. It is represented by triangle GBF in the upper area of Fig. 200a, or by the equal triangle GHB in the upper area of Fig. 200b.

The heat losses are, therefore, independent of whether the reversing action takes place from a high return speed to a low forward speed or vice versa.

This does not apply for the amounts of energy provided by the drive during the reversing action. For accelerating the masses from a fast return speed to a slow forward speed only a small amount of energy is required. In addition, the large amount of kinetic energy stored during the high speed return (Fig. 200b) is available, and only a small amount of energy need be provided by the drive. However, for reversing the masses from a low speed forward rotation to a high speed return rotation (Fig. 200a), more energy is required to accelerate the masses, and only a small amount of the kinetic energy is transformed into heat. For this reason, more energy must be provided by the drive.

Although, therefore, at each reversing action the clutch transforms the same amount of energy into heat, the drive must provide different amounts of energy at the end of each run, and the ratio between these two energy values is $\omega_2 : \omega_1$. This can also be seen by comparing the two rectangles $ECBF$ in Fig. 200a and $GHCE$ in Fig. 200b.

In Fig. 201 these relations are shown for a complete cycle of forward and reverse running, complete with accelerations and decelerations. After the previous explanation, the diagram should be easily understood. It must be read from the bottom upwards and comprises: starting (see Fig. 199)—return rotation—reversing from return to forward rotation (see Fig. 200b)—forward rotation—reversing from forward to return rotation (see Fig. 200a)—return rotation—again reversing and finally braking to a stop. During the braking process the kinetic energy is, of course, transformed into heat.

The previous considerations would appear to point to a simple means of reducing the energy required from the drive or the current supply, i.e. to reduce the heat losses, which are the only causes of energy losses.

They occur between the drive and the accelerated mass and are generated:

(a) in the clutch linings when clutches are used,

(b) by slipping of the belt on the pulley when reversing with belt drives, and

(c) in the resistances of the rotor when electric reversing is used.

In the equation which indicates the heat losses during the reversing action:

$$A_W = \frac{J}{2}(\omega_2 + \omega_1)^2 \tag{17}$$

the energy loss is proportional to the moment of inertia, J. The smaller the value of J, the lower is the energy required for reversing. The other parameter affecting the heat losses is the change in the angular velocity; its effect is greater as it is proportional to the square of the angular velocity. On the other hand, angular velocities should not be too low because at equal power the torque increases with decreasing angular velocity and the dimensions of the clutch become excessive if a high torque has to be transmitted. The usable range between maximum and minimum values of ω_1 and ω_2 is not very large, and there are, therefore, only two major possibilities of reducing the heat losses:

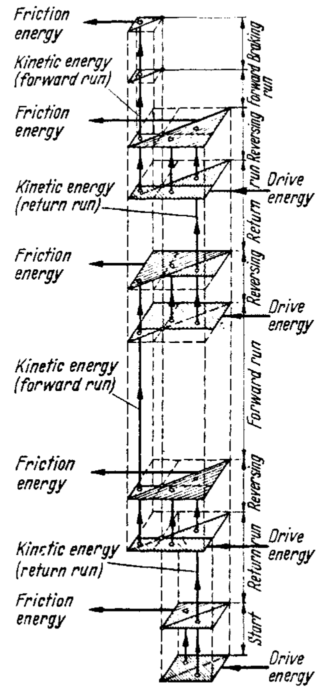


FIG. 201

1. During the deceleration of the masses the kinetic energy is not transformed into heat and, therefore, dissipated, but is stored and returned for acceleration in the opposite direction. There exist various mechanical and electrical methods which are based on this principle.

2. The acceleration process is controlled in such a manner that only the necessary amount of energy for accelerating the masses is drawn from the drive, whilst during the deceleration process, surplus kinetic energy is returned to the drive.

A study of the previous diagrams shows that less energy is required if the starting process is carried out in several steps, as the friction losses can be reduced (Fig. 202). If the starting action is carried out without intermediate steps from $\omega = 0$ to ω_{10} the kinetic energy and the heat energy would be equal, i.e.

$$A_W = W_k = \frac{J}{2} \times \omega_{10}^2$$

represented by the triangular areas OCB and ODB respectively. If, on the other hand, the movement is started by moving from zero to ω_1, from ω_1 to ω_2, from ω_2 to ω_3, etc., the resulting heat loss is only equal to the small crosshatched triangles (compare Fig. 202 with Figs. 198 and 199). The amount of kinetic energy is, of course, still represented by the triangle OCB, in other words, by the sum of the vertical trapezia, the bases of which are the increments from ω_1 to ω_{10}.

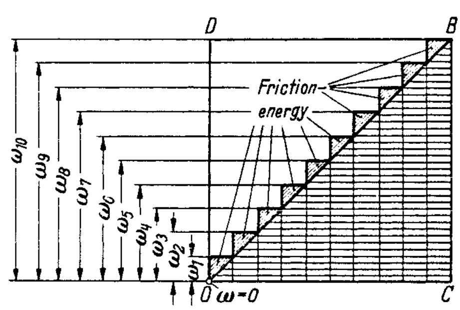


FIG. 202

The greatest saving of heat losses is obtained if the number of steps between zero and full speed approaches infinity, i.e. if a controlled infinitely variable speed increase is provided. In this case, the small triangles shown in Fig. 202 become infinitely small and their areas approach zero. In other words, no energy is required to make up for heat losses so that on reversing the rotating masses need not transform their energy into heat, but can return it to the drive. During a complete cycle, 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.

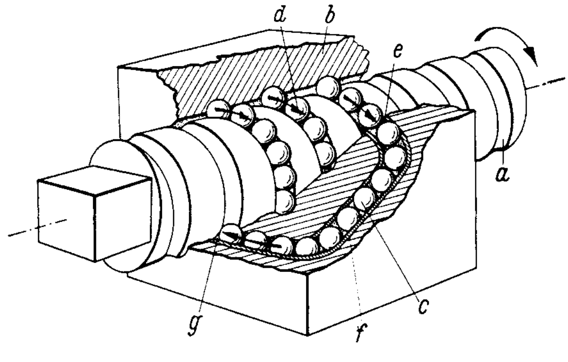


FIG. 203. Recirculating ball nut.

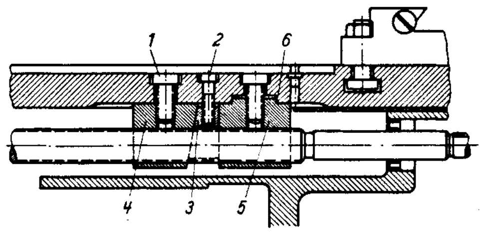


FIG. 204. Backlash elimination in the transverse feed of the Schaerer lathe (Industriewerke, Karlsruhe, Germany). After loosening screw *1*, the wedge *3* is drawn upwards by tightening screw *2*. This results in the movable portion of the nut *4* being pushed to the left. When the backlash is thus reduced 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.

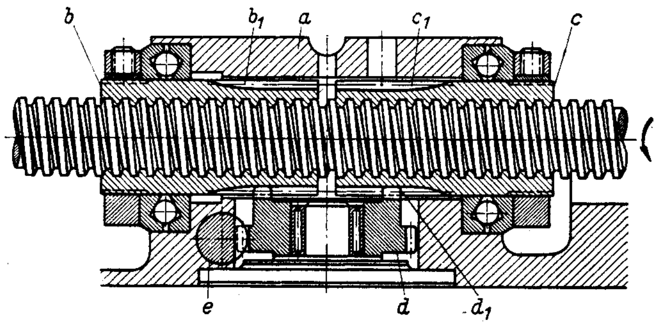


FIG. 205. Backlash eliminator in the feed drive nut of a milling machine table. (The Cincinnati Milling Machine Company, Cincinnati, Ohio, U.S.A.).

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.

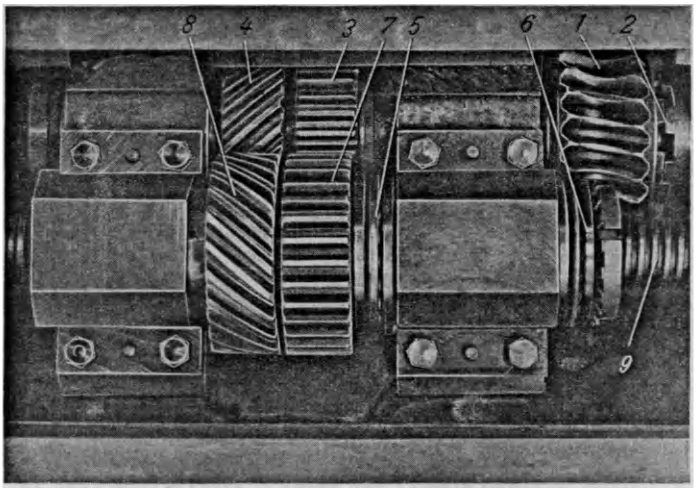


FIG. 206. Backlash eliminator in the feed drive nut for the table of the milling machine (Fig. 133).

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

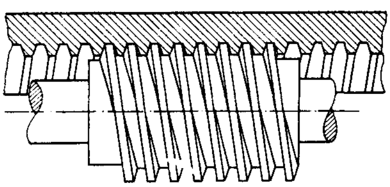


FIG. 207

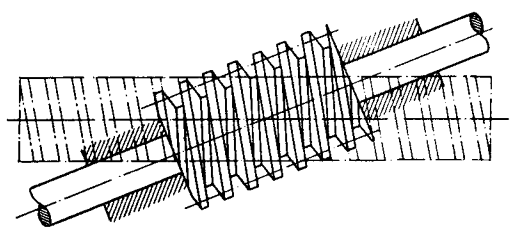


FIG. 208

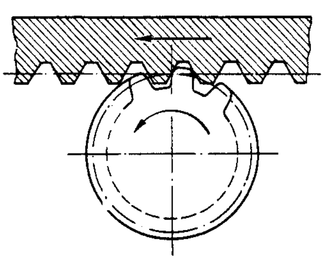


FIG. 209

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 preload 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.

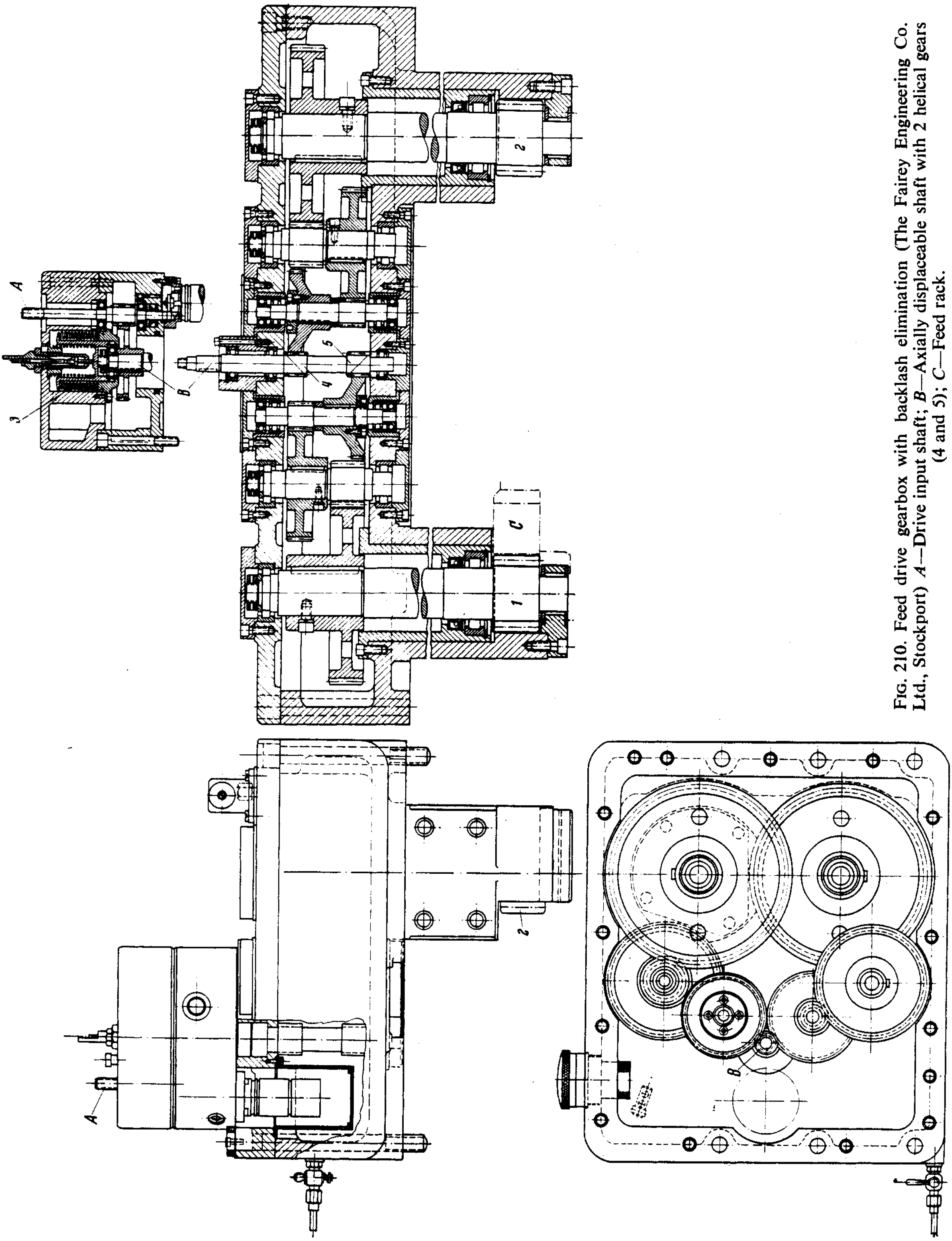


FIG. 210. Feed drive gearbox with backlash elimination (The Fairey Engineering Co. Ltd., Stockport) *A*—Drive input shaft; *B*—Axially displaceable shaft with 2 helical gears (4 and 5); *C*—Feed rack.

instance, rack–pinion–rack, see Fig. 212). It is also possible for a rotating hydraulic motor to drive a rectilinear movement through a mechanical intermediary (for instance, pinion and rack, or screw and nut if irreversibility is desirable, see page 148 and Fig. 211). Such an arrangement with infinitely variable speed may be superior to a purely mechanical drive, because the rotating masses which have to be reversed at the end of a working stroke are small, and smooth reversing with high acceleration is possible.

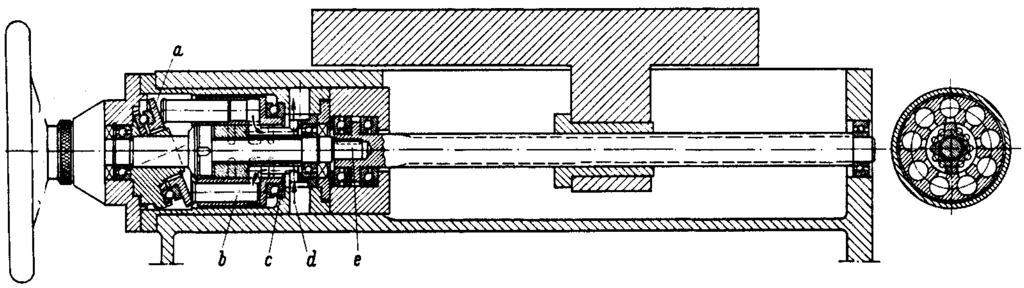


FIG. 211. Hydro-mechanical feed drive (hydraulic motor and feed screw) (Heller, Nürtingen, Germany) a—Swash plate; b—Nine pistons; c—Cylindrical distributor; d—Pressure oil inlet; e—Motor shaft.

Figure 212 shows various possibilities and the required minimum lengths of different arrangements. Arrangements 1 and 2 are relatively simple. If forward and reverse speed are to be equal, a variable oil flow (variable delivery pump or throttle adjustment) is necessary, as the piston rod reduces the effective piston area on one side. This difficulty is avoided in arrangements 3 and 4, where arrangement 4 results in a shorter overall length. The length of stroke is limited by the stiffness, the buckling strength and the sag of the piston rod. It can be increased by combining two piston drives in series together with an intermediate moving cylinder housing. The two-cylinder arrangement in one housing can act either in one direction (arrangement 6) or in two opposing directions (arrangement 5). A piston rod on both sides of the piston may be used in the design of arrangement 5.

If plungers are used instead of pistons, the surface finish of the internal cylinder walls is not critical. However, if forward and reverse strokes are to be driven by such an arrangement, two cylinders and two plungers must be provided (arrangement 7), unless a plunger is completely enclosed and pressure can be supplied to both sides. In this case, a transmission must be provided between the plunger and the driven slide, e.g. rack, pinion and rack (arrangement 9). In the case of piston and cylinder drives, an infinitely variable speed adjustment similar to the speed adjustment of hydraulic rotating motors (see page 134) is not possible because the size of the piston area cannot be infinitely varied. Infinite variation of the piston or slide speed can, therefore, be obtained only by throttling or by pump adjustment, i.e. by variation of the oil supply.

A stepped speed adjustment is, however, possible by arranging, for instance, a piston and a plunger in series (arrangement 8) and supplying either piston and plunger (minimum speed), piston alone (medium speed) or plunger alone (maximum speed). The possible length of stroke can also be increased by attaching to the piston rod a pinion which is driven along a fixed rack and in turn drives another rack fixed to the moving slide (arrangement 10). Pinion and rack are also used with an oscillating vane motor, the length of stroke depending upon the angle of oscillation α (arrangement 11). In a further arrangement the output shaft of a variable speed hydraulic motor carries a pinion which drives the rack of the moving slide (arrangement 12). The stroke of this arrangement is limited only by the length of the rack.

Let A_P be the area of the piston (diameter D), and the effective area on the piston rod side (outside diameter D, inside diameter equal to the diameter d of the piston rod) be A_{PR}. The oil volume Q_{2P} absorbed on the piston side at a velocity v_P is then

$$Q_{2P} = \frac{\pi}{4} \times D^2 \times v_P$$

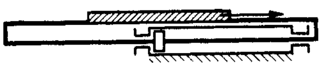
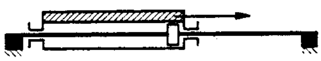
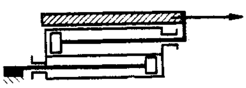
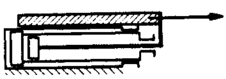
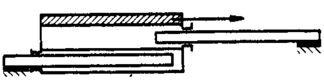
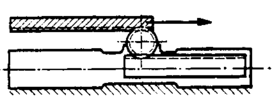
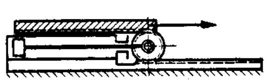
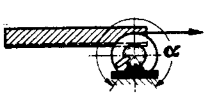
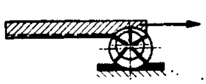
No.	Arrangement	Forward and reverse speed at constant pump delivery	Stroke of each piston	Length of table stroke L	Minimum length
1		different	H	H	$2L = 2H$
2		different	H	H	$2L = 2H$
3		equal	H	H	$3L = 3H$
4		equal	H	H	$2L = 2H$
5		different	H	$2H$	$1{\cdot}5L = 3H$
6		different	H	$2H$	$1{\cdot}5L = 3H$
7		equal	H	H	$2L = 2H$
8		different	H	H	$2L = 2H$
9		equal	H	H	$2L = 2H$
10		different	H	$2H$	$1{\cdot}5L = 3H$
11		equal			$2L$
12		equal			$2L$

FIG. 212

and on the piston rod side (velocity v_{PR})

$$Q_{2PR} = \frac{\pi}{4} \times (D^2 - d^2) \times v_{PR}$$

If the oil pressure is p, the force exerted on the piston is

$$P_P = \frac{\pi}{4} \times D^2 \times p$$

or

$$P_{PR} = \frac{\pi}{4} \times (D^2 - d^2) \times p \qquad \text{respectively.}$$

The power of the piston drive is, therefore

$$N_{2P} = P_P \times v_P = Q_{2P} \times p$$

$$N_{2PR} = P_{PR} \times v_{PR} = Q_{2PR} \times p$$

If the angle of oscillation of an oscillating vane (Fig. 213) is α radians (see No. 11 of Fig. 212), the volume of oil required for one oscillation from left to right, or vice versa, neglecting the thickness of the vane is

$$Q'_{2V} = \frac{\pi}{4} \times (D^2 - d^2) \times b \times \frac{\alpha}{2\pi}$$

$$= \tfrac{1}{8} \times (D^2 - d^2) \times b \times \alpha$$

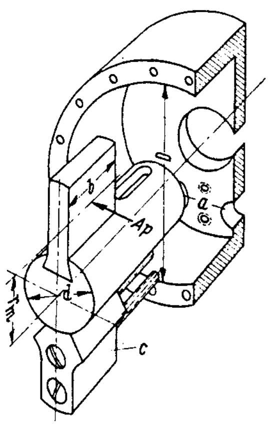


FIG. 213. Oscillating vane motor.
a—Oil inlet and outlet respectively.
c—Fixed dividing block between suction and pressure chamber.

For a complete oscillation (forward and reverse) the following oil volume is required

$$Q''_{2V} = 2Q'_{2V} = \tfrac{1}{4} \times (D^2 - d^2) \times b \times \alpha$$

and if the frequency of these oscillations is f

$$Q_{2V} = \frac{\alpha \times f}{4} \times (D^2 - d^2) \times b$$

The torque is

$$T_V = \left(\frac{D - d}{2}\right) \times b \times p \times r_m$$

But

$$r_m = \frac{D + d}{4}$$

and hence the torque is

$$T_V = \left(\frac{D^2 - d^2}{8}\right) \times b \times p$$

The required power, etc. can be determined as shown on page 132.

The control of the various movements can be obtained by adjustable pumps, valves, etc. which serve for the following purposes:

1. Variation of the oil pressure.
2. Variation of the oil flow.
3. Directional control of the oil flow.

The required characteristics of the equipment for adjustment and control depend upon their particular applications in different circuits.

The calculation and design of cylinders and pistons, valves, control devices, pipes, filters, etc. are described in detail in the relevant literature.[44,47] In the following paragraphs the basic considerations only concerning the design of the circuits will be discussed.[48]

The parameters are (Fig. 214):

A_1 = piston area (full area)
A_2 = effective piston area (piston rod end)
P = force acting on the piston rod
Q_1 = oil supply per minute (full area)
Q_2 = outgoing oil quantity (piston rod end)
R = resistance of the throttle valve
p_1 = oil pressure (entry)
p_2 = oil pressure (outlet)
p_d = pressure drop in the throttle valve
v = piston velocity.

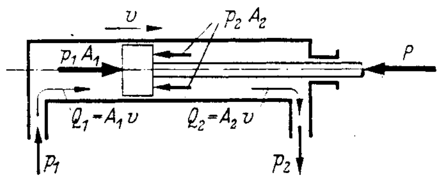


FIG. 214

In the following calculations the oil compressibility and any leakage losses are neglected.
If the piston moves in one direction (from left to right, see Fig. 214) then

$$P = p_1 \times A_1 - p_2 \times A_2$$

$$v = \frac{Q_1}{A_1} = \frac{Q_2}{A_2}$$

In certain cases, for instance, with circular saws, the cutting resistance (piston load P), must be kept constant by varying the feed rate in accordance with the changing cross section of the part (rolled profile) which has to be cut. In most cases, however, it is important for a required piston

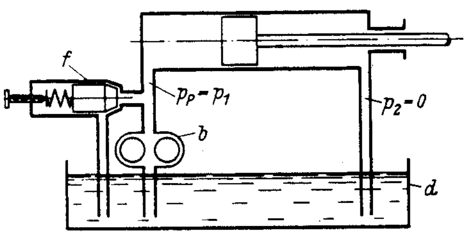


FIG. 215

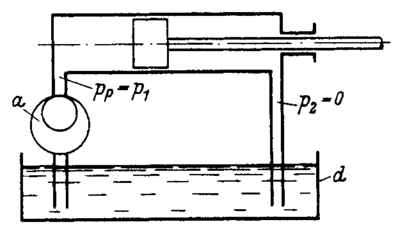


FIG. 216

velocity to be set as accurately as possible and not to vary even if the force acting on the piston rod changes. The conditions for a hydraulic circuit are, therefore, either:

1. P = constant, or
2. v = constant

In the following illustrations (Figs. 215 to 229) the following symbols have been used:

a = variable delivery pump
b = constant delivery pump
c = auxiliary pump
d = oil tank
e = throttle valve
f = overload valve
g = reduction valve
h = pressure valve
i = differential valve.

Figures 215 to 217 show simple "open" circuits without counter-pressure ($p_2 = 0$), in which

1. The pressure p_1 is produced by a constant delivery pump b and limited to a permissible maximum by an adjustable overload valve f (Fig. 215), or

2a. the delivery Q_1 can be varied by means of a variable delivery pump a (Fig. 216), or

2b. the oil volume delivered to a cylinder by a constant delivery pump b can be varied by means of a throttle valve e (Fig. 217). The relation between pressure drop and oil delivery is

$$p_d = Q^n \times R$$

where R is the throttling resistance and n a coefficient lying between 1 and 2.

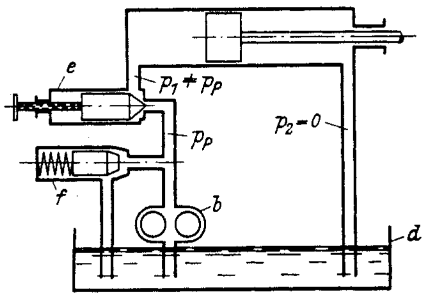


FIG. 217

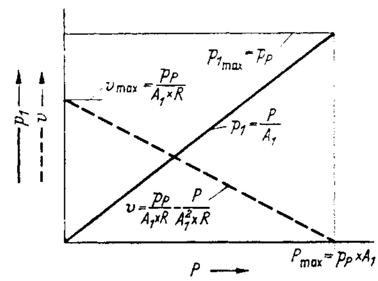


FIG. 218

For reasons of simplicity, constant temperature and constant oil viscosity may be assumed. In this case R can be considered constant for a given setting of the throttle valve. If, in addition, and again for the purpose of simplicity, laminar flow is assumed, then $n = 1$ and $p_d = Q \times R$.

The oil volume Q_1 which flows through the throttle valve at a given setting depends upon the pressure drop $p_d = p_P - p_1$

$$Q_1 = \frac{p_d}{R} = \frac{p_P - p_1}{R}$$

Moreover

$$p_1 = \frac{P}{A_1}$$

and

$$v = \frac{Q_1}{A_1}$$

Hence

$$v = \frac{p_P - p_1}{A_1 \times R}$$

$$= \frac{p_P}{A_1 \times R} - \frac{P}{A_1^2 \times R}$$

At a given setting of the throttle valve in the arrangement (Fig. 217), the piston velocity is not independent of the load. It grows with decreasing pressure p_1, i.e. with decreasing resistance P (Fig. 218).

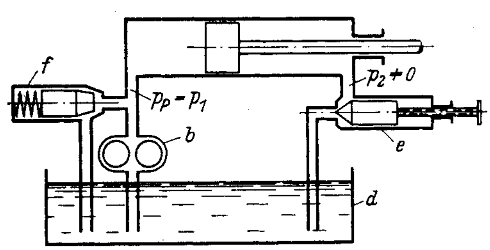


FIG. 219

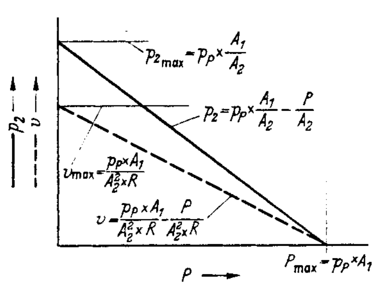


FIG. 220

In the previous arrangements, the counter-pressure p_2 is zero. If, therefore, the working resistance suddenly drops, the piston will jerk forward, because p_1 decreases with decreasing P. The oil will then slightly expand, and the leakage losses, which are proportional to the pressure, will also decrease. In most machine tool drives, the working resistance may be subjected to heavy pulsations and for this reason circuits in which the piston is not held by a counter-pressure p_2 are unsuitable. A simple means for producing a counter-pressure p_2 is a throttle valve e in the oil return pipe (Fig. 219). In this arrangement:

$$v = \frac{Q_2}{A_2}; \qquad Q_2 = \frac{p_2}{R}; \qquad v = \frac{p_2}{A_2 \times R}$$

$$P = p_1 \times A_1 - p_2 \times A_2 = p_P \times A_1 - p_2 \times A_2$$

$$p_2 = p_P \times \frac{A_1}{A_2} - \frac{P}{A_2}$$

$$v = \frac{p_P \times A_1}{A_2^2 \times R} - \frac{P}{A_2^2 \times R}$$

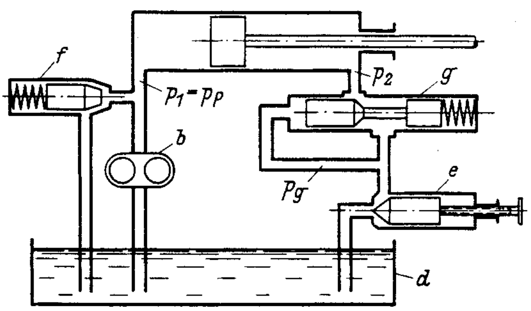


FIG. 221

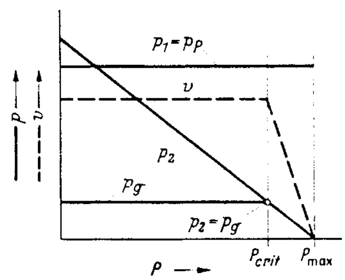


FIG. 222

Although the counter-pressure p_2 increases with decreasing force P, the velocity is again not independent of the load and increases with decreasing resistance (Fig. 220).

For both the above cases (Figs. 217 and 219), some of the oil supplied by the pump is returned to the tank via the overload valve f, without having done any useful work, except in the case of the maximum piston velocity. The maximum pressure p_P in the supply line between pump b and throttle valve e (Fig. 217) is limited by the overload valve f. The pressure p_P and the power of the pump remain, therefore, approximately constant and independent of the piston velocity. If the piston velocity is low (small output power), such an arrangement has a very low hydraulic efficiency, viz.:

$$\eta = \frac{P \times v}{p_P \times Q_P}$$

p_P is the pressure, Q_P the oil delivery of the pump.

It is possible to make the piston velocity v much more independent of the working resistance P if a reduction valve g (Fig. 221) is provided in the return pipe. This valve makes the pressure drop in the throttle valve e independent of p_2, and, therefore, independent of the working resistance P. Valve g ensures a constant pressure p_g before the throttle valve and, therefore, also a constant pressure drop, until above a critical working resistance P_{crit}, the pressure $p_2 < p_g$ (Fig. 222).

With

$$Q_2 = \frac{p_g}{R} \quad \text{and} \quad v = \frac{p_g}{A_2 \times R}$$

v will be constant as long as p_g is constant.

The efficiency of the circuit (Fig. 221) is again low for small piston velocities and low working resistance, because the power of the pump $p_P \times Q_P$ is constant and independent of the working resistance and the piston velocity. In the arrangement (Fig. 223), the counter-pressure p_2, which grows with decreasing resistance P, opens a pressure valve h in the oil supply line against the action of a compression spring, so that with growing working resistance P, the pressure valve h is closed, and the pressure $p_1 = p_P$ increases in proportion to the working resistance P. In other words,

$$\frac{P}{p_P} = \text{constant},$$

and the efficiency is:

$$\eta = \frac{P}{p_P} \times \frac{v}{Q_P} = \frac{\text{const.}}{Q_P} \times v$$

As Q_P is constant, the efficiency is proportional to the piston velocity until P reaches a critical value P_{crit}. This value is determined by the maximum pressure p_f which is limited by the overload valve f (Fig. 224).

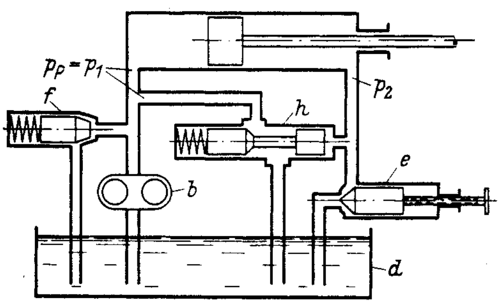


FIG. 223

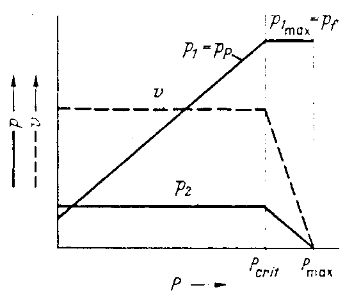


FIG. 224

If the counter-pressure p_2 must be high enough to hold the piston even if the direction of the working resistance suddenly changes, a differential valve i can replace the overload valve f (Fig. 225). The differential valve is opened by the combined pressures $p_1 + p_2$, against the action of a compression spring, and it controls the working pressure p_1. The smaller the working resistance P, i.e. the higher the counter-pressure p_2, the wider will valve i_1 be opened, and the pressure p_1 will drop. The working pressure p_1 acts on area i_1 and the counter-pressure p_2 on the annular area i_2. The force exerted by the spring is S. The equilibrium condition is, therefore:

$$p_1 \times i_1 + p_2 \times i_2 = S^*$$

$$p_1 = \frac{S}{i_1} - \frac{i_2}{i_1} \times p_2$$

The equilibrium condition in the cylinder is:

$$p_1 \times A_1 - p_2 \times A_2 = P$$

Hence

$$p_1 = \frac{S}{i_2(i_1/i_2 + A_1/A_2)} + \frac{P}{A_2(i_1/i_2 + A_1/A_2)}$$

and

$$p_2 = \frac{S}{i_1(i_2/i_1 + A_2/A_1)} - \frac{P}{A_1(i_2/i_1 + A_2/A_1)}$$

These relations are shown in Fig. 226.

* The actual amount by which the spring is compressed is small and the characteristic of the spring very flat. For this reason S is assumed to be constant.

Instead of using compression springs it is also possible to use the oil pressures themselves for controlling the valves. The so-called "balancing valves" which result from this have been described in detail by Dall.[48]

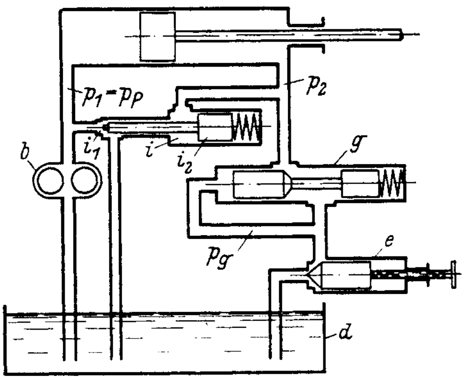


FIG. 225

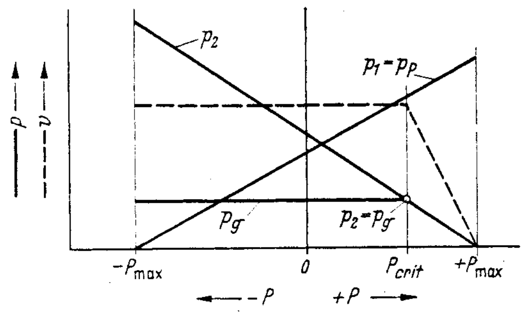


FIG. 226

Circuits employing constant delivery pumps and throttle valves are simpler and cheaper than those in which variable delivery pumps are used. However, with the latter higher efficiencies can be obtained.

It is possible to use a throttle valve in the return pipe (Fig. 227) and thus avoid the disadvantage of the circuit shown in Fig. 216 where $p_2 = 0$, the piston jumping forward if the working resistance should suddenly drop (see page 157). If an overload valve is also provided care must be taken to prevent the throttling effect raising the counter-pressure p_2 too much because the overload valve would open when $p_1 = p_P$ and the piston velocity would vary independently of the delivery adjustment of the pump.

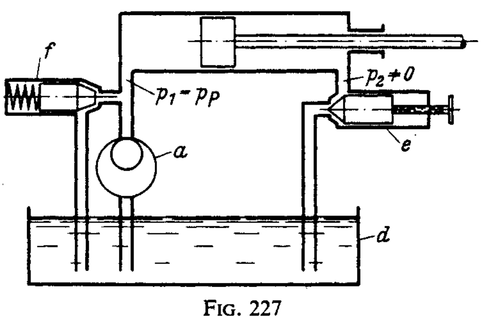


FIG. 227

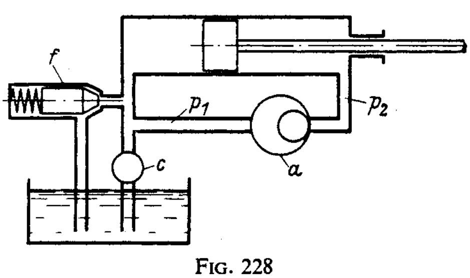


FIG. 228

In this arrangement, the necessary pump pressure p_P must be raised due to the additional counter-pressure p_2, and this has an unfavourable effect on the efficiency (see page 157).

The full benefit of variable delivery pumps can be obtained in "closed" circuits, in which the oil tank is unnecessary from a theoretical point of view (Fig. 228). In view of the unavoidable leakage losses and, in the case of a one-sided piston rod, it is, however, necessary to provide an auxiliary pump c and an overload valve f. These serve for compensating the differences in oil volume on the two sides of the piston, and prevent the entry of air into the circuit by a generous supply of surplus oil (see page 126).

Through the overload valve f in Fig. 228 the pressure p_1 is kept constant, and the equilibrium equation of the piston becomes

$$P + p_2 \times A_2 = p_1 \times A_1$$

$$p_2 = p_1 \times \frac{A_1}{A_2} - \frac{P}{A_2}$$

The counter-pressure p_2 drops with increasing working resistance P. The volume of oil which the piston pushes out of the cylinder during its movement is controlled, however, only by the volume which the pump a can absorb, and the piston can only jump forward by a small amount which is determined by the compressibility of the oil. In this respect the arrangement is, therefore, superior to the arrangement shown in Fig. 216.

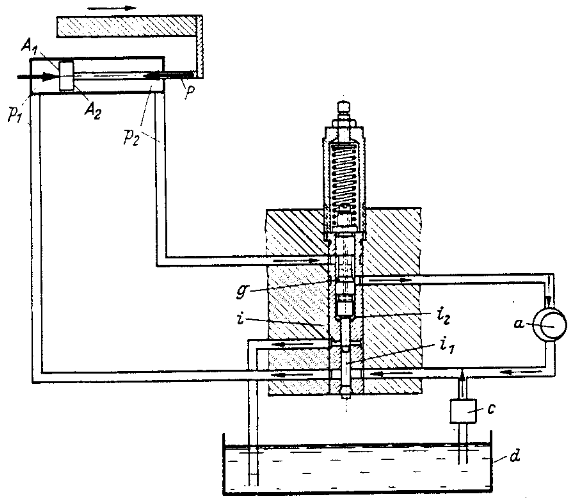


FIG. 229. Schematic layout of the "Cincinnati" differential valve (from G. SCHLESINGER, *Die Werkzeug maschinen, loc. cit.*).

It will be appreciated that exactly constant velocity is obtainable only if the oil flow Q_2 and the counter-pressure p_2 at the outlet side of the cylinder remain constant and independent of any variations of the force P. Only then will the compressibility of the oil have no effect upon the working of the piston drive.

When a variable delivery pump is used in a closed circuit the outgoing oil flow Q_2 is kept constant by the pump. The use of the "Cincinnati" differential valve is shown in Fig. 229 and this makes it possible to satisfy approximately the second condition (p_2 = constant) as well.

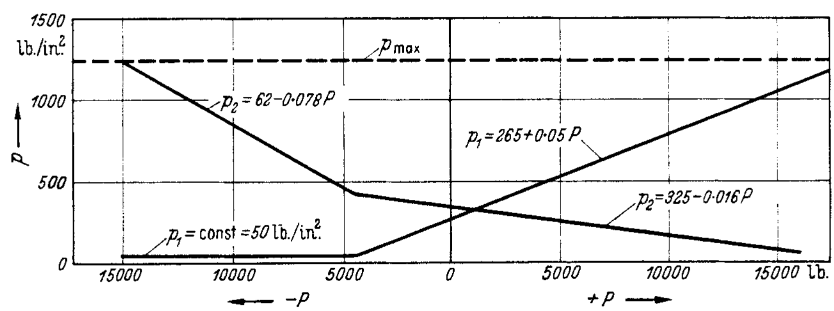


FIG. 230

The differential valve is a combination of the two valves i and g shown in Fig. 225. The pressure p_1 is controlled as a function of p_2 and not immediately as a function of the working resistance P. The relation is as follows:

$$p_1 = \frac{S}{i_1} - \frac{i_2}{i_1} \times p_2 \qquad \text{(See page 158).}$$

It is impossible, therefore, from a theoretical point of view to keep p_2 exactly constant. However, the sensitivity of the control increases with the ratio i_2/i_1. The values of p_1 and p_2 as functions of P are obtained from the equations shown on page 158:

$$p_1 = \frac{S}{i_2(i_1/i_2 + A_1/A_2)} + \frac{P}{A_2(i_1/i_2 + A_1/A_2)}$$

$$p_2 = \frac{S}{i_1(i_2/i_1 + A_2/A_1)} - \frac{P}{A_1(i_2/i_1 + A_2/A_1)}$$

these relations being shown in Fig. 230 for the example of a layout in which:

$$i_1 = 0{\cdot}03 \text{ in}^2$$
$$i_2 = 0{\cdot}09 \text{ in}^2$$
$$A_1 = 16 \text{ in}^2$$
$$A_2 = 13 \text{ in}^2$$
$$S = 37 \text{ lb}$$

The value of p_1 is, therefore:

$$p_1 = \frac{37}{0{\cdot}09 \times (0{\cdot}03/0{\cdot}09 + 16/13)} + \frac{P}{13 \times (0{\cdot}03/0{\cdot}09 + 16/13)}$$
$$= 265 + 0{\cdot}05P$$

and of p_2

$$p_2 = \frac{37}{0{\cdot}03 \times (0{\cdot}09/0{\cdot}03 + 13/16)} - \frac{P}{16 \times (0{\cdot}09/0{\cdot}03 + 13/16)}$$
$$= 325 - 0{\cdot}016P$$

In view of the throttling effect of valve i, the pressure p_1 cannot drop below a certain value, even if P becomes negative. In the example quoted, this minimum of $p_1 = 50$ lb/in² is reached when $265 + 0{\cdot}05P = 50$

$$P = -4300 \text{ lb}$$

Below this value, p_2 becomes, therefore:

$$p_2 = 50 \times \frac{16}{13} - \frac{P}{13} = 62 - 0{\cdot}078P$$

The foregoing considerations are all concerned with static loading conditions. Whilst, however, the kinetic energy of the oil may be neglected (see page 125), the stiffness of the layout (pipelines, cylinder, piston rods, oil column, valve springs, etc.) and its vibration behaviour cannot be neglected under conditions of dynamic loading, especially if it is possible for resonance to occur.

AUTOMATIC CONTROL[49]

1. PRINCIPLES AND CONSTRUCTIONAL ELEMENTS

The terms "automatic operation" and "automatic control" have been used for many years in connexion with various machines and mechanisms, where the functions of an operator are replaced by electrical, mechanical or hydraulic devices. These functions can be divided into the following groups:

(a) Driving the:
 (i) Cutting movements
 (ii) Feed movements
 (iii) Return movements (where required and, if necessary, at increased speed).
(b) Control of movements:
 (i) Starting
 (ii) Stopping
 (iii) Reversing.
(c) Clamping and unclamping the workpiece.

(d) Selection of the required speeds, setting of tools and starting and stopping the movements for different operations in accordance with a planned sequence.

(e) Measurement of the machined shapes and surfaces and rejection of workpieces whose dimensions lie outside the permissible limits or, alternatively, stopping of the machine if faulty workpieces are produced.

(f) Measurement of the machined shapes and surfaces and correction of the tool settings if the dimensions produced differ from those required.

(g) Checking and continuously correcting (if necessary), the workpiece or tool carrier positions during the operation.

(h) Transport of components from one machine to the next in accordance with the requirements of the machining programme (linking of machines, transfer).

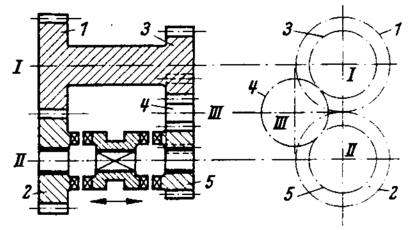


FIG. 231

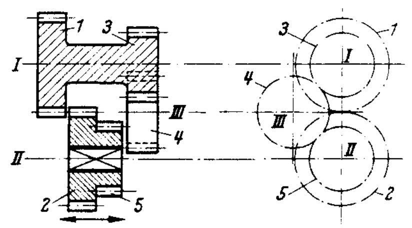


FIG. 232

The *mechanization of drives* for cutting, feed and return movements, which has been discussed in the preceding chapters, cannot really be considered as automatic control in the generally accepted sense. The idea of "automation" is usually applied when the presence of the operator is unnecessary not only between two subsequent operational steps, but also during a complete sequence of operations. In the case of a single machine, this would cover functions (b)(iii) to (g). On the other hand, functions (b)(i) and (b)(ii) may also be covered by automatic devices. A discussion of group (h), which would be applicable in the case of a complete production unit consisting of several machines, is considered to lie outside the scope of this book.

Electrical and hydraulic controls have been discussed earlier (see page 95). Among the mechanical means which serve for connecting or disconnecting a driving shaft (electric motor, reduction gear, hydraulic motor, etc.) and the gearbox of a machine tool, electromagnetic, mechanical or hydraulic clutches are used. Belts shifted from an idling to an operating pulley and vice versa are not often employed at the present time. Open and crossed belts or gears which drive either directly or through an intermediate gear and are operated by means of clutches (Fig. 231), sliding (Fig. 232) or radially engaged gears (Figs. 233a and 233b) are used for reversing the direction of movement.

Whilst the two latter should only be operated when the machine is stopped, belt drives and clutch devices can be operated when the machine is in motion, sometimes even though the machine is still under load when the reversing operation is started. No difficulties are encountered when friction or electromagnetic clutches are employed. Dog clutches are best engaged or disengaged when the drive is either stopped or at least running out. In their use it is necessary for the disengaging movement to be carried out at the maximum possible speed, since otherwise the disengaging operation results in excessive flank pressure, due to the load transmitting contact surfaces of the dog teeth becoming progressively smaller as the disengaging movement proceeds.

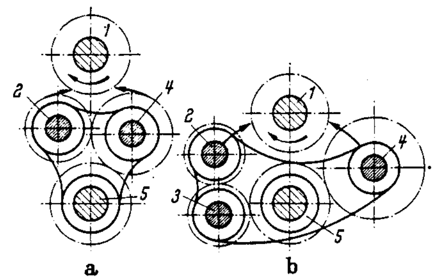


Fig.	Driving gear	Driven gear	Intermediate gears	
			For clockwise rotation of gear wheel 5	For anticlockwise rotation of gear wheel 5
a	1	5	4	2—4
b	1	5	4	2—3

FIG. 233. Two arrangements for reversing the drive between main spindle (gear *1*) and slip gear shaft (gear wheel *5*) for the feed drive of a centre lathe. a—Four wheel arrangement (danger of the toggle effect dragging the carrier between the gears *1* and *5* and breaking; b—Five wheel arrangement with radial movement of the intermediate gear towards the driving gear.

The teeth of the dog clutches can be double acting (Fig. 234a) or single acting (Fig. 234b). They must be designed in such a manner that the friction forces between the moving clutch member and the shaft and between the clutch teeth flanks, are smaller than the axial force available for carrying out the disengaging operation. For the sake of economic manufacture, it is advantageous to design dog clutches with odd numbers of teeth, because a minimum number of milling passes (equal to the number of teeth) (Fig. 234c) is then required.

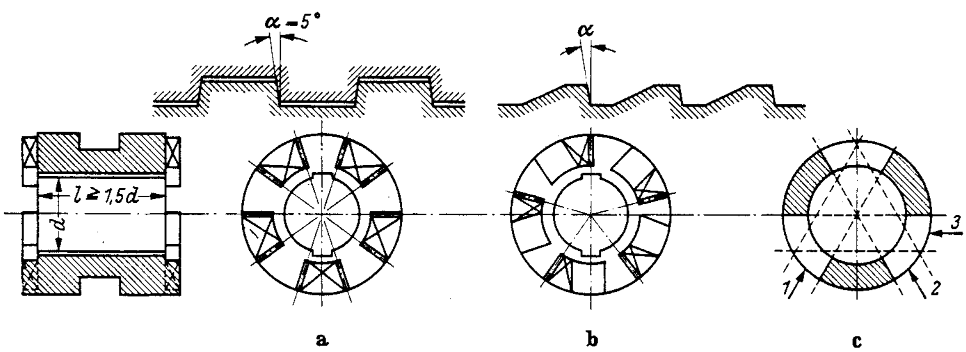


FIG. 234

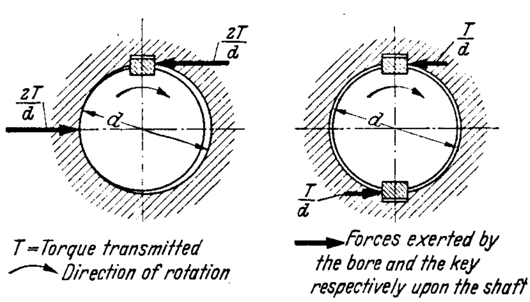


FIG. 235

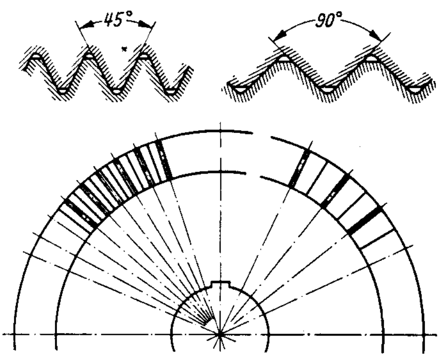


FIG. 236

The length of the sliding part should be approximately 1·5 times the bore diameter (see Fig. 234), in order to avoid binding. The same purpose is served by providing at least two symmetrically arranged feather keys or by using splined shafts. Both these arrangements prevent binding and reduce the torque transmitting forces between the sliding member and the shaft (Fig. 235). If the tooth flanks are slightly inclined (about 5° to 10°), the engagement of the clutch is easier and the force required for disengaging it reduced (see Fig. 234). The greater this angle, the easier is the engagement. This consideration leads to the arrangement of Fig. 236, which reduces the time required for engaging the clutch at a given rotational speed. However, in this case, some provision must be made for locking the clutch in position, as it would otherwise be disengaged by the axial component of the force transmitted. This locking can be obtained by toggle action or by an irreversible cam which shifts the movable clutch member and locks it in its extreme position. Such an arrangement is shown in Fig. 237, the example being that of a hand-operated clutch. This type of

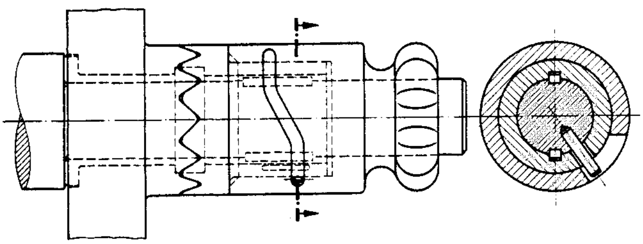


FIG. 237

clutch can also be used as an adjustable overload protection, the spring force which keeps the clutch engaged being adjustable to suit the maximum torque that is to be transmitted (Fig. 238). When

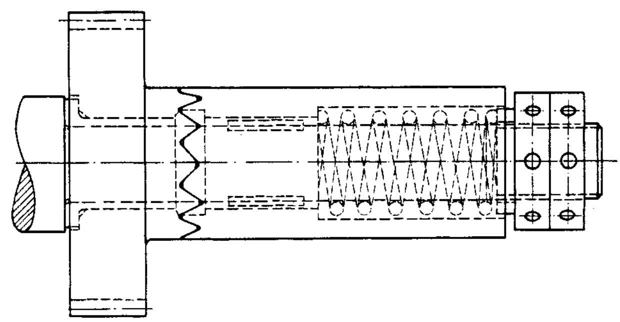


FIG. 238

the required torque, and, therefore, the corresponding axial component is exceeded, the spring will allow the clutch to slip.

For *locating and clamping the workpiece*, levers *a*, wedges *b* or jaws *c* are used as shown in Fig. 239, the clamping surface *x* pressing the workpiece *y* against the fixed locating face *z*. Concentricity can be ensured by making the clamping elements move concentrically and radially against the cylindrical external (*d*) or internal (*e*) surfaces of the workpiece *y*. In one particular application ball bearing races had to be ground accurately on an internal grinding machine, and it was important to avoid radial force components producing undesirable deformations of the workpiece. For this purpose, an oil-immersed disc was used to hold the workpiece axially against the face plate.[50] For workpieces of ferrous materials, magnetic clamping devices may be used, for instance, on grinding machines.

The shapes and dimensions of the workpiece may vary, of course, within the permissible limits and positive clamping is, therefore, difficult, if not impossible. For this reason, it is necessary to ensure that the clamping forces which act on the workpiece are of the required magnitude and remain constant, even if the energy transmission of the clamping device fails during the cutting operation.

The clamping mechanisms are, therefore, usually irreversible (cams and clamping levers, clamping cams or eccentrics, clamping screws). With the exception of the clamping screw, such devices have a constant working stroke and the variations in the dimensions of the workpiece, which must, of course, lie within the manufacturing limits, have to be covered by an elastic intermediate member.

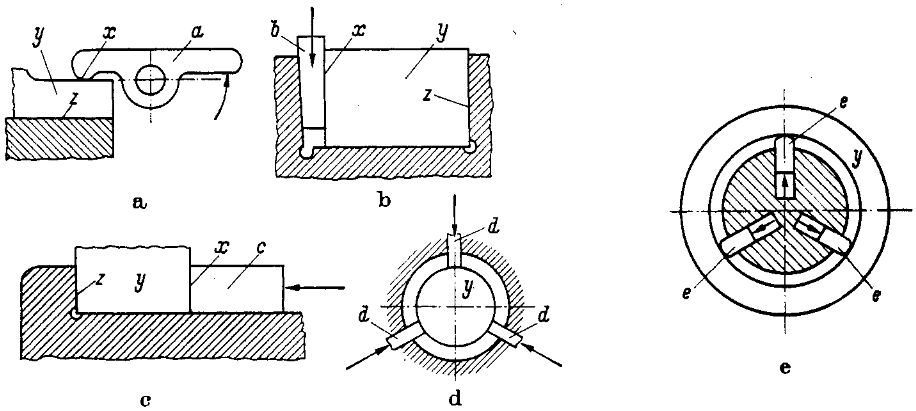


FIG. 239

The "spring constant" of this member must be so chosen that the required clamping force is maintained, whatever the actual dimensions of the workpiece.

The elastic member can be arranged between the driving (*1*) and the clamping (*2*) element (Fig. 240), or it can be in the drive itself, for instance, when a jaw is moved by a clamping screw (vice) driven by a sliding clutch (see Fig. 238), which controls the torque on the screw in accordance with the required clamping force.

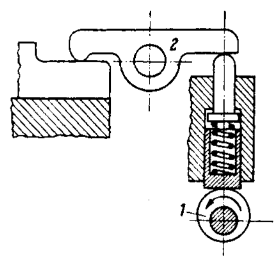


FIG. 240

In the case of turning operations, the clamping elements have to locate the workpieces concentrically, and prevent rotational or axial displacements (see Figs. 239d and 239e). In automatic lathes, the jaws are usually displaced radially by an axial movement of a taper a_1 in a hood b (Fig. 241). The axial displacement is produced by a tube g which pushes (Fig. 241a) or draws (Fig. 241b) the collet a into the hood b. In both cases, there is a possibility of the workpiece

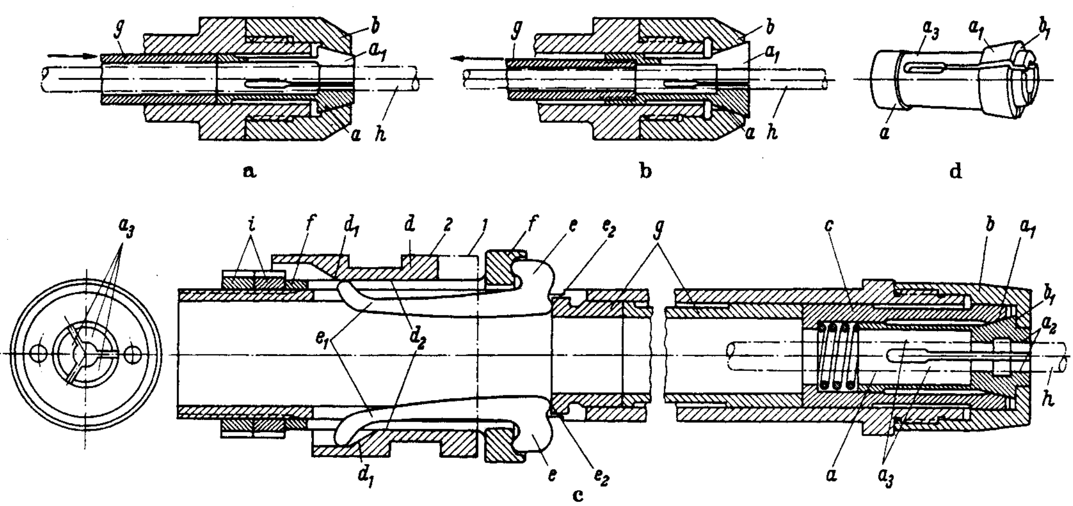


FIG. 241

(bar h) being axially displaced either to the right (Fig. 241a) or to the left (Fig. 241b), during the clamping operation. In the arrangement, Fig. 241c, the collet a (Fig. 241d) is held axially in position by nut b with thrust face b_1, in such a manner that the collet itself does not move axially at all during the clamping operation, when the clamping bush c is pushed over the cone a_1.

The clamping operation is initiated by a movement of the ring d from the right (position 1) to the left (position 2). The internal conical surfaces d_1 push the arms e_1 of the bell crank levers e inwards. The bell crank levers e are axially supported by the adjusting ring f and, as a result, their protruding parts e_2 push the tube g to the right. Tube g in turn pushes the clamping tube c to the right and thus over the conical surfaces a_1 of the collet a. As a result, the collet jaws a_2 are pressed radially against the workpiece h. The movability of the jaws a_2 is ensured by the elasticity of the jaw segments a_3 (see Fig. 241d). After the clamping operation, the bell crank levers are held irreversibly in position by the internal cylindrical surface d_2 of the ring d, so that the clamping action in position 2 is maintained.

Fine adjustment to suit the required workpiece diameter is obtained by axially shifting the supporting points on ring f for the bell crank levers e by means of the adjusting nuts i. The elastic member of the clamping mechanism is provided by the slender arms e_1 of the bell crank levers e.

The *mechanisms for selecting, producing and changing the working speeds and for transmitting the required energy* have been discussed earlier. They have to be controlled and operated in accordance with the required working conditions.

Apart from selecting and producing the required working speeds, it is necessary that the various *tools* required for different operations be *brought into their working positions* before a feed movement can be started. Instead of using a single cutting tool for a number of operations and progressively setting it to the required depth of cut for each operation, it is possible to assign to each operation a separate tool, which can be set exclusively for this one operation, so that it need only be provided with the necessary feed movement. The setting of the various tools must, therefore, satisfy the required dimensions of the workpiece. In the case of turning, for instance, this refers to the diameter and the lengths which have to be turned with each tool.

Each cutting tool or each set of cutting tools for a particular operation may either be clamped to its own slide, the slide being fed forward either separately or simultaneously with the others (Fig. 242), or alternatively, all tools can be located on a common tool holder, (Fig. 243), in such a

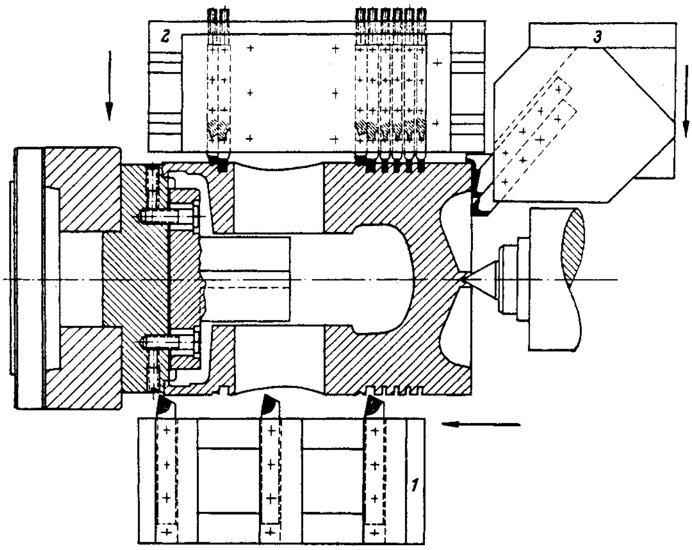
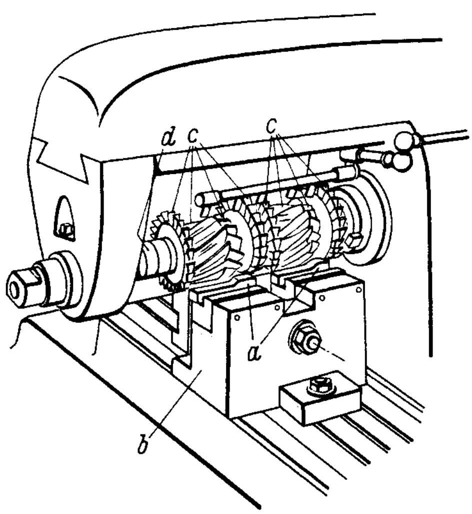


FIG. 242. Machining of a piston on a Churchill–Fay lathe (Churchill–Redman Ltd., Halifax). Operation 1: Cylindrical turning; Operation 2: a) Grooving (roughing); b) grooving (finishing with second, identical, cross slide); Operation 3: Surfacing (roughing and finishing).



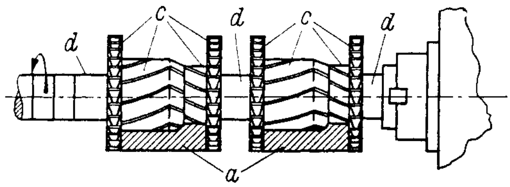


FIG. 243. *a*—2 workpieces; *b*—Milling fixture; *c*—2 sets of milling cutters; *d*—Spacers on the milling arbor.

manner that their relative positions correspond to those dimensions of the workpiece which have to be machined simultaneously.

The application of a turret head (Figs. 244 to 246) permits a larger number of tool holders with fixed cutting tools to be put into their working positions one after the other, the feed movement

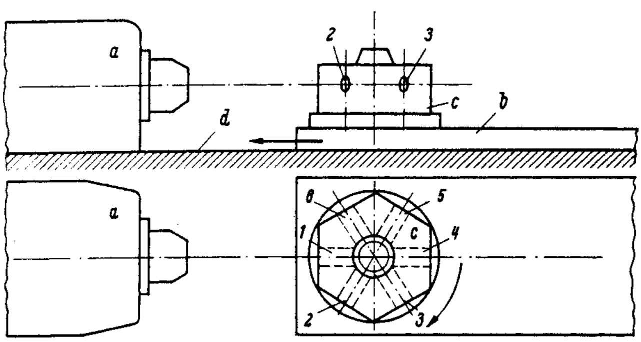


FIG. 244

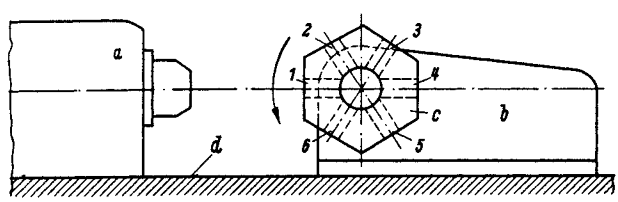


FIG. 245

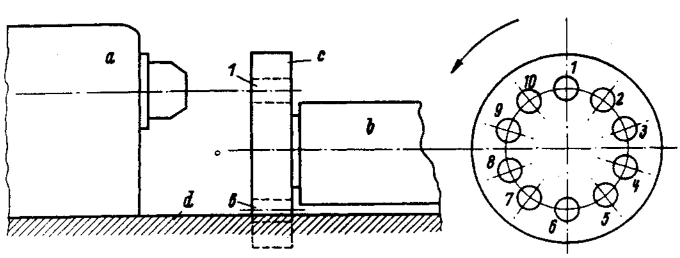


FIG. 246

FIGS. 244, 245 and 246. Turret head arrangements. *a*) Spindle head; *b*) Turret slide; *c*) Turret head; *d*) Machine bed. *1*, *2*, *3*, etc.—Tool holder bores or spigots in the turret head (No. *1* is drawn in line with the main spindle).

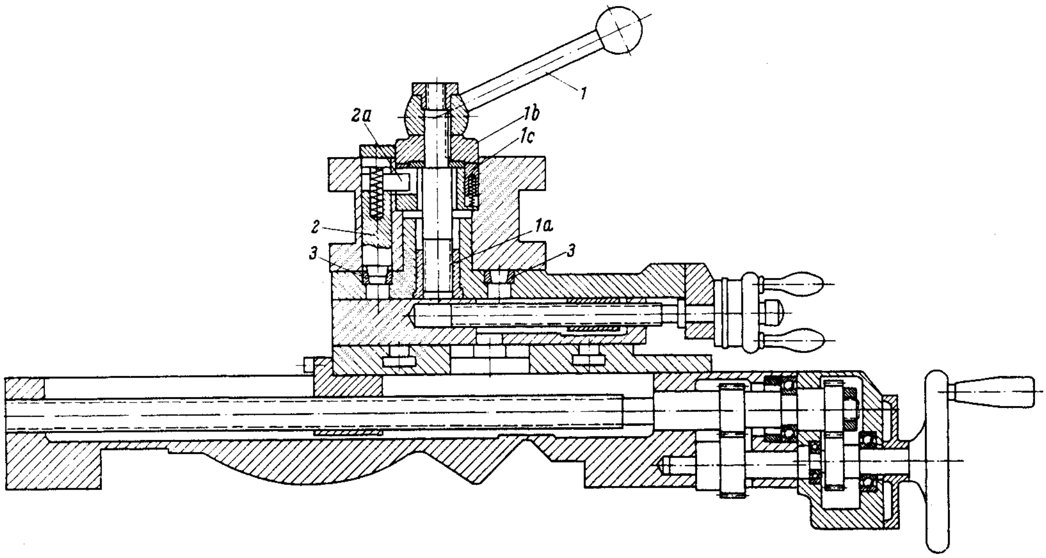


FIG. 247. Square turret (Dean, Smith & Grace Ltd., Keighley). *1*—Handlever for clamping (clockwise movement) and unclamping and turning (anti-clockwise movement) of the turret; *1a*—Clamping thread; *1b*—Rotating ratchet with latch *1c*; *2*—Index pin which is lifted from bush *3* by means of peg *2a* when the screw *1a* is turned and the turret is unclamped.

for all cutting tools being produced by a single slide, i.e. the turret slide. The turret head can be swivelled on a vertical (Fig. 244) or horizontal axis (Figs. 245 and 246), and the tools can be arranged on the circumference (Figs. 244 and 245) or on the face (Fig. 246) of the turret.

The support of the turret shown in Fig. 246 can be made stiffer than that of the turrets of Figs. 244 and 245, because the bearing length is not limited by the width of the saddle or the height of the axis above the machine bed. For this reason, turret heads of the type shown in Fig. 244 are usually supported by a thrust ring of large diameter and clamped in position by a device acting on the largest possible diameter (see Fig. 249).

If such a turret head is used, on the other hand, the height of the cutting tools above the bed remains constant and the tools can be swung freely above a cross slide in the middle of the bed. Those tools which are not in operation can, therefore, be moved away from the operating range of the machine easily permitting the provision of a cross slide. This is particularly important for heavy profiling and surfacing operations. The number of tool holders can be increased by the use of a square turret on the cross slide in adddition to the turret head on the saddle (Fig. 247, see also Fig. 255).

With the turret head of the type shown in Fig. 246, it is possible to carry out light surfacing

FIG. 248. Surfacing with a turret head of the type (Fig. 246). a—Surfacing over a large diameter. The tools *A* and *B* are roughing one half (*1* and *2*) each of the surface and tool *C* comes out to complete finishing operations (*3*). b—Surfacing of two separate concentric annular faces. The tools *A* and *B* are roughing the two faces (*1* and *2*) simultaneously, and the tools *C* and *D* are afterwards finishing the same faces (1^s and 2^s) simultaneously.

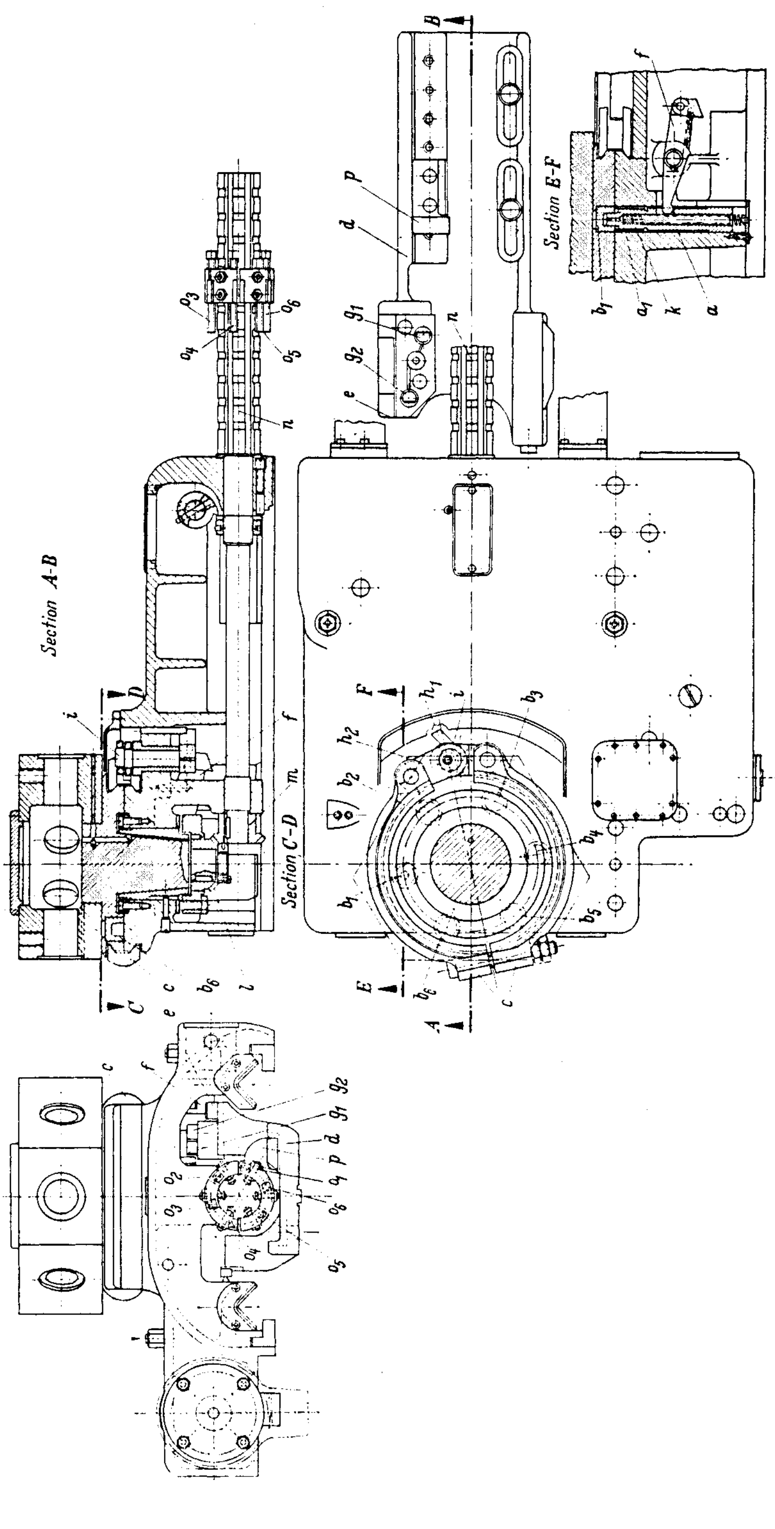


FIG. 249. Saddle of a turret lathe (Gebr. Heinemann A.G., St. Georgen, Germany).

operations by rotating it round its axis, the tool carrying out a surfacing movement on a circular arc (Fig. 248).

The most important problems in the design of turret heads are the stiffness of the bearing, the speed of the swivelling movement and the accuracy and stiffness with which the head is locked in its operational position. The turret head (Fig. 249) can be locked in six positions by means of an index pin *a* which is guided in a hardened bush a_1 and whose tapered end falls into one of the six hardened bushes (b_1–b_6). In order to relieve the load on the index pin during the turning operations, the turret head can be clamped in position by a ring *c*. The turret is unclamped and unlocked when the turret slide is withdrawn (from left to right) from its working position. The plate *d*, which is fixed to the bed of the machine, withdraws the index pin from the indexing bush by means of the stop *e* and the lever *f*, whilst the spring loaded stop g_1 via lever h_1 and eccentric *i* releases the clamping ring *c*. After the operator has turned the turret head by hand into one of its six working positions, and during the further withdrawal movement of the turret, the lever *f* is released and the spring *k* pushes the index pin *a* into one of the hardened bushes (b_1–b_6), the tapered end of the index pin compensating for slight inaccuracies in the swivelling movement and pulling the turret head exactly into the required position.

When the turret head is now moved forward again (from right to left), the spring loaded step g_2 and lever h_2 tighten the clamping ring *c* again and the turret head is ready for the next operation. As the turret head is rotated, it turns the stop carrier *n* via bevel gears *l* and *m*. This brings one of the six adjustable stops (o_1–o_6) into line with the fixed stop *p*, and the longitudinal movement is thus limited in accordance with the requirements of the operation for which the turret has been set.

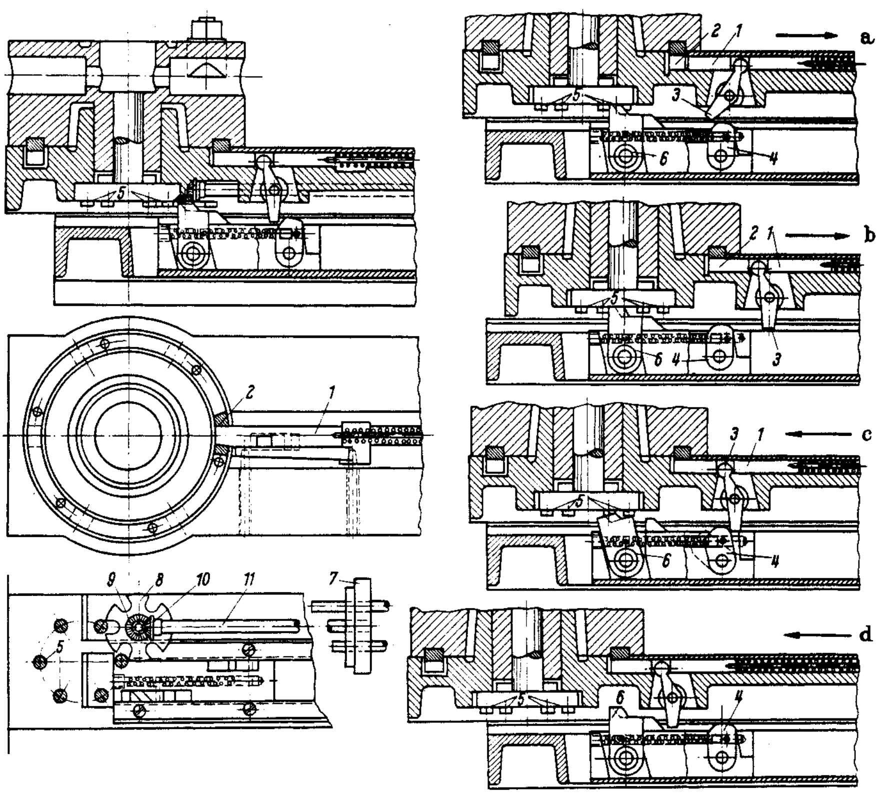


FIG. 250. Switching of the turret head (Loewe, Berlin, Germany) (from G. SCHLESINGER, *Die Werkzeugmaschinen, loc. cit.*). a—*Withdrawal*: Locking pin *1* is withdrawn from slot *2* by means of lever *3* which is operated by stop *4*. b—*Swivel*: Turret head is rotated by pegs *5* and stop *6*, while the drum *7* is also moved correspondingly ($\frac{1}{6}$ of a turn) by means of pegs *5*, Geneva wheel *8*, bevel gears *9* and *10* and shaft *11*. c—*Forward Movement*: Spring loaded stops *4* and *6* allow lever *3* and pegs *5* to pass. d—*Working stroke*: The whole above mechanism is out of action.

In an older design (Fig. 250), the turret is unlocked, swivelled and re-locked as the operator withdraws the turret slide from the workpiece.

Turret heads are used not only for centre lathes with horizontal axes, but also for vertical boring (Fig. 251a), drilling (Fig. 251b), and other machines.

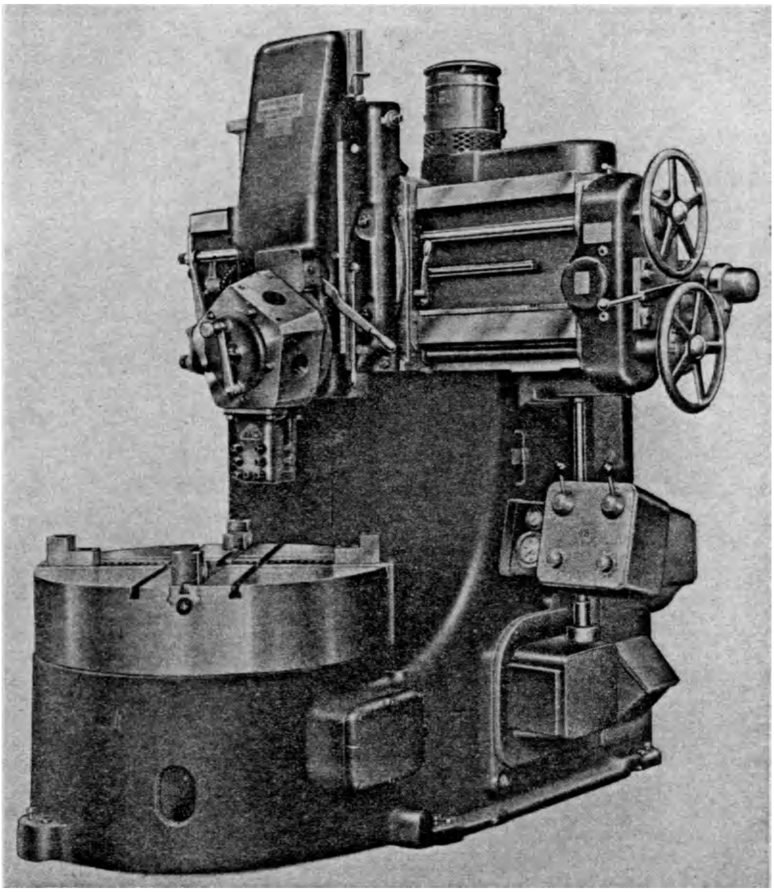


FIG. 251a. Vertical boring machine (Webster & Bennett, Coventry).

The equipment for *measuring the workpiece* and for *setting the machine* in accordance with this measurement is best discussed within the framework of a general survey of automatic control operations.

An automatic control operation can be divided into the signal which initiates the operation, and the actual control operation.

(a) *The signal* can be produced and transmitted:

(i) at specified intervals, e.g. by means of a cam shaft which rotates at a given speed,

(ii) as a function of operational movements, i.e. during or at the end of the movement of a slide, e.g. by means of stops or trip dogs,

(iii) as a function of actual operations, i.e. by independent measurement of the result, e.g. by measuring the length of traverse of a slide or the dimension produced by a machining operation, and by comparing this measurement with a required value which has been previously set or stored. In this case the signal for terminating one operation or for initiating the subsequent one will be given only when the required and actual values coincide within permissible limits,

(iv) in accordance with a planned sequence of operations *and* as a function of the results, i.e.

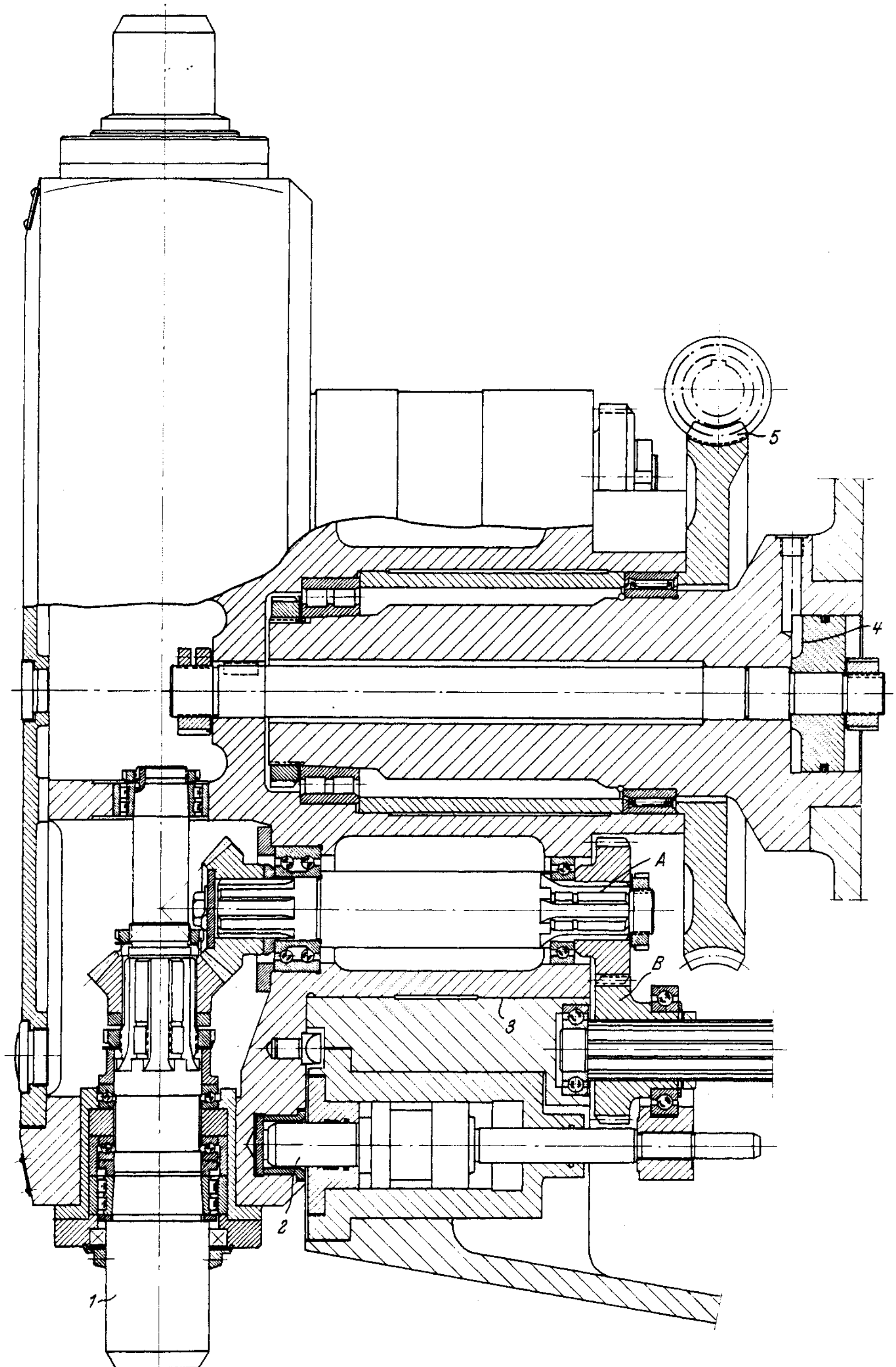


FIG. 251b. Section through the turret head of a drilling machine (Gebr: Heller, Nürtingen, Germany). *1*—Drilling spindle; *2*—Hydraulic index; *3*—Turret bearing arrangement; *4*—Turret clamping; *5*—Turret rotation. Note that only the spindle, which is in the working position, is actually power driven, its driving gear *A* being swung out of mesh with the main driving pinion *B* as soon as the turret rotation moves it away from its vertical downward position.

by checking the results continuously during operations, by means of independent measurements, and correcting the signals accordingly (feedback).

These four alternatives will now be discussed in detail.

(i) One of the oldest applications in which the various operations are started and stopped at specified time intervals can be found in automatic lathes, where a master control shaft rotates at a speed n which corresponds to the time T required for the production of one workpiece ($n = 1/T$). The control shaft carries a number of cams, each of which initiates a certain movement at a given instant and controls it until its completion. A complete cycle, corresponding to the production of one workpiece from the feeding-in of the workpiece material to the parting-off of the finished article, occurs during one revolution of the cam shaft and is repeated with each subsequent revolution. The instantaneous positions of the various control levers and tool carriers are, therefore,

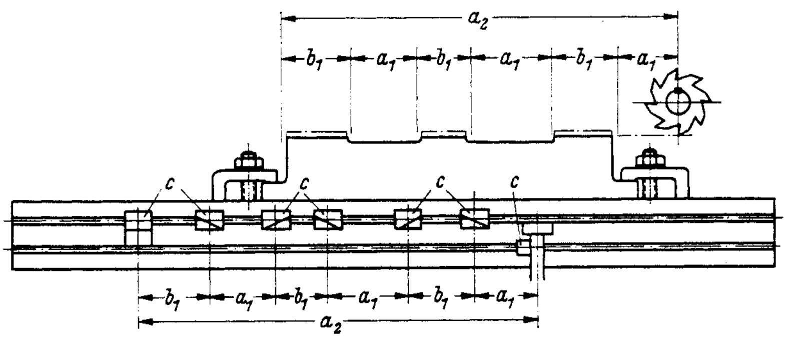


FIG. 252. a_1—Rapid table traverse to the right; a_2—Rapid table traverse to the left; b_1—Feed movement of the milling machine table to the right; c—Adjustable trip dogs on the milling machine table.

determined by the angle of rotation of the control shaft at any moment and a particular operation which corresponds to a certain rotational position of the cam shaft is initiated and then carried out, whether or not the previous operation has been completed. This would almost correspond to a control by punched card, punched tape or magnetic tape, in which signals are given only for starting and stopping different operations without any check on the satisfactory completion of one operation before the next one is started. If, for instance, through fracture or wear of a load-transmitting element, or through slipping of a friction drive, one operation could not be completed, this would not prevent the subsequent operation being started at the moment determined by the angular position of the cam shaft.

(ii) This difficulty is eliminated in cases where a moving slide or a lever at the end of its stroke produces a signal which initiates the drive for the next operation. The sequence of operations is then controlled by adjustable stops in such a manner that the driven and not the driving parts provide the signals. Deformations due, e.g. to the strain of the load-transmitting elements, cannot influence the accuracy of timing the signal. The adjustable stops may serve for starting or stopping a feed movement, a rapid traverse or a return stroke, and thus complete working cycles can be carried out automatically.

A simple example of such an arrangement is shown in Fig. 252. The workpiece is clamped to a milling machine table which carries the workpiece at rapid traverse speed to the milling cutter as soon as the table drive is engaged. The drive then changes to the required feed rate for milling the first machined surface, again passes the recess at rapid traverse speed until the cutter reaches the beginning of the next surface which has to be machined, moves again at the milling feed rate when machining this next surface, and so on, until at the end the drive is reversed and the table with the workpiece returned at rapid traverse speed to its initial position and stopped. The finished workpiece can then be removed and a new one clamped to the table. The accuracy of this operation depends upon the setting of the trip dogs and the response speed of the controls. A simple template control can be considered as a development from trip dog devices. The template would represent a continuously acting trip dog, the profile of which has been specifically adapted to the requirements of the work.

The problem has also been solved electrically.[51] In this case, the drive for each movement has its own circuit which is operated by a relay. If this type of control is applied to a turret lathe, it controls, for instance, the following: opening the collet, feeding the bar material, closing the collet, moving the turret head forward, withdrawing the turret head, moving the cross slide forward, withdrawing the cross slide, etc. Once the sequence is set, each drive is started when the corresponding relay is switched on following the operation of the limit switch which stops the previous operation.

(iii) This type of control is used, for instance, for positioning the tables of horizontal boring machines and jig borers. The operator initiates an operation by starting the positioning movement. An independent measuring device transmits a signal as soon as the moving part (saddle or table) has reached the required position. This signal causes the positioning drive to be disengaged, the table to be clamped in position and the boring operation started.

If the dimensions or the shape of a machined surface can be changed by a movement of the tool carrier, these dimensions can be checked after completion of a cut and the position of the tool carrier adjusted in order to correct deviations from the required values. If such a measurement occurs only after completion of an operation, the adjustment of the tool setting ensures that the dimensions of subsequent workpieces will be kept within the permissible limits. However, the measurement can also be carried out before completion of an operation, so that the required dimensions of the workpiece that is actually being machined can be obtained by additional cuts.

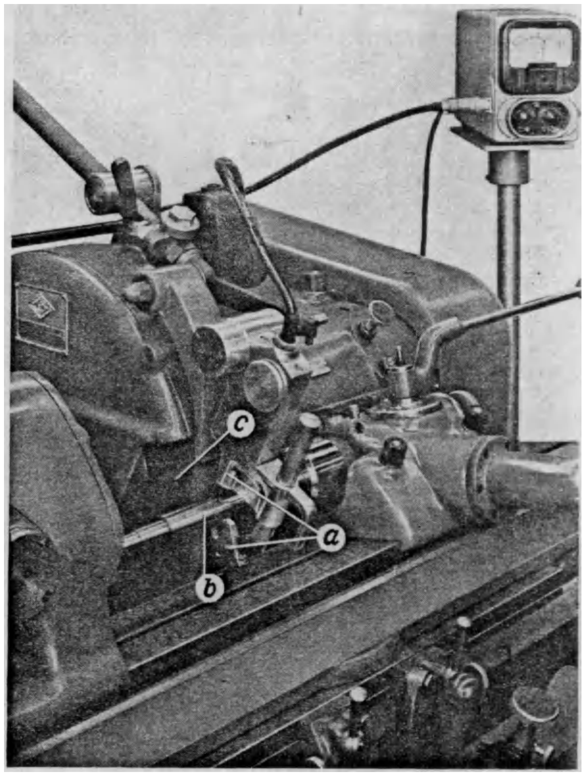


FIG. 253. Cylindrical grinding machine with electronic measuring device (Landis–Lund, Keighley).

The setting of the tool can be obtained in various ways. In the case of grinding operations, it is possible to dress the grinding wheel in a given position of the wheel head, to suit the required dimensions of the workpiece. Alternatively, it is possible to feed the tool carrier towards the workpiece until the required dimension is obtained. The sizing device of the cylindrical grinding machine (Fig. 253), works on the principle of a Wheatstone bridge, which is balanced when the measurement of the electronic gauge *a* corresponds to the required diameter of the workpiece *b*, which has been set before the operation is started. As soon as this balance is reached, the action of a relay stops the infeed of the grinding wheel *c* and withdraws the wheel head. The accuracy of this instrument is claimed to be 0·001 mm (0·00004 in.).

(iv) Instead of carrying out the measurement at the end of a cut it is possible to measure continuously during the operation and to correct, if necessary, the relative positions between workpiece and tool.

When profiles are machined in this manner, continuous checking and correcting of several slide movements may be required. This type of control requires measuring devices which work independently of mechanical influences and check continuously the changing positions of the various parts of the machine. By transmitting the results, they produce "error" signals which in turn control the various driving devices. Only this last step, the closing of the control loop, leads to complete automation, i.e. to full substitution of the thinking machine operator. The place of the drawing normally handed to the operator is then taken by the "programme", which is introduced into the machine by means of punched card, punched tape, magnetic tape, or other means.

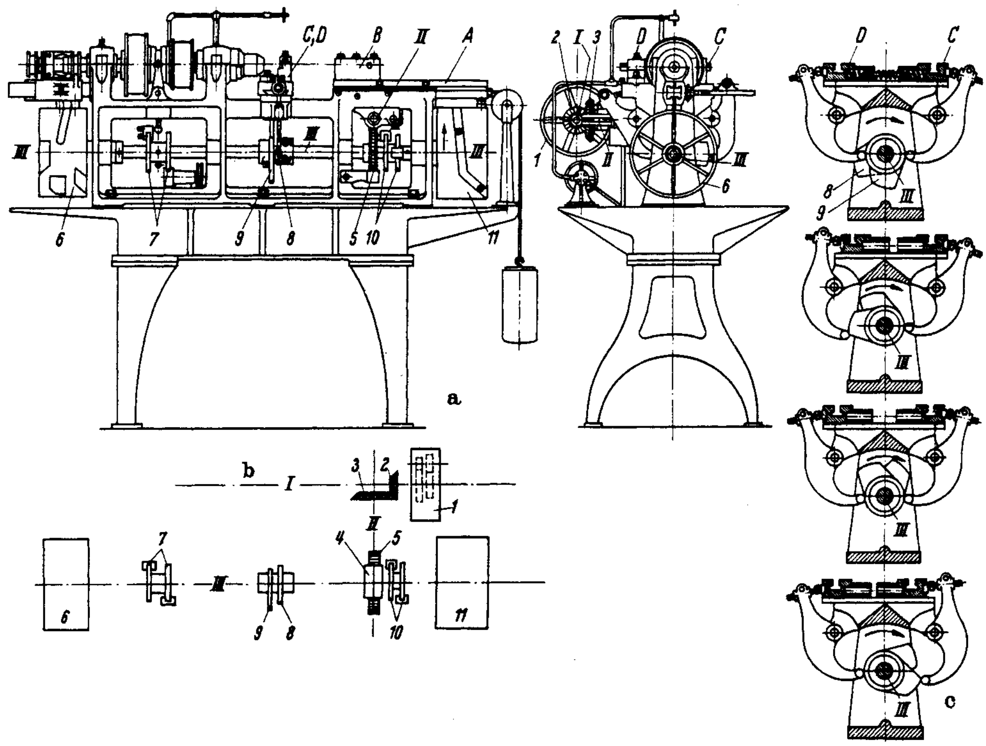


FIG. 254. Automatic lathe with cam control (from SCHLESINGER, *loc. cit.*). a) View of the machine; b) Drive for the cam shaft; c) Feed drive for cross slides. *I*—Driving shaft for cam shaft; *II*—Worm shaft; *III*—Cam shaft; *A*—Turret slide; *B*—Turret head; *C, D*—Cross slides; *1*—Epicyclic gear for feed and rapid drive of the cam shaft; *2, 3*—Bevel gears; *4, 5*—Worm and worm wheel; *6*—Cams for material feed and clamping; *7*—Dogs for reversing the spindle drive; *8, 9*—Cams for cross slide feed; *10*—Dogs for controlling the epicyclic gear; *11*—Cams for turret slide.

(b) The *energy* for carrying out the actual control operations can be provided and transmitted in four different ways:

(i) introduced together with the signal,
(ii) provided by the controlled machine part during its operational movement,
(iii) stored by the controlled part during its operational movement and released at the appropriate moment by the signal mechanism,
(iv) provided independently of the controlled part and the signal mechanism and released at the appropriate moment by the latter.

These four methods will now be discussed.

(i) This was used in the first automatic lathes. They were equipped with cam controls, the cams acting simultaneously for transmitting the signals and the drive for the feed movements (Fig. 254). The advantage of such an arrangement is its simplicity, its disadvantage—the necessity of providing sufficient stiffness and strength for the control mechanism to transmit the required energy. As a result, the control elements become relatively heavy and the possible speed of signal transmission is reduced. The accuracy is also unfavourably affected by speed variations of the control shaft and unavoidable deformations and wear of the highly stressed load transmitting elements. Moreover, the operating speed of the control elements is a function of the rotational speed of the master cam shaft, which in turn depends upon the work which the machine has to produce ($n = 1/T$, see page 173). This can lead to switching operations which are uneconomically slow, but the difficulty can be overcome by special high speed devices (see Fig. 254), in which the master cam shaft changes, so to speak, its own speed for initiating switching operations after a cutting operation and returns to its original speed before the next cutting operation is started.

(ii) The separate supply of energy for control operations, as distinct from that for signal transmissions is not necessary if the movement of the part which is to be controlled can provide the energy for carrying out the control operation. If it is a question only of stopping the movement of a machine part, the movement of a clutch lever or interruption of an electrical contact might be

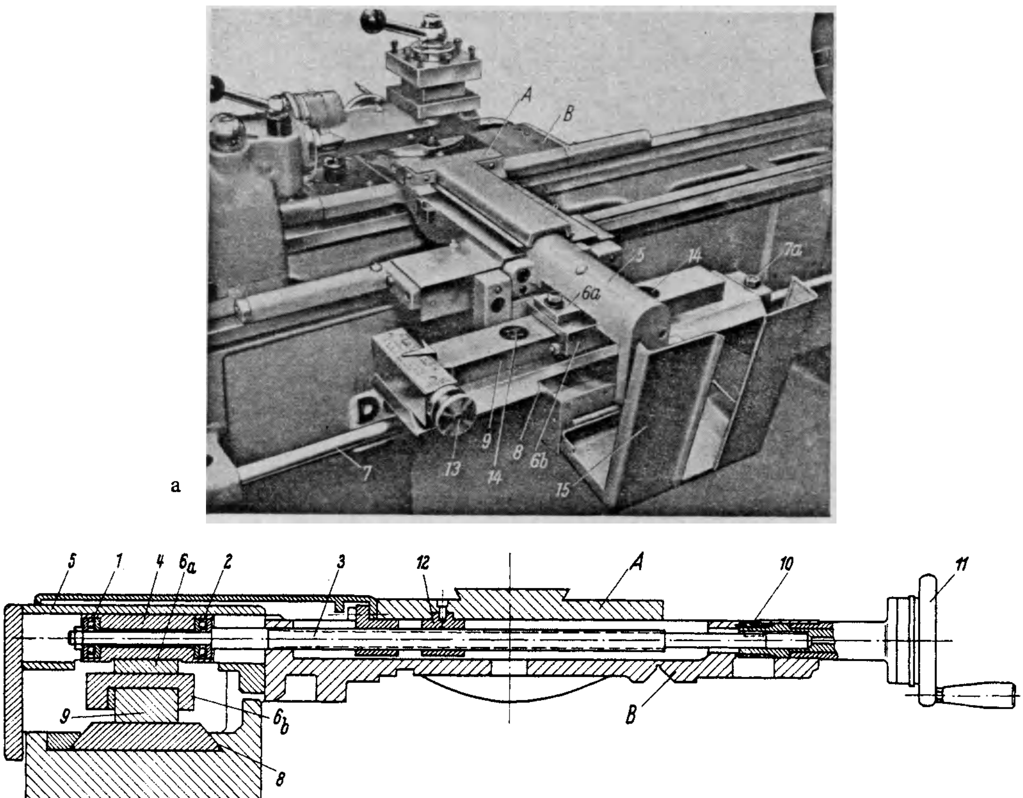


FIG. 255. Taper turning attachment (Dean, Smith & Grace Ltd., Keighley).

considered sufficient to disengage the drive or to stop the driving motor and thus the operational movement. However, the speed of disengagement depends upon the moving speed of the part which is being controlled and in the case of a dog clutch, the speed of disengagement must not be too low (see page 163).

The problem becomes more difficult when not only a specific movement has to be stopped but also a further operation has to be started, e.g. when the movement of a machine part has to be reversed. In this case, the drive for the return movement must be engaged after the drive for the forward movement is out of action. An example is the older type reversing drive of planing machines which works by means of open and crossed belts. As long as the table is driven by one of the two belts, it shifts the belts into their required positions. At a certain moment, however, both the forward and the reverse drives are disengaged for a short period during which the kinetic energy of the table must be used for completing the reversing operation. It will be appreciated that in such cases, the reversing accuracy cannot be very high, as it depends upon the table speed, the friction conditions and other variables. The problem is simpler when, for instance, only the speed has to be changed, the direction of movement remaining the same. An example is the free wheel arrangement (see page 100) which can be switched easily from "feed" to "rapid traverse" as the slow feed drive continues to operate. Here the control energy is available until the moment the rapid traverse drive is switched on and it is again available as soon as this rapid traverse drive is switched off and the ordinary feed drive continues.

In a similar manner the movement of one slide can be used for controlling the movement of another. In the taper turning attachment (Figs. 255a and 255b), the traverse of the cross slide *A* is controlled by the longitudinal movement of the saddle *B*. For this purpose, the thrust bearings *1* and *2* of the cross traverse screw *3* are located in a cylindrical block *4*, which is carried in a housing *5* and guided by parts *6a* and *6b* on the swivelling bar *9*. Bar *9* is connected to the lathe bed by means

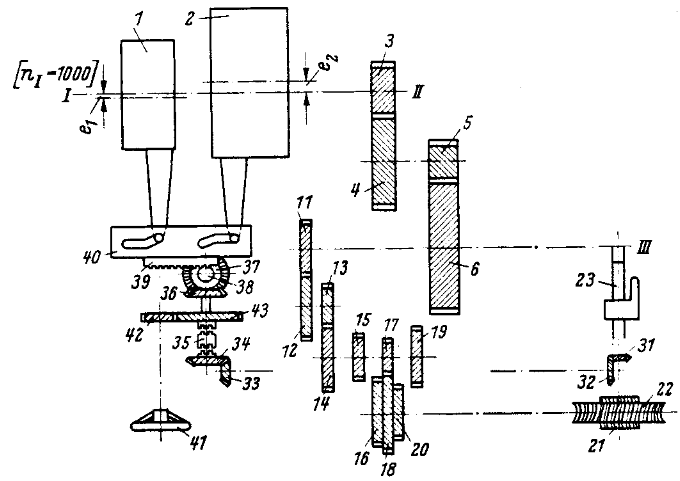


FIG. 256. (From SCHLESINGER, *loc. cit.*).

of rod *7*, clamping screw *7a* and dovetail slideway *8*. During the longitudinal movement of the saddle, the cross traverse screw *3* can move axially in the pinion *10* in accordance with the inclination of the guide bar *9* relative to the axis of turning and thus produce the required cross traverse movement of cross slide *A* via nut *12*. The operator can set the depth of cut by means of hand wheel *11*, cross traverse screw *3* and nut *12*, even whilst the taper turning operation is in progress. The required inclination of the guide bar *9* and thus the taper angle of the workpiece is adjusted by setting screw *13*. The guide bar *9* can be clamped in position by screws *14* and protected by a cover *15*.

This device enables tapers up to 20° apex angle to be turned. For larger angles, friction may cause difficulties. The length of the taper which can be machined with this device depends upon the length of the guide bar (*9*).

The schematic drawing (Fig. 256)* shows how the movement of a cross slide can be used for changing the speed of a hydraulically driven lathe spindle in order to keep the cutting speed constant during surfacing operations. The spindle is driven by a hydraulic unit (pump *1* and motor *2*), infinitely fine speed variation being obtained by changing the eccentricities of the pump (e_1) and the motor (e_2).

The pump shaft *I* is driven at constant speed (n_I = 1000 rev/min) and in turn drives the motor shaft *II*, which drives the lathe spindle *III* via gears *3/4* and *5/6*.

The lathe spindle drives the worm *21* via gears *11/12* and *13/14* and sliding gears (either *15/16*, *17/18*, or *19/20*). The cross traverse screw is then driven by the worm and worm wheel (*21/22*).

As the position of the cross slide depends upon the angle of rotation (measured from a given datum position) of the cross traverse screw, the rotation of the latter can be used for controlling the spindle speed. For this purpose, the cross traverse screw via bevel gears *31/32*, *33/34*, clutch

* Krug (ref. 44) has pointed out that this arrangement is unsuitable for turning diameters above 500 mm (20 in.).

35, bevel gears *36/37* drives the pinion *38* and the rack *39*. This rack shifts the cam plate *40*, which carries two slotted cams. These vary first the pump eccentricity e_1 and then the motor eccentricity e_2. If the spindle speed has to be manually adjusted the clutch *35* can be engaged with gear *43* and the cam plate *40* then be moved by hand wheel *41*, gears *42/43*, clutch *35*, bevel gears *36/37*, pinion *38* and rack *39*.

If the volumetric efficiency of the hydraulic drive is η the spindle speed is

$$n_{III} = \eta . \frac{c_1 . e_1}{c_2 . e_2} . n_I . u \qquad \text{(see page 135)},$$

u being the transmission ratio of the drive *3/4*, *5/6*. Under the simplified assumption of a constant mean leakage loss of 5 per cent,

$$\eta = 0{\cdot}95 \quad \text{and} \quad n_{III} = 0{\cdot}95 . 1\,000 . \frac{c_1 . e_1}{c_2 . e_2} . u$$

At the turning radius r, the cutting speed v is

$$v = 2\pi r n_{III}$$

and in order to keep this constant, it is necessary that

$$r n_{III} = \text{const} = \frac{v}{2\pi}.$$

It will be appreciated that the cutting speed can be kept constant only down to a minimum turning radius, because for $r = 0$, n_{III} ought to be infinite. Below this minimum radius, therefore, the spindle speed will remain constant and the cutting speed will drop.

If the pump is adjusted, the motor eccentricity remains constant at its maximum value (see page 135).

$$e_2 = e_{2_{\max}} = \text{const.}$$

For this reason:

$$r n_{III} = r . 950 . u . \frac{c_1}{c_2} . \frac{1}{e_{2_{\max}}} . e_1 = \frac{v}{2\pi}$$

and

$$e_1 . r = \text{const} = \frac{v}{2\pi} . e_{2\max} . \frac{c_2}{c_1} . \frac{1}{950u}$$

In the case of motor adjustment, the pump eccentricity remains constant at its maximum value, so that

$$e_1 = e_{1_{\max}} = \text{const.}$$

and

$$\frac{e_2}{r} = \text{const} = \frac{2\pi}{v} . e_{1_{\max}} . \frac{c_1}{c_2} . 950u.$$

Figure 257 shows these relationships for a set cutting speed of 65 ft/min and a speed range ratio of 8 : 1. The maximum spindle speed is 355 rev/min. For a cutting speed of 65 ft/min, this corresponds to a minimum turning radius of 0.35 in., below which the cutting speed drops. In the case of the hydraulic unit described before, the pump adjustment (R_1) serves up to a spindle speed of 85 rev/min and above this the motor adjustment (R_2) is used.

(iii) In the discussion of the dog clutch (see page 163), the danger of too slow a disengagement under heavy load was pointed out and especially at low speeds, when the kinetic energy of the moving mass is small. However, the energy required for operating such a clutch can be obtained from the drive and stored until it is required when the disengaging operation is to start. An example is shown in Fig. 258. A spring *e* is compressed by the driven part *a* during its movement from right to left. As soon as the highest point of cam *c*, fixed to the operating bar *b*, passes the peak of the spring loaded cam *d* (spring fully compressed), spring *e* throws the bar *b* by means of the other side of cam *c* further to the left, so that lever *f*, which now touches the right-hand end of slot *g*, is thrown to the left and the clutch *k* is disengaged.

The functioning of this arrangement is greatly influenced by friction conditions. Nevertheless, in the design shown in Fig. 258 this need not lead to complete failure, because even if the spring fails to disengage the clutch, the continuing movement of part *a* to the left would eventually operate the bar *b* and the lever *f*, thus still disengaging the clutch *k*, although at a much reduced speed of operation.

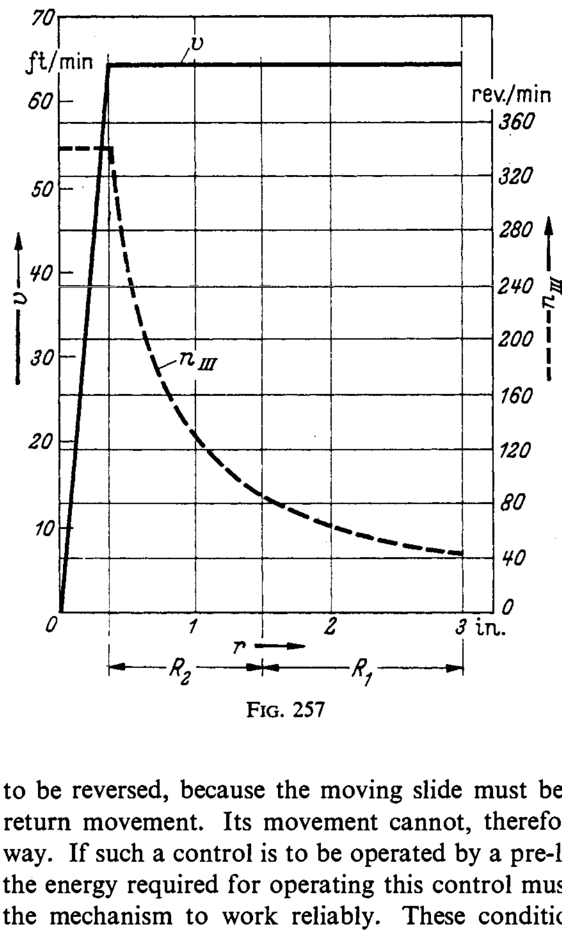


FIG. 257

Such a clutch can thus be disengaged, even if the force of the spring which is pre-loaded by the movement of the slide, is insufficient to overcome frictional and other resistances, because continuing movement of the slide will result eventually in the connexion between drive gearbox and feed screw being interrupted. The conditions are, however, more difficult when a slide movement has

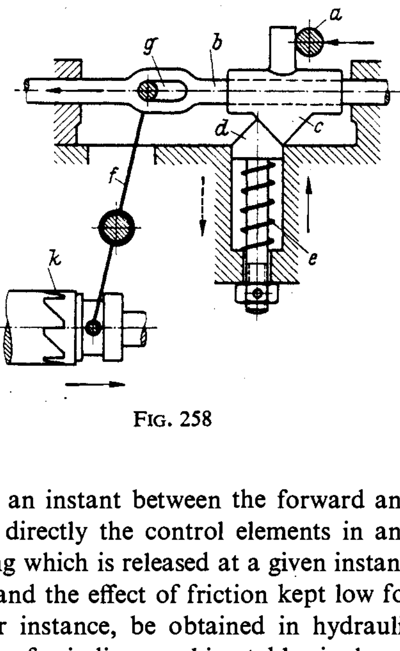


FIG. 258

to be reversed, because the moving slide must be at rest for an instant between the forward and return movement. Its movement cannot, therefore, operate directly the control elements in any way. If such a control is to be operated by a pre-loaded spring which is released at a given instant, the energy required for operating this control must be small and the effect of friction kept low for the mechanism to work reliably. These conditions can, for instance, be obtained in hydraulic devices. A latch mechanism used for reversing the movement of grinding machine tables is shown in Fig. 259. The reversing lever, which is pushed either to the right or to the left by trip dogs on the table, is rigidly connected to a lever *a*, and this carries two spring loaded plungers b_1 and b_2. These plungers rest against lever *c*, which operates valve spool *d* and is held in one of two extreme positions by latches e_1 and e_2, the latter being hinged in the housing. Lever *c* cannot, therefore, immediately follow any movement of lever *a*. Lever *a* carries at both ends adjustable set screws f_1 and f_2 and when it is slowly moved from one end position to the other, these screws act on latches e_1 and e_2, in such a manner that both lever *c* and valve spool *d* are slowly pushed from one position to the other, thus throttling the inlet and outlet channels of the valves and reducing the piston velocity. When at a given moment, e.g. during movement from right to left, the latch e_1

releases lever c and this is now pushed over at high speed by the spring acting on plunger b_1, the valve spool is brought into its extreme left-hand position and effects the reversing action. A throttle valve is inserted in the pipe g in order to reduce to a minimum any possible impact effects and to adjust the speed of reversing. If the table is to be reversed by hand, the two latches e_1 and e_2 can be lifted by means of cams h thus allowing an easy movement of the valve spool from right to left or vice versa. The weakness of this design lies in its sensitivity and the necessity for very accurate adjustment.

Before a control operation can be carried out by a mechanism which stores the energy in the way described above, the part whose movement is to be controlled (slide or lever) must move over a certain distance in order to pre-load the operating spring. This requires what may be called artificial backlash and therefore a certain minimum length of traverse, which in the case of the example shown in Fig. 258, would be determined by the length of the slot g. For very small strokes and high operating accuracy, such mechanisms are, therefore, unsuitable.

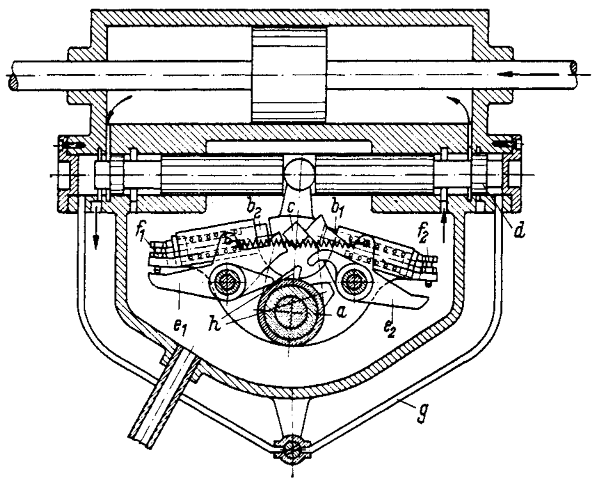


FIG. 259. Latch mechanism for the control of a hydraulic drive (German patent no. 445859).

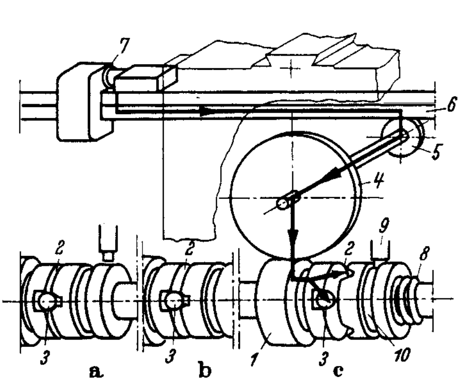


FIG. 260. Device for turning against rigid stops on a Schaerer lathe. (Industrie-Werke, Karlsruhe, Germany.)

For very accurate stroke limitation, e.g. for limiting the feed movement of lathe saddles or cross slides, overload protecting devices are used which disengage the driving clutch when a rigid stop is reached, the stop often being set with the aid of a micrometer screw. In this case, however, the overload protecting device must not only slip but fully disengage the drive as soon as the overload is encountered, or the rigid stop reached. Figure 260 is a schematic diagram of such a mechanism. The worm *1* is driven via sliding clutch *2* (see Fig. 238). In order to reduce friction losses and to increase the operating accuracy, the torque in the clutch is transmitted by means of hardened steel balls *3* which act between sloping flanks. The drive is transmitted from the clutch and the worm to worm wheel *4* and pinion *5*, the latter running on the rack *6*, which is fixed to the bed and thus drives the saddle of the lathe.

When the saddle touches a rigid stop *7*, or if the feed component of the cutting force exceeds a certain permissible value, the sliding clutch member, which is held in position by spring *8*, is moved from its normal position (Fig. 260a) to the right (Fig. 260b) (the arrows show the force transmission in the clutch). As soon as the clutch is completely disengaged (Fig. 260c), a pin *9* drops into the groove *10* and the drive to the worm is completely interrupted. In order to re-engage the feed drive, the pin *9* is withdrawn by means of a hand lever (see Fig. 453), whereupon the clutch can again transmit the required torque unless, of course, the cause of the overload has not been removed.

Another solution is the "dropping worm mechanism". Here, the driving worm is thrown out of engagement when an overload occurs. The worm is held in a tilting cradle (the tilting axis S) lying at right angles to (Fig. 261a), or parallel to (Fig. 261b) the axis of the worm. In the former case, a universal joint is required between driving shaft *I* and worm shaft *II*; in the latter case a set of gears (*1*, *2*) is sufficient. The worm cradle has a stop *3* which rests on a fixed surface *4* of the apron housing and thus keeps the worm engaged. The worm is driven via an overload clutch (see Fig. 238), which is held engaged by a spring *5*. If an overload occurs or if the saddle is prevented from moving by a micrometer stop the clutch is disengaged against the action of the spring *5* and thus moves the stop *3* off the surface *4*, so that the worm drops out of engagement with the worm wheeel *7*, often under the urge of a disengaging spring. The drive is thus interrupted.

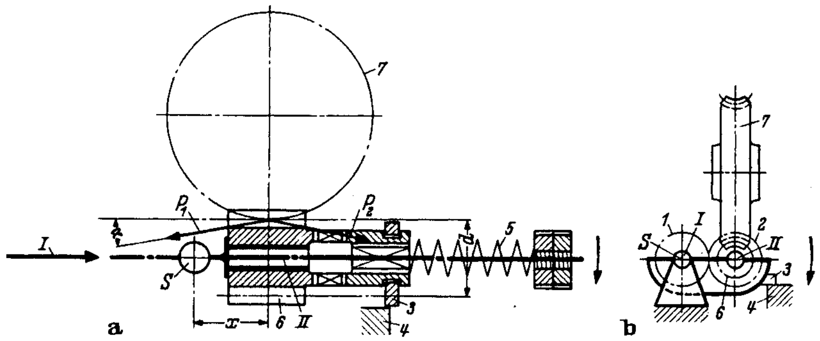


FIG. 261

It is important to arrange the tilting axis of the worm-carrying cradle in such a manner that the force acting on the worm teeth does not counteract the disengaging movement, as otherwise the tooth pressure, which increases with the load, can become so high that it prevents the disengagement of the worm. If, e.g. in the arrangement shown in Fig. 261a, the distance x between the tilting axis S and the worm wheel centre is less than $d/2\tan\alpha$ (α being the pressure angle of the worm), a tooth pressure acting in the direction P_1 would keep the worm engaged, whilst in the case of rotation in the opposite direction, the tooth pressure P_2 would assist in throwing the worm out of engagement. The tooth pressure between gears *1* and *2* (Fig. 261b) is much smaller because of the higher speed of these gears and the lower torque which they transmit, and this arrangement is therefore easier to control. It is, of course, also possible to arrange the dropping worm ahead of a reversing gear, so that the worm drive always acts in the same direction, i.e. in the direction which favours the disengagement.

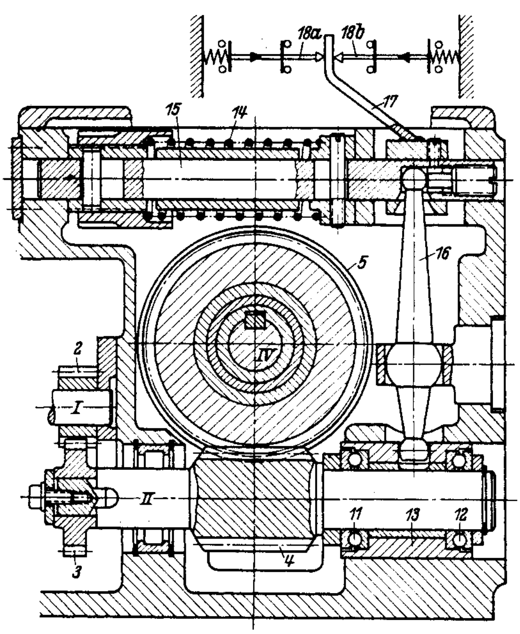


FIG. 262

Instead of the slipping clutch, which is disengaged by its axial movement, an electromechanical design (Fig. 262)* is used in connexion with the rapid traverse mechanism (Fig. 131b). Here, use is made of the axial force component in a worm drive which displaces the worm, this movement then being transformed into an electrical signal. The signal can be used not only for stopping the driving motor, but also for initiating a new operation after the moving slide has reached the position determined by a rigid stop (see *7*, Fig. 260). The thrust bearings *11* and *12* of the worm shaft *II* are held in an axially movable bush *13*. The force required for displacing the bush *13* in either direction (i.e. forward or reverse movement) is determined by a spring *14*, which holds the bush *13* in its axial working position by means of bar *15*

* Gebr. Honsberg, Remscheid-Hasten.

and lever *16*. When an overload occurs (i.e. when a rigid stop is reached or when the feed force component exceeds a pre-determined value), the axial component of the tooth pressure compresses the spring *14* and thus moves bar *15*. The latter carries a bar *17*, which operates either switch *18a* or *18b* and thus initiates the required electrical operation.

The following points must be observed in the design of any overload device which may serve for controlling the traverse of slides by means of rigid stops. In contrast to mechanisms which store the required control energy during a movement of the slides, the energy is here stored in the driving mechanism, even if the movement of the slide has already ceased (because it is actually held in position by the rigid stop). This means that the whole driving mechanism is increasingly strained until the drive is actually disengaged, and the strain energy thus stored is suddenly released when the drive is disconnected. The lower the stiffness of the mechanism, i.e. the bending stiffness of levers and gears and the torsional stiffness of shafts and wheels, the greater will be the deformations

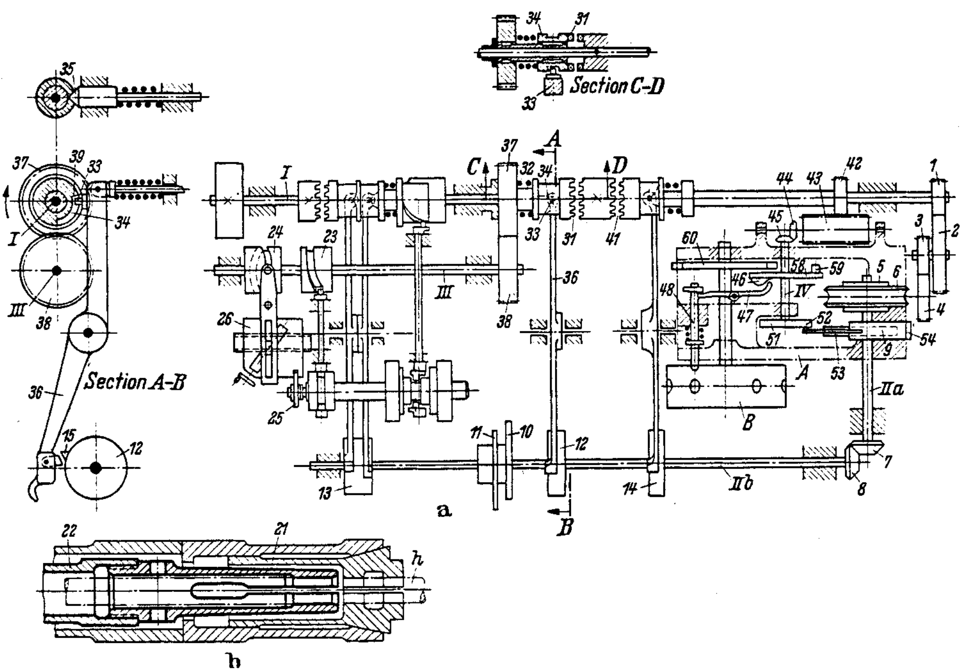


FIG. 263

of the various parts under strain and these deformations will then return to their original values as soon as the drive is disengaged, thus causing a return movement of the moving slide. As such a sudden return of the moving slide may result in damage to the machined surfaces, the highest possible stiffness of the whole drive is essential.

(iv) The complete separation of signal and energy transmission leads to the full exploitation of the possibilities offered by automatic control devices, as it not only permits an adaptation of the order of magnitude of the transmitted energy to the requirements of the operation, but also enables electrical, mechanical, hydraulic and pneumatic means to be applied in any desired combination. The first application of separate signal and energy transmission was perhaps realized in the mechanically controlled automatic lathe,[52] in which a master control shaft provides the signal and an auxiliary shaft the control energy. In this design, originated by Brown and Sharpe (1898) (Fig. 263), the master shaft makes one revolution per workpiece, exactly as in the case of the machine (Fig. 254). The energy is, however, provided by an auxiliary control shaft *I*, which runs at constant speed, independent of the production time required for the workpiece. The design is so arranged that the constant speed auxiliary shaft *I* also drives the variable speed master shaft *II*. The speed of shaft *II* must, however, be set to suit the machining times of different workpieces, and is changed by means of a set of pick-off gears (*1*, *2*, *3*, *4*), arranged between shaft *I* and worm drive *5*, *6*. By means of bevel gears *7*, *8* the master shaft *II* is divided into two parts (*IIa* and *IIb*), positioned at right angles to each other. Shaft *IIa* carries the feed cams *9* for the turret slide *A*, and shaft *IIb* carries the feed cams *10* and *11* for the cross slides and the "signal trip dog discs" *12*, *13* and *14* which initiate various control and switching operations, such as bar feed and clamping (disc *12*), change of spindle speed and direction of rotation (disc *13*), withdrawal, switching and forward movement of the turret head (disc *14*).

The manner in which the various switching operations are executed, may be shown for the example of the bar feed and the clamping of the bar material. The bar feed is operated by an inwardly sprung collet *21* (Fig. 263b). This collet is withdrawn through tube *22* by an amount equal to the required length of the bar feed movement. During this withdrawal the clamping collet (*a*, see Fig. 241c) is closed, so that the feed collet slides over the bar material without moving it. The clamping collet is then opened and the feed collet moved forward, thus pushing the bar material forward by the same amount. The stroke of the tube, which operates the clamping collet, is independent of the workpiece diameter and length and its movement is controlled by cam *23*. The stroke of the feed collet must be adjustable to suit the length of the workpiece and for this reason, the lever arrangement *26* with variable transmission ratio, is inserted between the operating cam *24* and the push rod for the feed collet *25*.

In order to obtain maximum switching speeds, the masses which have to be accelerated must be kept small. For this reason, the auxiliary shaft *I* runs continuously and the cams controlling different operations are coupled with it by means of clutches *31* only as and when required. Each of these clutches is preloaded by a spring *32* which engages the clutch as soon as the holding pin *33* is withdrawn from the groove *34*. In some designs the clutches are also held against any unwanted rotation by a stop pin *35*. As soon as the adjustable trip dog *15* on disc *12* (master shaft *IIb*) lifts the lever *36*, the pin *33* is withdrawn and the clutch engaged through spring *32*. The auxiliary shaft *I* then drives via clutch *31* and gears *37*, *38* the shaft *III* with cams *23* and *24*, which in turn operate the bar feed and the clamping operation. After about $\frac{3}{4}$ revolution of the clutch *31*, the stopping pin *33* is reset and thus disengages the clutch, at the same time preventing any overrun (face *39*).

The auxiliary shaft *I* usually runs at 120 rev/min, so that—except in special cases where particular precautions are taken—one switching operation is carried out in $\frac{1}{2}$ sec.

The separation of slow operational and rapid switching movements has also been introduced into the control of the turret head. The switching operation of the turret *B* on slide *A*, which is moved forward by the cam *9* on master shaft *IIa* (Fig. 263a), is started by a trip dog on the disc *14*. The shaft *IV* makes then a complete revolution driven by clutch *41*, gears *42*, *43* and bevel gears *44*, *45*.

The switching operation proper (rapid withdrawal, unlocking by means of cam *46*, lever *47* and index pin *48*, $\frac{1}{6}$ rev of the turret head, rapid forward movement into the working position), is produced by means of a crank drive and a Geneva mechanism.

The various steps during one complete switching operation can be seen in Fig. 264.

Position (*a*): The turret head *A* is in its farthest position forward from the previous cutting operation. Shaft *IV* begins to rotate.

Position (b): The crank drive *51*, *52*, *53* withdraws slide *A* to the right. The rack *54* moves in slide *A*. The spring *55* prevents lever *56* from lifting off cam *9*.

Position (*c*): The slide *A* is prevented from moving further back by stop *57*. As a result, the crank drive now lifts the feed lever *56* from cam *9*. The switching ($\frac{1}{6}$ turn) by means of the Geneva mechanism *58*, *59*, 60 begins.

Position (*d*): The slide *A* remains in its position. Lever *56* is lifted further off cam *9* and the switching operation *58*, 59, *60* continues.

Position (*e*): The switching operation is completed and lever *56* is moved back towards cam *9*.

Position (*f*): The lever *56* now rests on the next working cam *9* and the crank drive *51*, *52*

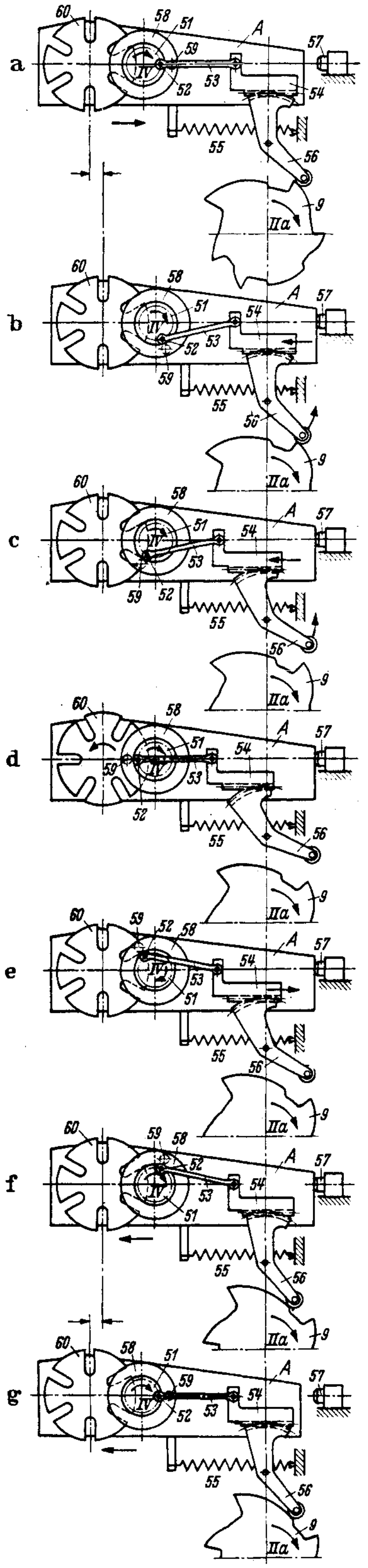


FIG. 264. (From G. SCHLESINGER, *Die Werkzeugmaschinen, loc. cit.*).

53 pushes the turret head *A* rapidly forward to the left. The rack *54* moves in slide *A*.

Position (*g*): The rapid forward movement is completed, the crank drive *51*, *52*, *53* is in its toggle position and therefore irreversible, the turret head is locked (index pin *48*, Fig. 263), and the feed movement (cam *9*, lever *56*, rack *54*, slide *A*), begins.

For operating hydraulic drives a preliminary device is often used for controlling the available hydraulic energy. Figure 265 shows such an arrangement. The hydraulically driven grinding machine table *1* operates by means of trip dogs *1a* or *1b*, a control lever *2*. This lever *2* transmits a signal by moving the valve spool *3* into one of its two extreme positions, resulting in an auxiliary oil stream being directed to one or the other side of the piston *4* which operates the main control valve. This piston *4* is rigidly connected with the main valve spool *5* and thus pushes the latter into one of its two extreme positions and determines the direction of the table movement.

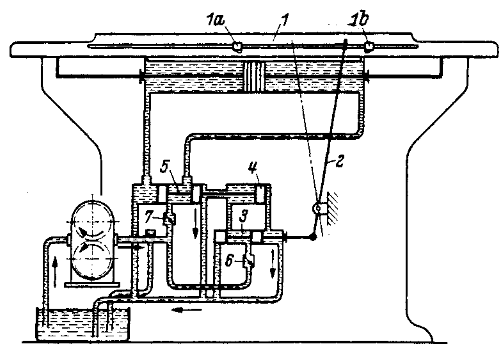


FIG. 265. Table drive with hydraulic control (German patent no. 342463).

Without providing the energy for the reversing operation the movement of the table controls it. This means that the reversing speed is independent of the speed and kinetic energy of the table, and even if the table remains momentarily stopped, this cannot prevent the reversing operation being completed.

By means of the throttling valve *6*, the reversing speed can be adjusted to such an extent that the table can dwell, i.e. remain stationary for a given time at the end of its strokes. This may be important, for instance, when "sparking out" is desirable. The throttle valve *7* serves for varying the table speed.

Other designs superimpose the preliminary and the main control actions in such a manner that, e.g. a rotational movement of the control valve spool serves for the preliminary operation. This in turn produces an axial displacement which controls the main operation. Many such designs, with and without pressure compensation for the control elements, have been described in detail.[44]

2. APPLICATIONS (EXAMPLES OF AUTOMATIC MACHINES)

Automatic machines can be divided into two groups:

1. Machines that are set up from time to time for a particular cycle of operations, which is repeated over and over again in accordance with the originally determined plan for as long as the required workpiece material is supplied;

2. Machines that are supplied with a "programme" of work in the form of a punched card, punched tape, magnetic tape or other means, and which execute any operation in accordance with that programme, without having to be especially "set up".

1. The machines mentioned under (1) are, so to speak, multi-purpose machines which can be converted into single purpose machines by means of adjustable stops, trip dogs, specially designed cams, adjustable tool holders, etc.

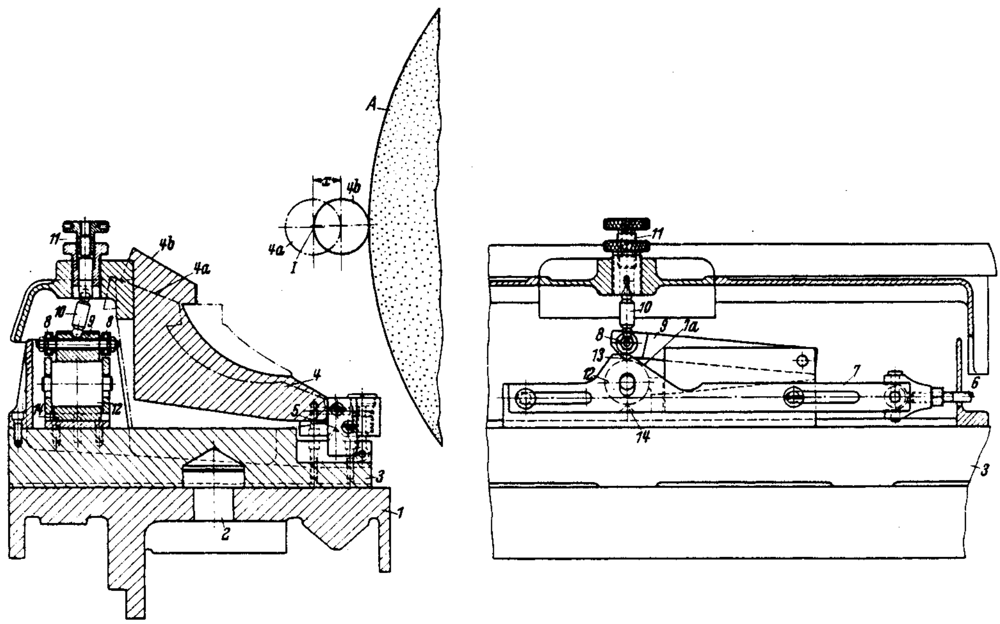


FIG. 266. "Fulcro-Sizer" cylindrical grinding machine (The Churchill Machine Tool Co. Ltd., Broadheath).

In general, the application of single purpose machines is economical only if a specific minimum number of identical workpieces has to be machined in one batch. A similar consideration applies to machines of the first group which includes, for instance, the previously mentioned automatic lathes, trip dog operated milling machines and others. Such machine tools also are often used as so-called "semi-automatic machines", when an operator is required for clamping and removing the workpiece and for checking the finished product from time to time although the actual machining sequence is maintained automatically. Two examples of this type are the grinding machines shown in Figs. 266 and 267, in which hydro-mechanical and hydro-pneumatic controls are employed.

The coarse depth setting for the grinding wheel *A* of the "Fulcro-Sizer" is obtained by the normal infeed movement of the wheel head. For fine adjustment, however, the workpiece headstock and the tailstock (workpiece axis *I*, Fig. 266), can be tilted on a horizontal axis, thus moving the workpiece axis towards the grinding wheel. The longitudinal slide *1* carries the table *3* which, for the purpose of taper grinding, can be swivelled round a spigot *2*. A carrier plate *4* on the table supports the workpiece headstock and the tailstock and can be tilted on knife edges *5*. The tilting movement (from position *4a* to *4b*) is hydraulically controlled via piston rod *6*, cam plate *7* with sloping surface *7a*, ball-bearing *8*, tilting lever *9* and connecting rod *10*, the latter being finely adjustable by the micrometer screw *11*.

The ultimate fine positioning for the grinding operation is obtained by the precision roller *12*, which is carried by cam plate *7* and rests between knife edges *13* and *14*, thus taking over the support of lever *9* from cam *7a*. Through the tilting movement of the carrier *4* from positions *4a* to *4b*, the workpiece axis *I* moves by an amount x towards the grinding wheel.

Once set the total stroke of the tilting movement is thus constant for all operations and it is, therefore, necessary for the operator to make the accurate adjustment of the relative position between grinding wheel and workpiece before each new batch of workpieces. This is done by means of the normal coarse infeed adjustment (to within approximately 0·0005 in.), and by fine adjustment of the micrometer screw *11* when the carrier is in its extreme tilted position (*4b*). When automatic operation is initiated, the final relative position between the workpiece axis and the grinding wheel is then consistently repeated by the carrier always being brought into the same position through the hydraulically operated tilting movement (accuracy 0·0001 in.).

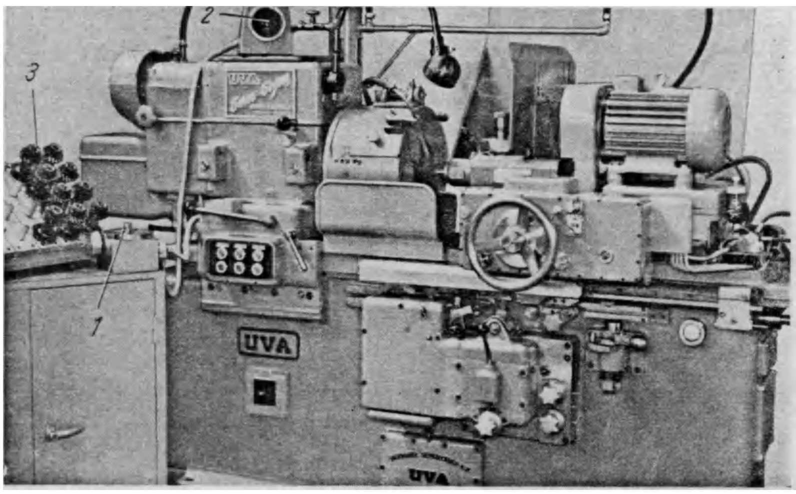


FIG. 267. Pneumatic measuring and control device for an internal grinding machine (Ulvsunda Verkstäder A.B., Bromma, Sweden).

In the "Fulcro-Sizer" grinding machine, the position of the workpiece carrier at the end of each operation cycle is ensured to be within small limits of the original setting. However, the accuracy of the ground workpiece can be guaranteed only if the position and size of the grinding wheel remain unchanged, because the ground diameter of the workpiece will increase with the wear of the grinding wheel, unless a corresponding correction of the final position of the workpiece carrier is made by means of the micrometer setting *11*. In other words, the machine operator has to check from time to time, whether the final size of the workpiece is still within the required limits and, if necessary, correct the end position of the workpiece carrier.*

On the internal grinding machine (Fig. 267), such a correction, when necessary, is carried out automatically. The dimension of the finished workpiece is checked and the result of this measurement used for controlling the correcting device. The operator measures the ground bore of a workpiece *3*, by means of a pneumatic gauge *1*, the measurement being indicated on the scale *2*. The use of a pneumatic measuring device eliminates the possibility of gauge wear. Deviations of the ground bore diameter from the specified value control the position of the dressing diamond within 0·00004 in. The dressing operation then produces an accurate grinding wheel diameter to which the wheel carrier setting can be adapted, thus ensuring the production of the correct bore diameter.

Copying and tracer controlled machines represent another type of machine belonging to the first group. Here a tracer, either located in the tool or workpiece carrier, follows the outline or surface

* An interesting automatic sizing device is described by E. ROTZOLL, *Industrie-Anz.*, 7 July, 1959.

of templates or masters, the control loop usually being closed by the actual position of the tool or workpiece carriers themselves.

Amongst the many and varied copying devices which follow either the outline of a template or the shape of a master, the two-dimensional electro-hydraulic and hydro-mechanical tracer controls may first be discussed. The method used differs basically from the direct control of a slide, such as encountered in the taper turning device (Fig. 255), in that the force acting between the guiding device (template or master) and the tracer is considerably smaller than the force required to move the slide, i.e. to overcome friction, inertia and cutting forces between the tool and the workpiece. As a result, specific pressure, friction and wear of the template are relatively low. Through the tracer the template or master provides a signal which controls the driving energy for moving the tool or workpiece carrier.[53]

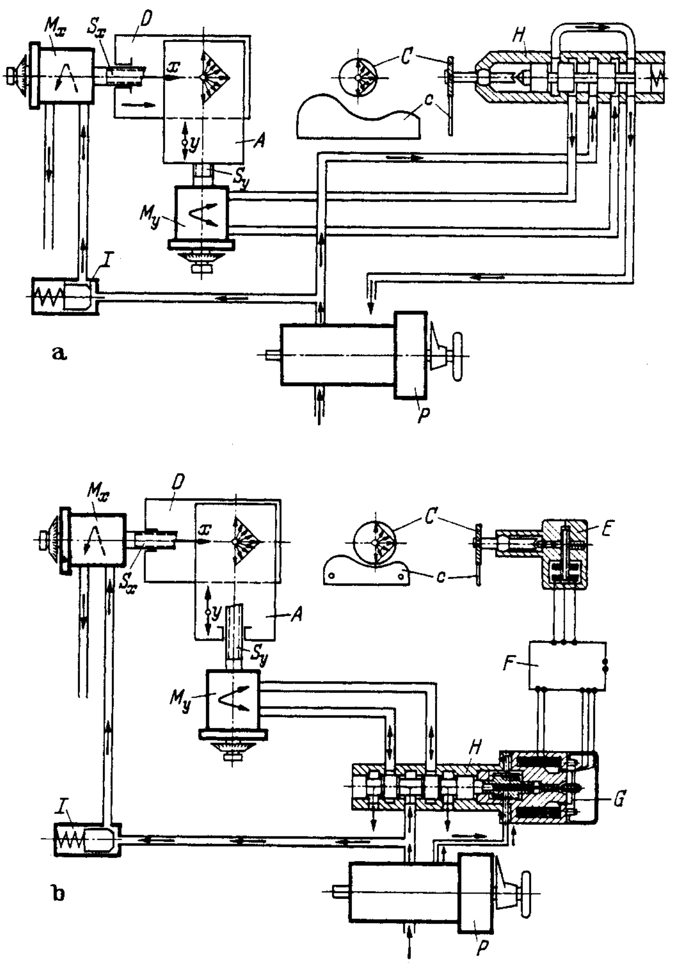


FIG. 268. Hydro-mechanical (a) and electro-hydraulic (b) tracer control for copy-milling (Gebr. Heller, Nürtingen, Germany).

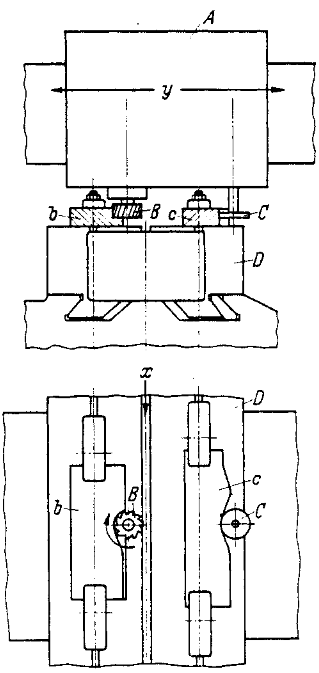


FIG. 269

Figure 268b shows the schematic layout of an electro-hydraulic, and Fig. 268a that of a hydro-mechanical tracer control for a milling machine table A which carries a workpiece b and a template c (see also Fig. 269). The slide D moves in the direction of the x-axis at a constant feed rate (hydraulic motor M_x and lead screw S_x). The movement of the table A in the direction of the y-axis (hydraulic motor M_y and lead screw S_y) is controlled by means of template c in relation to the position of milling cutter B and tracer roller C. It will also be possible (Fig. 269) to allocate the movement in the direction of the x-axis to the table D, workpiece b and template c and the movement in the direction of the y-axis to the spindle head A with the milling cutter B and the tracer roller C. If the diameters of the tracer roller and of the milling cutter are equal, the profile produced by the milling operation is the same as that of the template.

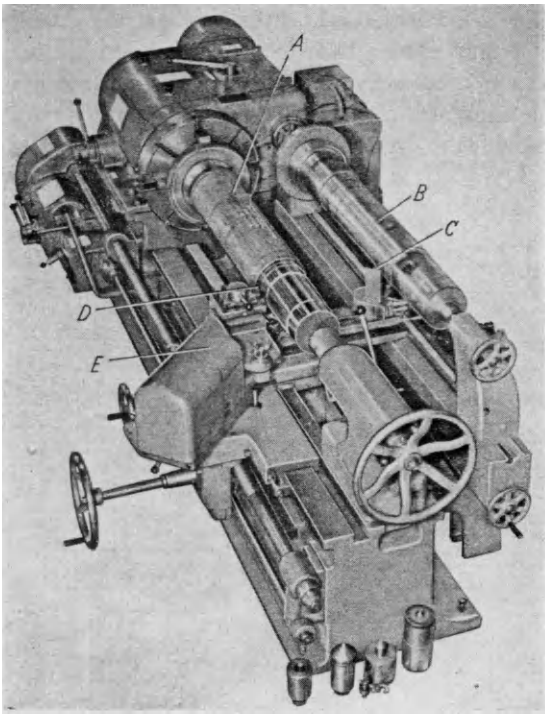


FIG. 270. Copying device for centre lathes (A.E.I. Ltd., Manchester).
A = workpiece.
B = sample.
C = tracer.
D = cutting tool.
E = copying slide.

Such tracer control devices are operated by means of the differences between the required (template) and the actual (moving slide) positions, which result in displacements of the tracer lever or the tracer roller from its mean position. The minimum difference (error) required for putting a control operation into action depends upon the forces and velocities involved, and is generally less than 0·002 in. The template c operates by means of tracer roller C a hydraulic valve H either directly (Fig. 268a) or via an electronic transducer E, an electronic amplifier F and a solenoid G (Fig. 268b). This valve H directs the oil supplied by a hydraulic pump P to the hydraulic motor M_y for either right- or left-hand rotation and thus produces the forward or reverse movement of the slide in the direction of the y-axis. The feed movement in the direction of the x-axis is produced by the oil supplied from pump P to motor M_x, the actual pressure being adjusted by the valve I.

Through the insertion of an electronic device between tracer and control valve, the mass of the tracer system is reduced and its natural frequency increased. This allows increased amplification and results in a higher working accuracy of the system, without danger to its stability.

Such two-dimensional hydraulic controls have been widely developed for lathes and similar machines. The profiles are turned usually

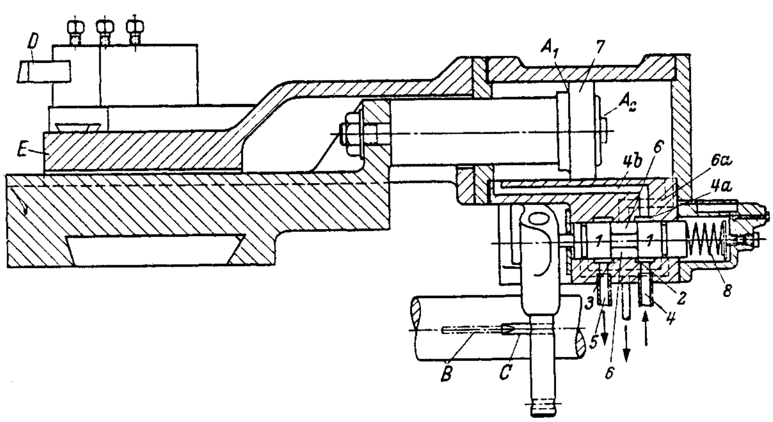


FIG. 271. Section through the copying device (Fig. 270).

with a constant longitudinal feed and the cross slide movement is controlled by means of template and transducer.*

Instead of using a specially made template, it is also possible first to produce a workpiece by manual operation of the lathe and to use this as a master for the automatic production of subsequent workpieces. Such a device is shown in Fig. 270. It is produced as a self-contained unit and can be applied easily to any normal centre lathe. As in most copying devices of this type, the cross traverse is not arranged at right angles to the turning axis, but at an angle of 45°. This makes it easier to control the tool carrier for the machining of shoulders and surfaces which lie at right angles to the turning axis. The copying slide E carries the hydraulic cylinder and moves relative to the piston which is rigidly connected to the cross slide (Fig. 271.) Both sides of the piston are under oil pressure, thus ensuring that the copying slide cannot jump forward or backward with a rapidly varying cutting force (see page 157).

The copying movement is controlled by valve spool *1* which in its centre position keeps channels *2* and *3* equally open. As a result, the oil (pressure p) which is supplied through pipe *4* flows to the distributing chamber *6* and through pipe *5* back to the tank. If the pressure in pipe *4* is p, and the pressure in pipe *5* is zero (atmospheric pressure), and if the throttling effect of channels *2* and *3* is equal, the pressure drops in *2* and *3* must also be equal, i.e. each must be $p/2$. In the middle position of the valve spool *1*, the pressure p_6 in the distributing chamber *6* is therefore, $p_6 = p/2$. The piston *7*, which moves the copying slide, is so dimensioned that the annular area A_1 on the left-hand side (determined by the diameter of piston and piston rod), is half the area A_2 of the full piston on the right-hand side, $A_2 = 2A_1$.

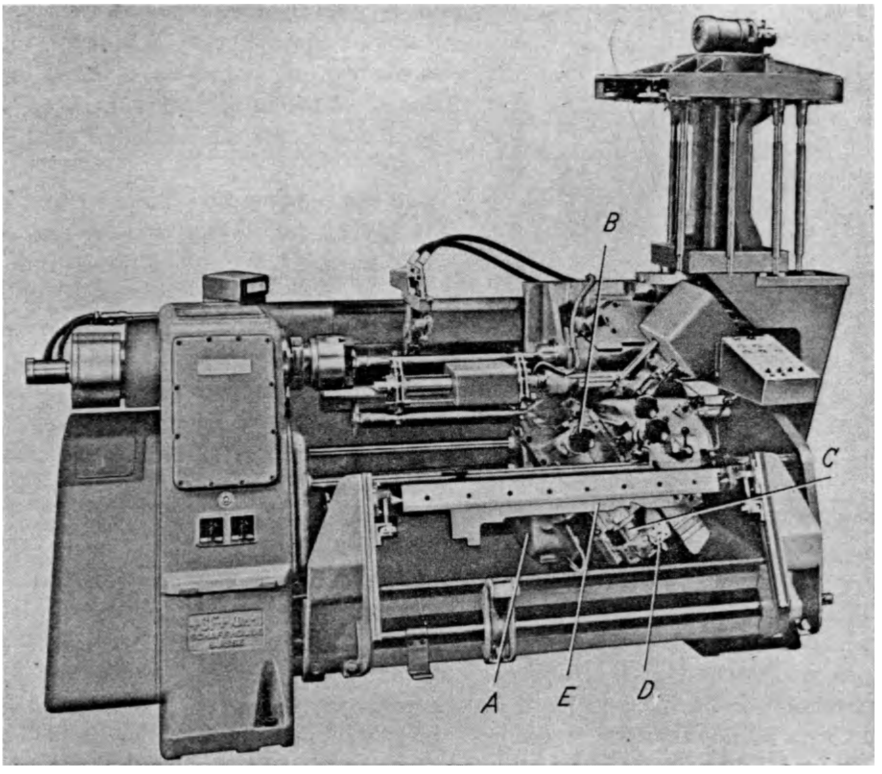


FIG. 272. Copying lathe with loading device (Georg Fischer A.G., Schaffhausen, Switzerland).

* A very detailed discussion and description of the design principles can be found in: E. SALJÉ, Nachformvorrichtungen an Drehbänken, *Konstruktion*, August, 1958. Design principles are discussed in: B. L. KOROBOCKIN, Zweckmäßige Auslegung der hydraulischen Kopiersysteme für Werkzeugmaschinen. *Stanki i instrument*, 1956, No. 6, reported by J. PEKLENIK, in *Industrie-Anz.*, 7 June, 1957.

The area A_1 is connected with pipe *4* via groove *4a* and bore *4b*, so that it is continuously under the full pressure p. The area A_2 is connected with chamber *6* via bore *6a* and is, therefore, always under the pressure p_6. When the valve spool *1* is in its middle position, the area A_2 is therefore under a pressure $p/2$ and the piston must be in equilibrium, because

$$p_6 \times A_2 = \frac{p}{2} \times 2A_1 = p \times A_1$$

When, during the longitudinal movement of the saddle, the tracer C is pushed back by the template or the master B, due to its diameter increasing, the valve spool *1* is pushed to the right against the action of the spring *8*, and the throttling effect of inlet *2* is reduced, whilst that of outlet *3* is increased. As a result, the pressure p_6 in chamber *6* and thus on piston area A_2 increases and the cylinder with the copying slide is moved to the right. In other words, the tool on the copying slide and the tracer are withdrawn from the workpiece until the valve spool is again in its middle position, i.e. when the turning diameter corresponds to the dimension of the template or master. Similarly, for the case of the master diameter decreasing, the valve spool *1* is pushed to the left by spring *8*, the pressure p_6 in the chamber *6* is reduced and the tool slide is pushed forward, resulting in a smaller turning diameter. It is important to remember that the pressure acting on the template is independent of the cutting resistance which acts on the tool and is in practice determined only by the preload of the spring *8*.

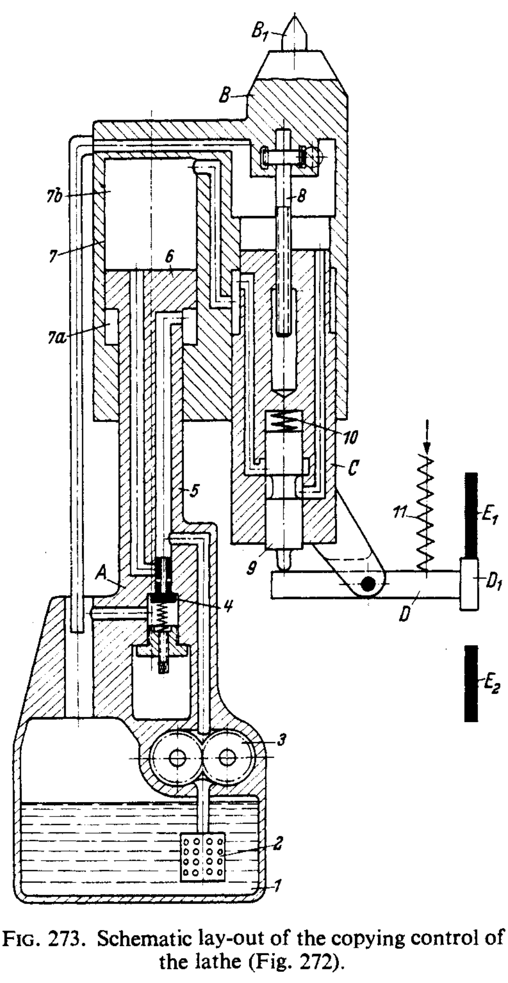


FIG. 273. Schematic lay-out of the copying control of the lathe (Fig. 272).

In the copying device (Fig. 270), the hydraulic unit (pump, filters, etc.) and the tank have to be connected to the hydraulic copying device by means of flexible pipes. This is avoided in the special copying lathe by Georg Fischer, Schaffhausen (Fig. 272). Here, the whole hydraulic equipment is incorporated in the saddle, thus eliminating the need for pipes and pipe connexions. The tool carrier is arranged below the turning axis. Incorporated in the saddle A are the oil tank *1*, filter *2*, constant delivery pump *3*, piston rod *5* with the pressure control valve *4* and piston *6* (Fig. 273). The tool carrier B with the cross traverse cylinder *7* slides on the fixed piston rod *5* and piston *6*. A valve housing C is movable in the tool carrier B, in such a manner that the distance between cutting edge B_1 and tracer lever D can be adjusted by means of the screw *8* to suit the depth of cut. The withdrawal side *7a* of the cylinder is always under the full pressure maintained by valve *4*, whilst the pressure at the other side *7b* depends upon the position of the valve spool *9*, which controls the outgoing oil flow.

In general, the valve spool *9* is pressed against the top template E_1 by spring *10*. If the diameter of the workpiece is to be increased, the template presses the tracer D_1 on lever D downwards and therefore lifts the valve spool *9*. This allows the oil flow through the control valve to increase and more oil can, therefore, escape from the cylinder side *7b* where the pressure drops. As a result, the tool carrier will be withdrawn from the turning axis which lies above it. If the turning diameter is to be decreased, the template E_1 allows the valve spool *9* to go down under the pressure of spring *10*, so that the pressure at the cylinder side *7b* increases and the tool carrier is pushed upwards towards the turning axis. By putting into operation the spring *11* which is normally kept out of action by a lever device and which can exert about twice the pressure of spring *10*, it is possible to make use of the lower template E_2 and to reverse the control action of the tracer. If this template allows the tracer roller D_1 to move downwards, spring *11* pushes the valve spool *9*, upwards, and so on. Through this arrangement the top template E_1 can be used for external operations, when the workpiece diameter increases with growing template size, whilst the bottom template E_2 can be used for internal turning operations, when the bore diameter decreases with increasing template profile. This results in the profile of the workpiece being a logical copy of the template (Fig. 274).

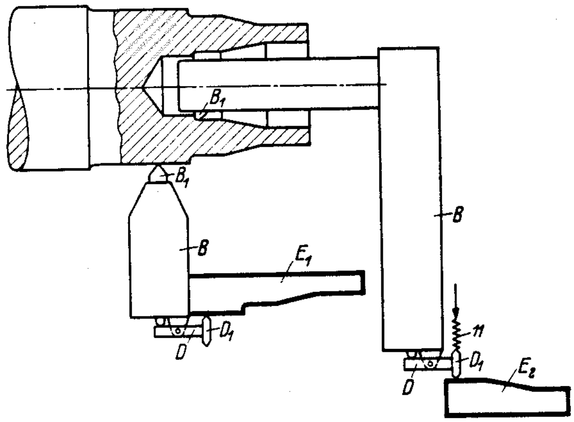


FIG. 274. Arrangements of the templates on the machine (Fig. 272).

Instead of the direct mechanical connexion between tracer and valve spool, an electrical signal transmission, as shown on page 187 and in Fig. 268b also can be used. In addition, it is possible to carry out the complete control operation solely by electrical or electronic means when the tracer, which moves under the action of a template or a master, controls electrical elements (contacts, relays, transducers, etc.), and thus starts, stops or reverses, various feed motors as required.[54] Because of the practically non-existent inertia of the control elements and the simplicity of signal transmission (electric conductors instead of flexible or telescopic hydraulic pipes), electrical control is often used not only for two- but also for three-dimensional copying devices, and especially for profile milling operations (Fig. 275).[50]

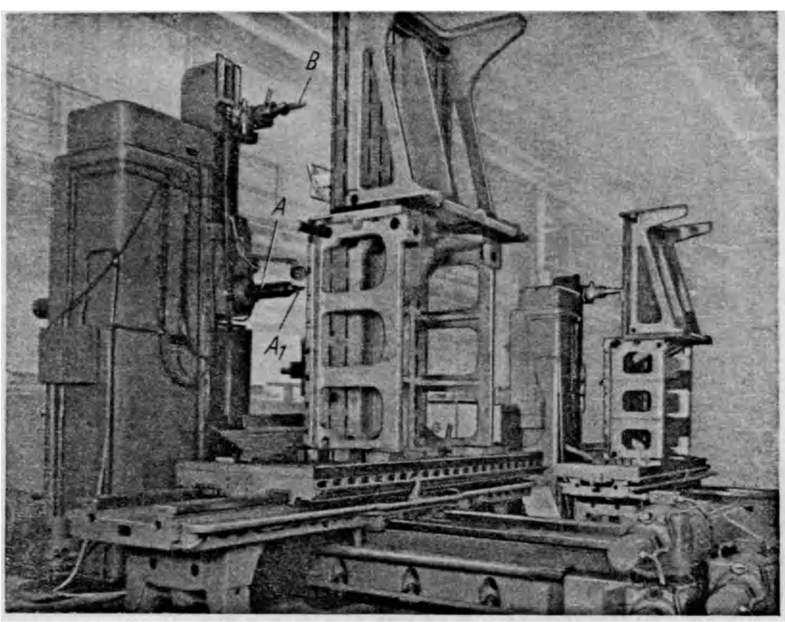


FIG. 275. Horizontal boring and milling machine with electronic tracer control (Giddings & Lewis, U.S.A., Electronic Control, A.E.I.). A = Milling spindle. A_1 = Milling cutter. B = Tracer.

The tracers for three-dimensional copying must produce independent signals for the different coordinates of the feed movements. Designs of such tracers have been described in many publications.[55] When the template or the master is exactly like the profile which has to be produced, the shape and size of the tracer must again be equal to that of the cutting tool, otherwise the profile of the template must be corrected to suit the conditions.

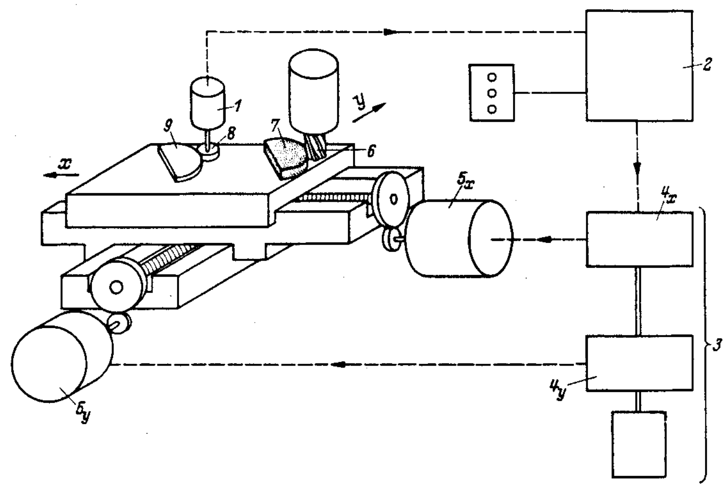


FIG. 276

In a two-dimensional control device developed by the British Thomson–Houston Company (A.E.I. Limited), the displacement of the tracer lever in the tracer head *1* produces, by means of two circuits, a.c. signals which are proportional to the displacement components in the direction of the two coordinates x and y. These signals control via electronic amplifiers *2* and a motor generator set *3* (Ward–Leonard unit with two generators, 4_x and 4_y), two variable speed d.c. motors 5_x and 5_y. The motors drive the corresponding feed mechanisms and thus produce the required instantaneous relative position between milling cutter *6* and workpiece *7* in accordance with the relative position between tracer *8* and template *9* (Fig. 276).

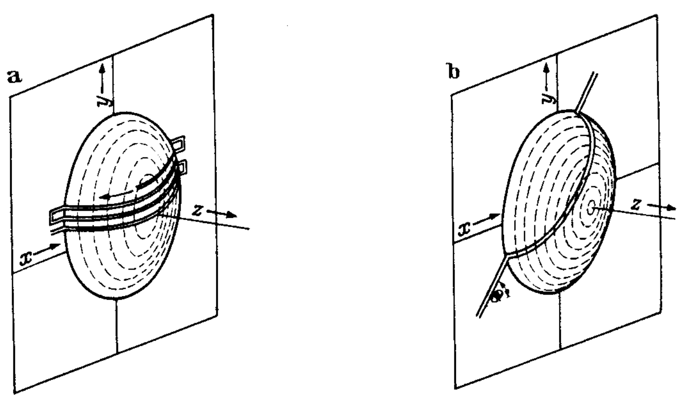


FIG. 277

Here again, the displacement of the tracer lever from its middle position is equal to the difference between required and actual positions of the workpiece table relative to the milling cutter. If the feed rate during milling is about 2 in/min, the error is usually less than 0·001 in. If an error of 0·002 in. is permissible, the feed rate may be increased to 5 in/min.

Three-dimensional copying can be achieved in two different ways. In schematic form Fig. 277a shows how a three-dimensional profile can be produced by controlling two dimensions directly from the master and producing the third at the end of each working stroke by a depth setting in the direction of the third coordinate.

However, it is sometimes undesirable, especially if a good finish of the machined surface is to be achieved, to limit the machining marks to two directions only of the coordinate system. Figure 277b shows again in schematic form the functioning of a control device working simultaneously in the directions of three coordinates. Control devices for three-dimensional copying require, of course, a considerable amount of additional equipment.

The following parameters affect the working accuracy of copying systems.[56]

(i) The static behaviour:

(a) the velocity amplification C, i.e. the increase in velocity of the working slide compared with the tracer deviation,

(b) the threshold U, i.e. that tracer deviation from the zero position which does not produce a movement of the working slide (Fig. 278a),

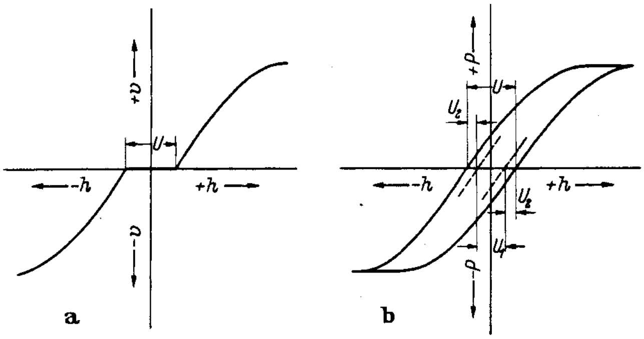


FIG. 278. h—Tracer deviation; v—Slide velocity; P—Driving force acting on the slide.

(c) the force amplification E, i.e. the increase in the operating force as a function of the tracer deviation.

(ii) The dynamic behaviour:

(a) the effect of pulsating cutting forces and of the masses which have to be accelerated,[57]

(b) friction effects.

The dynamic behaviour becomes more favourable the greater the amplification, because the damping of the control operation decreases until of course above a certain value instability occurs.

A detailed discussion of these points falls into the field of control engineering and is outside the scope of this book. However, it must not be overlooked that the parameters given above are not only dependent upon the design and quality of the control system, and its elements, but also upon the machine tool itself, because stiffness, natural frequencies of the machine elements, friction conditions (especially stick-slip) between moving parts and slideways, backlash in the driving elements, etc., reduce the efficiency and possible performance of the control equipment.[58]

For example, the threshold U, which can be recognized in the hysteresis curve (Fig. 278b), is composed of two parts, of which U_1 depends upon the stiffness and the backlash in the transmission elements, whilst the other, U_2, is determined by the friction forces which counteract the movement. U_2 is proportional to the friction force P_R and inversely proportional to the force amplification E.

$$|U_2| = \frac{P_R}{E}$$

The greater the amplification, the smaller the tracer deviations required to overcome the friction force P_R. Stiffness and freedom from backlash in the transmission elements (U_2) are also essential if the threshold, and therefore copying errors, are to be kept within small limits.[56] On the other hand, the amplification must not exceed a certain critical value if instability is to be avoided.

The force amplification E has a greater effect upon the accuracy of the system than the velocity amplification C.[56] In order to increase the stability of a system, it is preferable therefore to reduce C and to keep E as high as the stability of the system permits. The permissible minimum value for C is determined by the maximum permissible velocity error.

The accuracy of workpieces which are produced by a copying system cannot normally be greater than that of the template or the master. However, for the machining of small components and in order to increase the workpiece accuracy, it is possible to insert a pantograph between the tool carrier and the slide which is controlled by the template. This enables a template to be used, which is larger than the workpiece by the ratio of the pantograph amplification, and this means that not only the size but also the inaccuracies of the workpiece are reduced in relation to those of the template.

2. Only machines which work in accordance with a "programme" in the form of punched cards, punched tapes, magnetic tapes, etc., can be considered as being really fully automatic, because here the storage of instructions and the checking of the finished article are carried out independently

TABLE 18

Stored Value	Power of 2				
	4	3	2	1	0
0	0	0	0	0	0
1	0	0	0	0	1
2	0	0	0	1	0
3	0	0	0	1	1
4	0	0	1	0	0
5	0	0	1	0	1
6	0	0	1	1	0
7	0	0	1	1	1
8	0	1	0	0	0
9	0	1	0	0	1
10	0	1	0	1	0
11	0	1	0	1	1
12	0	1	1	0	0
13	0	1	1	0	1
14	0	1	1	1	0
15	0	1	1	1	1
16	1	0	0	0	0
17	1	0	0	0	1
18	1	0	0	1	0
19	1	0	0	1	1
20	1	0	1	0	0
21	1	0	1	0	1
22	1	0	1	1	0
23	1	0	1	1	1
24	1	1	0	0	0
25	1	1	0	0	1

of each other by different elements of a control loop. Whilst the starting, stopping and reversing of motors, the operation of clutches, etc., for controlling speed, direction of movement, etc. are relatively simple, the control of setting and feed movements in accordance with a programme determined by coordinates, in other words "numerical control",[59] is more difficult and will be discussed in greater detail.

Although the coordinate systems (polar coordinates, Cartesian coordinates, etc.), can be chosen to suit the prevailing conditions, the following examples refer to systems which work to Cartesian coordinates.

The information is usually *stored and transmitted* in accordance with the binary system, because this requires only two transmission signals (1 and 0, yes and no),[60] which can be produced, e.g. by closing or opening an electrical circuit. In the punched tape system, these signals are expressed by punching (1) or non-punching (0) of holes in the line which expresses the required power of the base figure. In the simple binary system, the number of decimals which can be stored is relatively limited, but in a decimal binary system, it is possible to store any values encountered in practice.

The binary system is based on the fact that any number can be expressed as the sum of powers of 2. For example:

$$1 = 2^0$$
$$2 = 2^1$$
$$3 = 2^1 + 2^0$$
$$4 = 2^2$$
$$5 = 2^2 + 2^0$$
$$6 = 2^2 + 2^1$$
$$7 = 2^2 + 2^1 + 2^0$$
$$8 = 2^3, \quad \text{etc.}$$

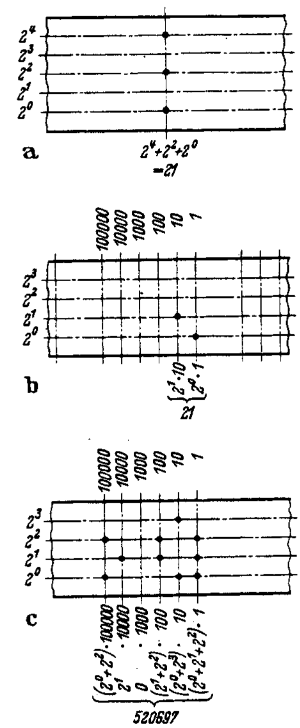


FIG. 279

By allocating a line of punched holes to each power of 2, any required value can be expressed by indicating whether (yes, 1) or not (no, 0) a power of 2 should be stored (Table 18).

In the binary system, therefore, one line corresponds to each complete value which is to be stored. In the decimal binary system, however, one line of punched holes is allocated to each decade (1, 10, 100, 1000, 10000, etc.), and specified in the binary system. With four powers of 2 (0, 1, 2, 3) and six lines of holes (1, 10, 100, 1000, 10000, 100000), it is possible to store values from zero to 999,999. For example, the value 21, which appears in the binary system as 10101 (Fig. 279a), is expressed in the decimal binary system as shown in Fig. 279b. A large value, e.g. 520,697, is expressed in the decimal binary system as shown in Fig. 279c.

On a punched card or a punched tape, considerably fewer signals can be stored than on magnetic devices (magnetic tape, magnetic disc, magnetic drum), and the length of information on drums and discs is more limited than, for instance, that on tapes. Another means of storing information is the photographic film, the optical signals being later transformed into electrical impulses.[61]

Brewer[59] makes an interesting comparison between the cost and storage capacity of various systems. However, not only the cost but also other considerations have to be taken into account, such as the special requirements of a control and measuring system applied in a particular case.

The storing device contains the *required* values of the coordinates which determine the instantaneous position to be produced by any setting or feed movement. These values are transformed into signals and transmitted to the control elements in such a manner that a corresponding movement is initiated. The *actual* position of the moving part at any moment is then determined by measuring devices which are, as far as possible, independent of the control mechanism and which send a corresponding signal (actual value) to the control equipment (feedback). The actual values of the coordinates are then compared with the stored values and any differences transformed into new signals which correct the position until the error signal is zero, i.e. required and actual values coincide within the prescribed limits. The accuracy of such systems again depends not only upon the design of the measuring and control mechanisms, but also upon the design of the machine tool (see page 193).

The devices and instruments employed measure either the increment of the path produced by a certain movement or the instantaneous position of the moving part in absolute values. In addition, they measure either rotational or rectilinear movements. In both cases either analogue or digital methods are applied. Whilst detailed descriptions* of the various systems would be outside the framework of this book, a brief description will be given of some of them.

Amongst the analogue methods for measuring rotational movements, simple potentiometers and inductive elements such as synchro-resolvers may be mentioned. They are small a.c. machines in which either the voltage induced in the rotor or its phase angle relative to a reference voltage is proportional to the angle of rotation. An extension of the resolver principle is the American "Inductosyn", in which two coils on two circular glass plates are inductively coupled.[62] The "Inductosyn" will be discussed later in greater detail as a measuring device for rectilinear movements.

FIG. 280. "Gray" disc.

In digital methods, discs with circumferential teeth are sometimes used. These transmit an impulse to an electronic counting device each time that a rotation by an amount equal to the tooth pitch has been completed.

In another method, the angle of rotation is measured by means of three contact discs of an analogue-digital converter.[63] These discs are geared to the shaft whose rotation is to be measured (gear ratios 1:1, 32:1 and 1024:1). This means that 1024 revolutions of the shaft can be measured with an accuracy of $\frac{1}{32}$ of a revolution. Figure 280 shows an example of such an unambiguously working "Gray" disc which, different from the previously mentioned methods, measures absolute values. In another digital method, use is made of the Moiré interference pattern produced by two superimposed optical gratings which are slightly inclined relative to each other. This method will be discussed in connexion with rectilinear measurements.

The measurement of angles of rotation for determining rectilinear movements requires the insertion of an intermediate member. For instance, the rectilinear movement of a slide driven by a screw and nut or by a rack and pinion can be measured by means of the angle of rotation of the screw or that of the pinion, if the pitch of the lead screw or the pitch circle diameter of the pinion respectively are known. In such cases, errors may be caused by the pitch error of the lead screw

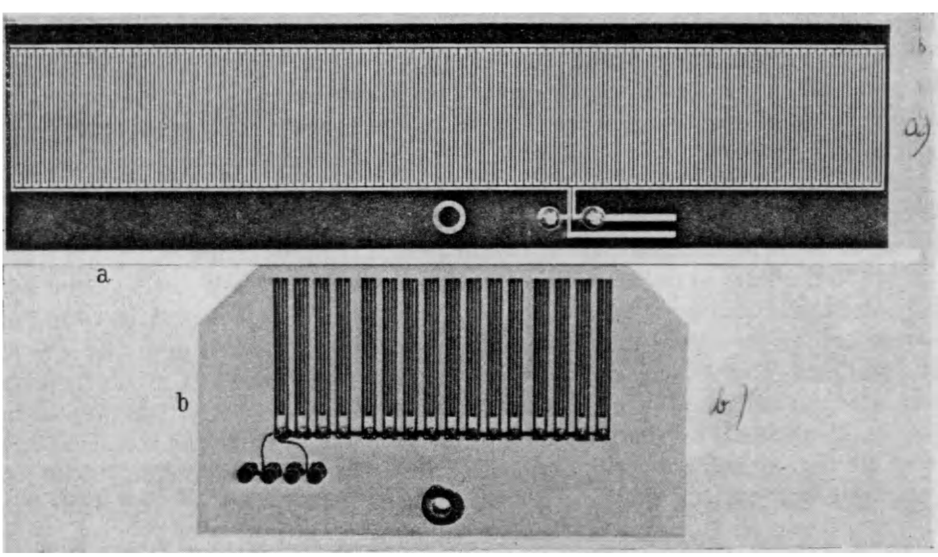


FIG. 281. "Inductosyn" (Farrand Controls Inc., New York, U.S.A.). *a*—Reference scale; *b*—Moving scale.

* See the many publications, some of which are quoted in connexion with this chapter.

or the pinion or by wear of these load carrying parts. Such errors will not occur if the rectilinear movement is measured directly or if the rotating element is not transmitting any working load but is employed exclusively for measuring the rectilinear movement.

The actual measurement or control of a rectilinear movement can be achieved either by comparing continuously with the control input the signal produced which corresponds to the distance traversed, or by displacing first the measuring instrument to the required amount in the opposite direction to the operational movement and then moving the working slide until the measuring instrument again reaches its zero position. A combination of both methods is also possible.

Amongst the analogue methods, the differential transformer plays an important part. Here, a sinusoidal voltage is generated by the axial displacement of a movable a.c. fed primary winding which is located inside a fixed secondary winding. One voltage cycle corresponds to the pitch of the helical winding, and the induced voltage becomes zero when both windings have been displaced by one-quarter of the pitch. By controlling or measuring accurately an additional rotation of the primary winding, it is possible to determine its axial position relative to the secondary winding with great precision.[59] (0·0001 in., if the pitch of the helix is 0·1 in.).

The linear "Inductosyn" of Farrand Controls Incorporated, New York, consists of two glass scales one of which, similarly to that of the rotational instrument (see page 196), carries the electronic reference scale. This reference scale is formed by a "hair comb" winding (Fig. 281a). The moving

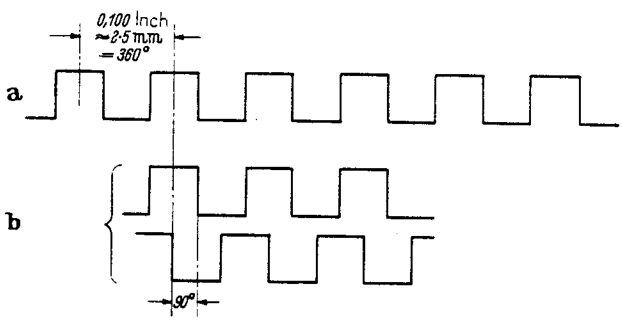


FIG. 282. a—Reference scale; b—Moving scale.

scale (Fig. 281b) carries two windings which are electrically shifted by 90° (Fig. 282). In general, the reference scale is located on the fixed machine part and the short moving scale fastened to the moving slide, the clearance between the scales being in the range 0·005 to 0·015 in. The tolerance in parallelism between the two scales is 0·002 in. The two windings are fed with audio-frequency voltages generated by the analogue control device, their amplitudes being proportional to the sine or cosine respectively of the displacement. As a result, a voltage is generated in the winding of the reference scale, and this voltage depends upon the ratio between the voltage in the slider windings and the relative displacement of the two scales (reference scale and slider). As long as the slider has not reached the required position relative to the reference scale, an error voltage is induced, which serves for driving the servo motor. As the pitch of the "Inductosyn" windings is 0·1 in. the induced voltage must go through zero after each movement of 0·05 in. It is, therefore, necessary to provide for a coarse setting between two scale divisions, for instance, by means of a synchro.

The accuracy of the "Inductosyn" is stated to be 0·0001 in. It is enhanced by the fact that 32 sets of poles on the slider simultaneously face the reference scale winding, so that the measurement is integrated over this distance and possible pitch errors of single windings are compensated for. The reference scales are produced in lengths of 10 in. and can be arranged in series over any distance.

Whilst the "Inductosyn" system works on an analogue principle and measures absolutely, the electro-optical Ferranti system[64] works on the digital principle and measures incremental movements. It consists of two optical gratings, one of which is fixed to the machine bed, the second being fastened to the moving part and slightly inclined to the first. These produce a Moiré interference pattern with a greatly enlarged distance between the lines (Figs. 283a to 283c). When one grating is moved relative to the other in a horizontal longitudinal direction, the horizontal dark lines move at right angles to this movement either upwards or downwards according to the direction of movement of the slides. By arranging a light source above and a photoelectric cell below the gratings, the amount of movement can be measured by electronically counting the number of interference lines which pass the photoelectric cell. The intensity of light which falls on to the photocell varies periodically during the longitudinal movement, one full period of variation in the

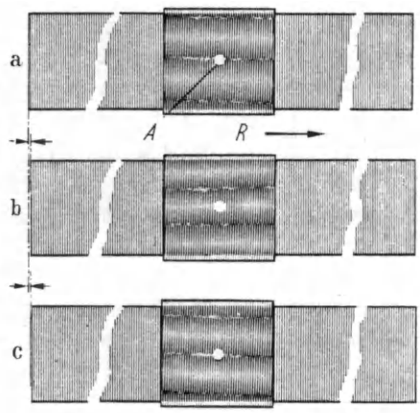


FIG. 283a–c. Optical grating.
A = Area covered by the photo cell.
R = Direction of movement.

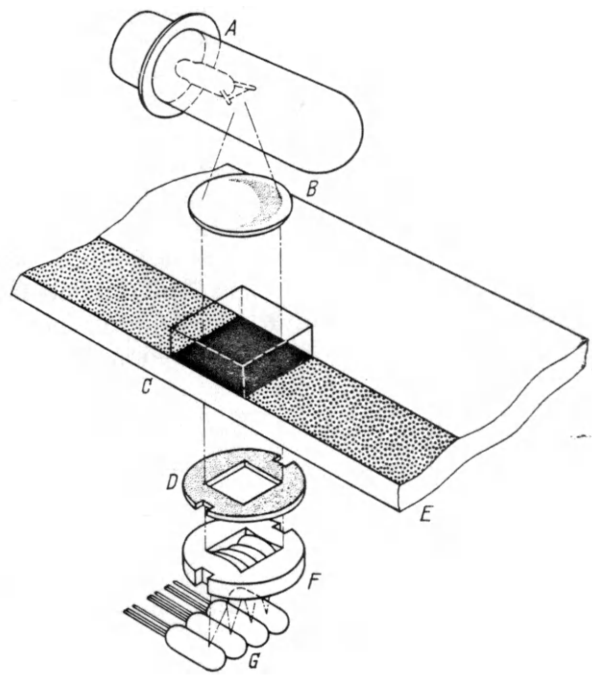


FIG. 283d. Schematic diagram of the Ferranti method (Ferranti Ltd., Edinburgh): *A*—Light source; *B*—Collimator Lens; *C*—Fixed grating; *D*—Light window; *E*—Grating fixed to the moving slide; *F*—Lens; *G*—Photocells.

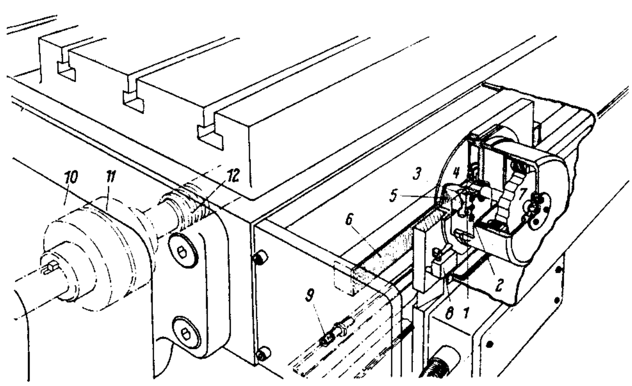


FIG. 283e. Electronically controlled table drive (Ferranti): *1*—Light source; *2*—Photocells; *3*—Lens system; *4*—Optical grating (fixed); *5*—Optical grating (rotating); *6*—Optical grating (fixed to moving table); *7*—Synchronous motor; *8*—Limit switch; *9*—Stop; *10*—Motor and gearbox; *11*—Coupling; *12*—Lead screw.

light intensity corresponding to a slide movement equal to the distance between the lines of the grating. By transforming these variations of intensity into electrical current pulses and transmitting these to an electronic counting device, the movement of the slide can be measured. In the original Ferranti method, four photo-electric cells are arranged in such a manner (Fig. 283d) that there is a phase shift of 90° between the amplitude intensity of the light falling on them, the 0° and 180° cells on the one hand and the 90° and 270° cells on the other being connected. This results in a two-phase electrical system, in which:

(a) the number of pulsations indicates the displacement;
(b) the frequency indicates the velocity, and
(c) the direction of the phase shift indicates the direction of the movement.

The gratings employed today usually have a line distance of 0·0002 in., and as the light intensity goes twice through zero during one period, one count occurs after each movement of 0·0001 in. In contrast to the "Inductosyn", the distance between the fixed and moving scales and their parallelism have to be kept within finer limits.

In a development of the Ferranti system (Fig. 283e), the transparent glass scale fixed to the moving part of the machine is replaced by a reflecting scale of stainless steel into which the grating lines are etched.[65] Above the lines of the steel scale, a transparent glass disc, with a spiral of black lines, rotates round a fixed axis. The radial distance between the spiral lines equals the distance of the grating lines on the steel scale, so that the steel scale reflects a Moiré pattern through the rotating glass disc. This pattern is again picked up by a photocell and transmitted in the form of signal impulses in exactly the same manner as in the case of the rectilinear movement of the gratings in the older Ferranti system. The intensity of the reflected light varies again in accordance with a sinusoidal law, the frequency depending upon the number of revolutions per minute of the glass disc and the pitch of the spiral. The glass disc is driven by a synchronous motor (104 rev/sec). As a result, a signal of 104 c/sec is transmitted from the photocell when the machine slide is stationary, and the Moiré pattern varies in the same manner as if a moving grating passes a stationary grating at a speed of 104 lines/sec. If the moving slide with the longitudinal grating moves longitudinally, a periodic variation of the Moiré pattern will result, the frequency differing from 104 c/sec in

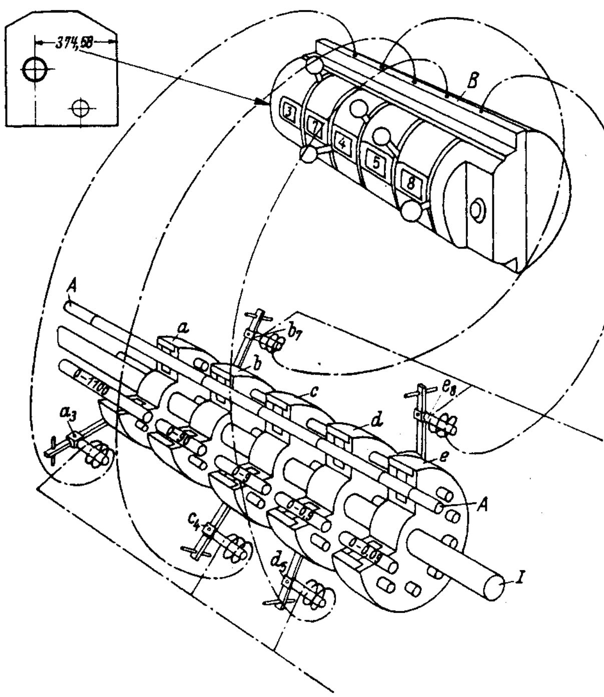


FIG. 284. Measuring system used in the "Schwartzkopff" jig borer.

accordance with the direction and speed of the movement. The difference in frequency is, therefore, a measure of the direction and speed of the movement, whilst the phase shift, i.e. the number of pulses, is determined by the magnitude of the movement. Gratings have been made with line distances of 0·01, 0·025 and 0·1 in. At 50 pulses per line spacing, movements of 0·0002, 0·0005 and 0·002 in. can be measured. In order to compensate for possible eccentricities of the rotating glass disc, a short reference grating can be arranged in the fixed housing and this measures any irregularities caused by eccentricity, correcting them by means of a suitable electronic circuit. This new Ferranti system enables the same measuring accuracy to be obtained as with the old one, but with much coarser and therefore cheaper gratings. A further development of an "interpolating system" using gratings, has come from the Research Department of the Staveley Group.[66]

Finally, two measuring methods may be mentioned, in which length scales are employed. In the Schwartzkopff jig boring machine, a reference scale based on a block gauge system (developed by Fosdick and Newall) is used (Fig. 284). Five drums (*a*, *b*, *c*, *d* and *e*) carry cylindrical bars, the lengths of which are stepped with the accuracy of slip gauges. The drums can be rotated by remote control and each measuring bar of a drum can thus be brought into the working position

A–A. The lengths of different bars in different drums can thus be combined with those of other bars in the other drums, in such a manner that the resulting sum of the lengths of all bars in the working position *A–A*, with their end faces in close contact, corresponds to the length of the required slide movement. The bars on each drum are stepped within one decade, the minimum length (zero bar) in each drum corresponding to a reference length "zero". The 12 bars in drum *a* are stepped from 0 to 1100 mm (i.e. minimum length to minimum length + 1100 mm), the 10 bars in drum *b* from 0 to 90 mm, in drum *c* 0 to 9 mm, in drum *d* 0 to 0·9 mm and in drum *e* 0 to 0·09 mm. It is, therefore, possible to obtain combinations between 0·00 to 1199·99 mm (47 in) in steps of 0·01 mm (0·0004 in.).

The setting and locking of the drums can be remotely controlled as follows. The drums are rotated by a friction drive on shaft *I* against rigid stops which prevent further rotation and hold them in a pre-determined position. On each drum, individual solenoid-operated stops are allocated to the measuring bars. In Fig. 284, only the stops for the measuring bars used in the example [i.e. numbers 3 in drum *a*, 7 in drum *b*, 4 in drum *c*, 5 in drum *d* and 8 in drum *e*] are shown. The

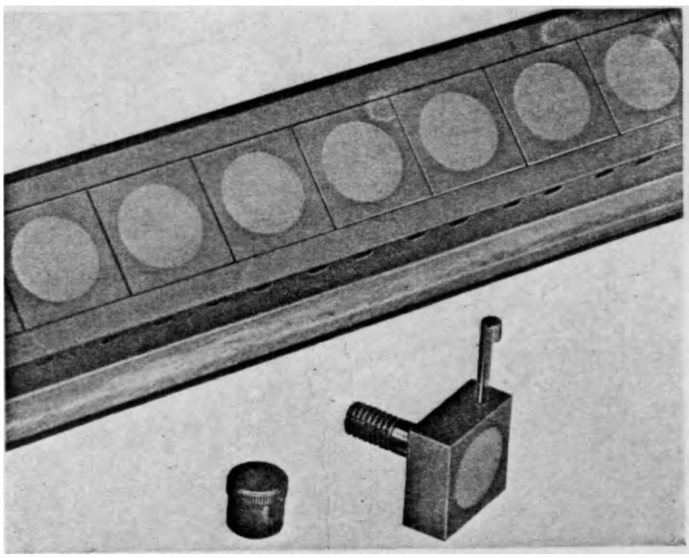


FIG. 285. Magnetic scale (A.E.I.).

solenoids are controlled from the switchboard (*B*, Fig. 284) on the machine. As soon as the solenoids, which correspond to the required setting, are energized, the central driving shaft returns at first all drums to their zero position, before moving them against the set of stops which are now in the operating position.

Whilst this drum system works on the digital principle, the magnetic scale of the British Thomson–Houston Company (A.E.I. Ltd.) is used for analogue systems. This scale (Fig. 285) carries steel blocks with bores which are spaced exactly by amounts of 1 in. (error not greater than 0·0002 in.). The exact position of these blocks can be adjusted very finely in the final assembly, by means of a setting wedge. In order to keep the surface of this measuring scale uninterrupted, the holes are filled with brass which is a non-magnetic material. The scale is fixed to the moving slide of the machine whilst an electromagnetic sensing head on the bed can be moved from its zero position in a short, very accurate guide, by means of a micrometer screw (pitch error less than 0·00002 in.). This movement serves for controlling by manual or punched card operation the decimal parts of the slide movement (0 to 0·9999 in.). The sensing head is a differential transformer, the magnetic flux of which goes through the parts of the measuring scale which are nearest to it. After the table has traversed a distance corresponding to the whole inch values of the required movement, the sensing head produces an error signal as long as it is out of line with the magnetic centre, determined by one of the holes in the measuring scale, by more than 0·00002 in. This error signal operates a servomechanism which in turn controls the table movement until the sensing head lies exactly opposite the magnetic centre.

Measuring systems and control devices have been developed in many parts of the world. It would be impossible to produce a complete survey within the framework of this book. However, some examples will serve to describe the basic principles on which they work.

The controls are divided into two groups these being in accordance with the purpose of the machine.

(a) Controls for Moving Slides into Defined, Fixed Positions

These are used in marking and drilling machines as well as in horizontal and jig boring machines; in other words, in machines, the slides of which have to be brought into position and clamped before the actual machining operation starts. Such machines are often economic even when single items or small jobs have to be machined and in particular, when a large number of holes is required in a component. In such cases, the time for marking out would be more costly and the possible rejects arising from faulty manual positioning, which are otherwise avoidable only by the use of expensive drill jigs, can be considerably reduced. The setting movements are not executed against the action of cutting forces, i.e. under more or less severe and possibly pulsating loads, and in this they differ from the controls mentioned later under (*b*). Moreover, the exact time at which the various setting movements (e.g. in the direction of the x, y and z axes) must be satisfactorily completed, is not critical, as long as care is taken that, for instance, in the case of drilling and boring machines, the table and the cross slide are clamped and the machining operation is started only when the moving parts have reached their final positions determined by the stored coordinate values. The accuracy of the velocity of movement and the synchronization of the different movements in the direction of the three coordinates, are not critical, and the control can be carried out and checked for each coordinate separately.

The information concerning the coordinates of the required slide positions can be transmitted to the control system by means of manually rotated knobs, or by punched cards or tapes inserted in a "reader".

The British Thomson–Houston system (Fig. 286) for fixed positioning is used by a number of machine tool manufacturers and is based on the magnetic scale (see page 200 and Fig. 285). The whole values (inches) of the displacement are covered by the synchro-controlled lead screw and the scale, whilst the decimal values are controlled by the sensing head, which is adjusted by means of the micrometer screw. In order to store the coordinates for a given table position, the transmitting synchros (S_1 to S_4) are either set by means of the six knobs (K_1 to K_6) for each coordinate (two values before and four behind the decimal point), or by means of a punched card and the "reader" R. Error signals are produced by the difference between the transmitting synchros S_2, S_3 and S_4 and the receiving synchros T_2, T_3 and T_4. They control via amplifier A the motor M_1 which displaces the sensing head F via gearbox D and micrometer screw E. In this way, the values behind the decimal point are determined by the displacement of the sensing head, whilst the values before the decimal point are determined by the control of the lead screw H. The transmitting synchro S_1 operates the receiving synchro T_1 and this is connected with the lead screw H through gear transmission I. The error signal due to the difference between S_1 and T_1 controls the motor M_2 via amplifier B and generator G, and the motor M_2 moves the slide O via gearbox U and lead screw H. When the moving slide is within $\frac{1}{4}$ in. of the required position, having approached this point with a speed of about 10 ft/min, the control is switched (relay L) to the sensing head F and amplifier C. By introducing a short time "false" signal, it is possible to approach the end position always in the same direction, thus eliminating the effect of backlash between screw and nut. The "false" signal is suitably switched on or off and the slide slowly moved until the hole in the magnetic scale N lies exactly opposite the sensing head F (see page 200). Difficulties which might be caused by stick-slip phenomena are reduced by suitable lubrication and the employment of stiff driving elements (see page 69). After the slide *O* is thus brought into the required position, it is clamped and ready for the machining operation.

Instead of using information on a punched tape or card, which has to be transformed into control impulses by readers (see *R*, Fig. 286), the design by L. Schwartzkopff (see page 199) uses punched cards which can be placed directly on a plug board. The holes in these cards allow the operator to insert only those plugs which rotate the measuring bar carrying drums into the required positions (see Fig. 284). Prior to reaching the final table position, the hydraulically controlled table movement is slowed down through solenoid operated valves, so that the movement against the stops is slow and smooth.

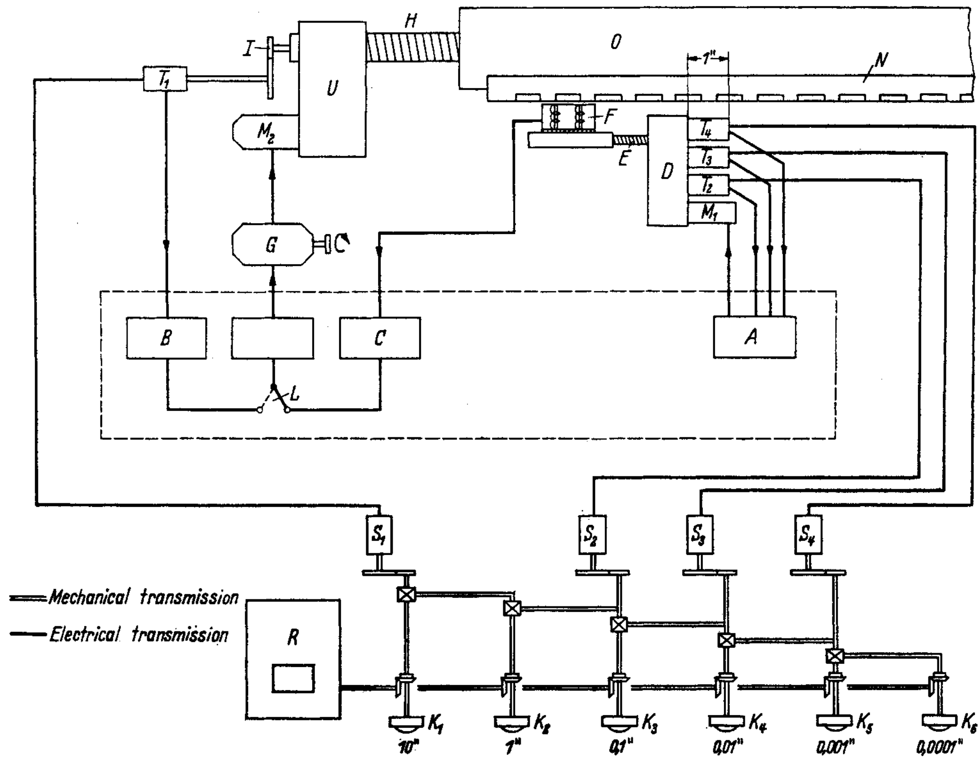


FIG. 286. "B.T.H." system (A.E.I.).

In order to compensate for inaccuracies in positioning the workpiece on the table of a jig-boring machine which works on the coordinate principle, provision is often made for the coordinate datum point to be displaced within a small range. In the Schwartzkopff machine the coordinate datum point can thus be moved anywhere within the given working range and any reference point of the workpiece, for instance, the centre line of a given bore can, therefore, be used without difficulty to serve as datum for the coordinate positioning operation.

With the B.T.H. system and the Schwartzkopff system, the operator has to initiate each coordinate positioning movement by hand. On a machine produced by the Société Genevoise (SIP), on the other hand, it is possible to store up to 30 coordinate values. Two magnetic drums are used as storage elements (one for coarse and the other for fine positioning) and the 30 coordinate settings can be controlled automatically between each boring operation and the subsequent one.[67] The operator stores the required values, by manually setting the table for machining the first workpiece, the coordinates of the various table positions being recorded on the magnetic drum through a push-button operated electronic device. After the first workpiece has been completed, any number of identical workpieces can then be machined in the same sequence, the information being obtained from the two magnetic drums. The position of the table is measured by means of a photocell microscope, the accuracy being of a very high order by virtue of the fact that over the last 20 mm, the velocity of the positioning movement is reduced in three steps, down to 0·1 mm/min.

The dividing accuracy of a high precision gear hobbing machine (Fig. 287)[68] is kept within ± 1 sec of arc and independent of the accuracy of the dividing wheel. Two radially arranged optical

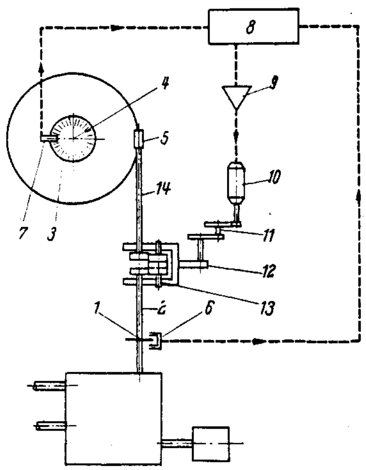


FIG. 287. Diagram of the pitch error correcting device for a gear hobbing machine (N.E.L.—David Brown).

1, *3* = Optical gratings.
2 = Worm driving shaft.
4 = Dividing wheel shaft.
5 = Worm.
6, *7* = Photocells.
8 = Phase comparator.
9 = Amplifier.
10 = Step motor.
11 = Gears.
12 = Eccentric cam.
13 = Epicyclic gear.
14 = Worm shaft.

gratings are used, grating *1* on the worm driving shaft *2* having 180 radial lines, grating *3* on the dividing wheel shaft *4* 21,600 radial lines. The ratio $\frac{180}{21600} = \frac{1}{120}$ corresponds to the transmission ratio between the single start worm *5* and the worm wheel with 120 teeth. As a result, the 180 lines of grating *1* pass the fixed photocell *6* with the same frequency with which the 21,600 lines of grating *3* pass a second photocell *7*. As long as the worm transmission and the dividing accuracy are without error, the signals of the two gratings are exactly synchronous. When an error occurs, a phase shift results and this controls via phase comparator *8* and amplifier *9*, a motor *10*, which in turn drives an epicyclic gear via wheels *11* and cam *12* in such a manner that an opposing phase shift is generated between the driving shaft *2* and the worm shaft *14*, thus correcting the error. The dividing accuracy thus depends only upon the accuracy of the gratings, without being affected by wear or other mechanical inaccuracies.

(b) Control of Feed Movements, for example, in Producing Profiles or Surfaces by Continuous Path Control

The control operations can cover two or three dimensions and the machine tool forms part of a closed control loop. The numerical control of profiling operations occurs under load and continuously, hence machines with type (*b*) control are different from the machines described under (*a*). The relative positions of the various moving slides have to satisfy the required conditions at any instant. Although the feed velocity as such is not critical, it is most important as far as the relative changes of position between the various slides are concerned. Moreover, the velocity resulting from movements in the direction of two or three coordinates should be kept, as far as possible, constant, in order to obtain a smooth surface. The control system must be designed in such a manner that all devices and drives work with the minimum possible delay. When the programme is worked out, the conditions determined by the controls and the drives of the machine must be considered and the accelerations required kept within obtainable limits. Moreover, requirements of high stiffness, low friction and limitation of moving masses must be strictly maintained if the machine is to satisfy the high accuracies and speeds which are obtainable with electronic controls.[69]

Stiffness of the machine tool is not only essential from the point of view of increasing the tool life and raising the quality of the finished product, but also from the point of view of ensuring a satisfactory performance of the servomechanisms. As the whole of the machine forms part of the control loop, it must be designed as such. The method sometimes encountered even today of combining any control device with any machine tool, will produce neither technically nor economically acceptable efficiency. In order to satisfy the above conditions, the following points in the design of the machine tools have to be observed.

1. *The machine tool as a whole.* Vibration behaviour, cutting forces, natural frequencies, tool life and tool wear must be considered from the point of view of working accuracy, surface quality and economy. An important point, for instance, is the fact that the performance of the servomechanism can be considerably affected by reducing or eliminating static friction in slideways and bearings.

2. *The feed drive.* The particular points which have to be considered in connexion with the electrical, electro-mechanical and electro-hydraulic drives generally used are: stiffness of the driving elements, reduction of frictional resistances and elimination of backlash. As far as possible, the effect of disturbances requires compensation. Such disturbances are: cutting forces and friction resistance, forces dependent upon acceleration (masses) and speed (friction, damping), and deformations of the control elements caused by operating forces and loading conditions. The time interval between input and output, i.e. between signal and control operation, is affected by these disturbances. It can be adjusted to suit the requirements by inserting delaying, integrating and differentiating elements in the control loop.

By establishing the relation between input and output, it is possible to analyse and check the working conditions. The first impulse comes usually from an "error signal". Consider that the error is ε and the amplification factor K_s. If a positional movement x is required, and an actual movement y only has taken place, then:

$$f(y) = K_s . \varepsilon \quad \text{and} \quad \varepsilon = x - y$$

The servomechanism has to reduce this error to as close to zero as possible. In the case of simple linear relationships, a differential equation can be established as follows:

$$m . \frac{d^2 y}{dt^2} + f . \frac{dy}{dt} + k . y = k . x$$

The effects of the stiffness of the control and driving elements k, of the velocity affected disturbances f and of the inertia forces m upon stiffness, frequency responses, etc., are clearly visible (see also page 193).

Within the framework of this book, a complete mathematical treatment of these questions is not possible. An incomplete treatment would be useless, especially in view of the fact that it would be wrong to over-emphasize the importance of the simple linear form given above which does not cover the non-linear characteristics of systems designed for highest accuracy requirements. Methods for establishing and solving the transfer function and for checking the sensitivity, accuracy and stability of the servomechanism, can be found in the available literature.*

The information can be transferred directly from the coordinates of the profile to be produced, as indicated on the drawing, on to the storage element (paper tape, etc.), or it can be "played" on to the storage as a recording by producing a simple workpiece (see page 202).[71] In the latter case, it is, of course, necessary for the machine to have completed any other preceding work before the new programme is recorded, in the same way as automatic lathes and turret lathes cannot be set up until their previous work is finished. In other words, the machine has to be withdrawn from the production programme for setting-up purposes. If the coordinates are transferred directly from the drawing to a paper tape, this can be done in an office without loss of productivity from the machine in the workshop.

In the case of a profile milling machine, the actual magnitudes of the various slide movements

* See references 58 and 70.

which are required for producing a given profile, depend upon the diameter of the cutter whose path relative to the workpiece must be determined. In one of the first numerically controlled milling machines,[72] this path was represented by a series of short straight lines, each joining two points which were determined by their coordinates. If this method is used to produce a profile within very fine limits the points, between which these straight line path elements are drawn, must be so close together that their number becomes large and the calculation of their coordinates very time-consuming. Instead of producing a profile as a series of straight lines, each lying between two points, it is also possible to create the profile from curved elements, each of which is determined by three points and generated by parabolic interpolation. Although with this method the coordinates of a relatively large number of points must again be calculated, this number is far less than in the earlier system.

Finally, it is possible to determine fairly accurately the coordinates for the exact path of the tool relative to the workpiece by means of an electronic computer, which is fed with the equations for the required curves together with the coordinates of two points, (usually the starting and finishing point of these curves). The coordinates for the movements of the workpiece and of tool carriers are transferred on to a magnetic tape, the tool diameter being automatically taken into consideration by the computer. The magnetic tape then contains the complete information for controlling the movements of the machine slides. In this case, it is not necessary to provide a separate, costly electronic computer for each numerically controlled machine tool, because it is possible to supply a number of machine tools with magnetic tapes from one central computer.

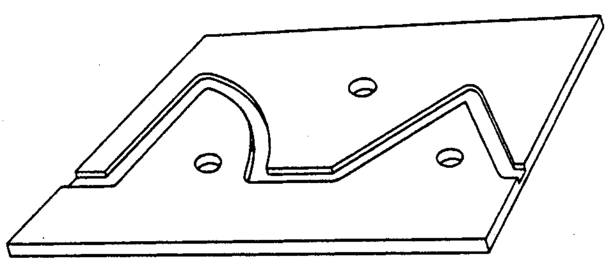


FIG. 288

The available control systems work either by analogue or digital methods. In the analogue system of the E.M.I.,* a punched paper tape is prepared by hand. The coordinate values of the profile which has to be produced can be corrected in accordance with the cutting tool diameter. The system is applicable to work with Cartesian or polar coordinates. If the use of an electronic computer is not envisaged, the coordinates are calculated for distances of about $\frac{1}{2}$ in. and punched into the paper tape, changes in direction of curvature being especially marked. If this work appears too time-consuming in the case of more complex curves, then an electronic computer should be used.

The control mechanism contains an interpolator in which not less than 1500 points can be interpolated between each set of three points originally established in the programme.[73] Small changes in the cutting tool diameter can be allowed for by suitable adjustment of the control device. In this system, acceleration and deceleration are not automatically included in the interpolation. They have to be considered, therefore, when the programme is prepared and must be added to the information on the punched tapes. This can be achieved, for instance, by decreasing the distances between the selected reference points in the case of deceleration and increasing them in the case of acceleration. In order to obtain smooth curves, it is advisable to select all change points and points of inflexion as reference points. The actual instantaneous slide positions are determined by means of the analogue system and therefore measured in absolute values. This is a considerable advantage because in the case of a faulty measurement by digital methods, if for instance the current pulses transmitted from an optical grating have to be counted and the counting mechanism fails, it would be necessary to return to the original datum point of the coordinate system and to recount. Figure 288 shows as a typical example a groove milled with the E.M.I. control system. Figure 289 shows the layout of the reference points which are determined by their coordinates, and Table 19 shows the programme on the basis of which the paper tape has to be punched.

The working accuracy depends greatly upon the type and design of the machine tool. The largest

* E.M.I. Electronics Ltd., Hayes.

error in profile milling on a small milling machine of standard design and controlled by the E.M.I. equipment was about 0·001 in.

In the two- or three-dimensional Ferranti system, the coordinates of the milling cutter path relative to the workpiece are calculated on an electronic computer and stored in digital form on a

TABLE 19

x	y	x	y	x	y
Start					
00250	03000	04197	06109	10200	05100
00500	03000	04609	05786	10500	05400
01000	03000	04970	05410	10750	05650
01400	03000	05280	04987	10950	05850
01700	03000	05525	04525	11050	05950
01900	03000	05704	04033	11100	06000
02050	03000	05768	03779	11150	05950
02100	03000	05829	03392	11250	05850
02100	03050	05840	03262	11450	05650
02100	03150	05848	03130	11700	05400
02100	03200	05850	03000	12000	05100
02100	03400	06050	03000	12300	04800
02100	03700	06250	03000	12600	04500
02100	04000	06500	03000	12900	04200
02100	04400	06800	03000	13200	03900
02100	04800	07150	03000	13500	03600
02100	05200	07450	03000	13750	03350
02100	05600	07700	03000	13950	03150
02100	05950	07900	03000	14050	03050
02100	06150	08050	03000	14100	03000
02100	06350	08100	03000	14150	03000
02100	06550	08150	03050	14250	03000
02100	06700	08250	03150	14450	03000
02100	06750	08450	03350	14750	03000
02230	06748	08700	03600	15000	03000
02492	06729	09000	03900	15000	03000
02651	06693	09330	04200	15000	03000
03258	06566	09600	04500	15000	03000
03744	06370	09900	04800	End.	

magnetic tape. The versatility of the electronic computer as a mathematical tool is exploited fully by calculating exactly the coordinates for curves which satisfy quadratic equations and by determining the coordinates of other curves by parabolic interpolation (see page 205). For this reason, with the Ferranti system it is only necessary to establish the change points and points of inflexion,

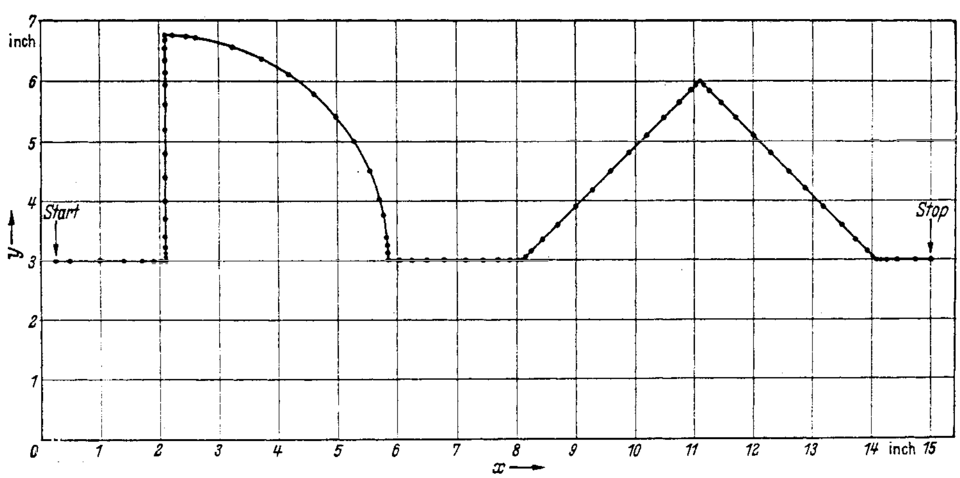
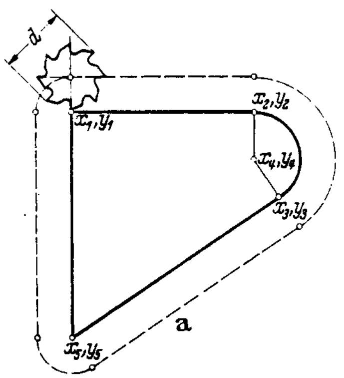


FIG. 289

to specify the curves lying between them and to code these on the punched tape which supplies the information to the computer. As already mentioned, in this case a single electronic computer can keep about 50 machine tools going. The distance, speed and direction of the slide movement are measured by means of optical gratings (see page 198 and Fig. 283), the slide speed being determined by the frequency of the control impulses. An impulse-frequency meter produces an analogue voltage, and this is compared with a voltage generated by an accurate tacho-generator which measures the slide speed. The difference is transmitted to the servomechanism through an amplifier. To this analogue voltage is added a voltage which is proportional to the difference between required and actual positions of the slide. This is generated unless the impulses from the magnetic tape are cancelled by those generated by the grating.



Cutter diameter d
Feed per minute s
Coordinates of the starting point x_1, y_1

Curve	Coordinates of the points of change	Coordinates of centres of curvature
Straight line	x_2, y_2	
Circle	x_3, y_3	x_4, y_4
Straight line	x_5, y_5	
Straight line	x_1, y_1	

b

FIG. 290

Figure 290a shows schematically a simple drawing of a profile to be produced on such a milling machine. Apart from the cutter diameter *d* and the required feed rate *s*, the coordinates of the points of change are indicated, together with the shapes of the curves which lie between two such points, for instance, straight line, circular arc, ellipse, parabola, etc. (Fig. 290b).[74]

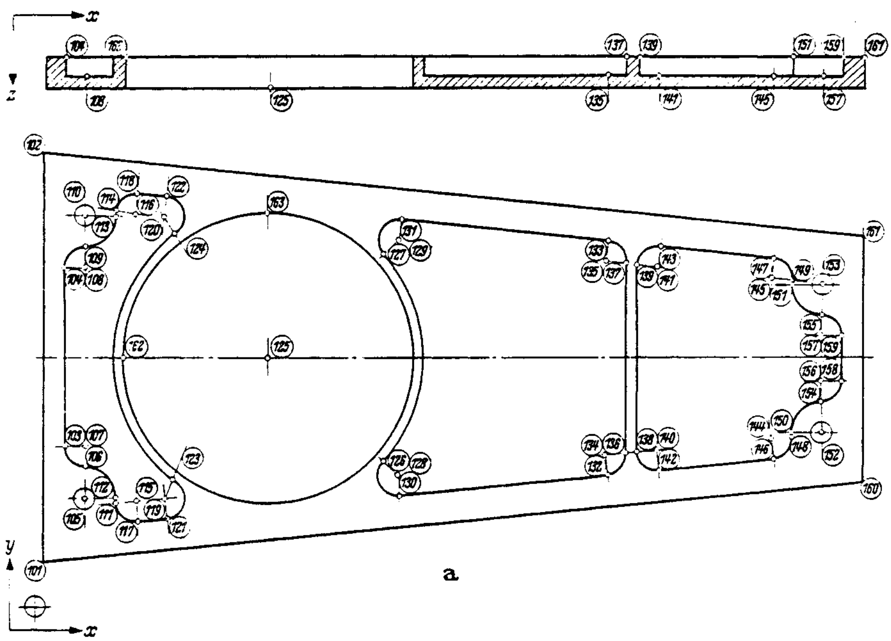
Figure 291a shows a complete workshop drawing which has been prepared three-dimensionally for use with a punched tape and an electronic computer.[74] All points are determined by their *x*, *y* and *z* coordinates. The datum point of the coordinate system has been chosen outside the workpiece in order to avoid negative coordinate values. Instead of entering all coordinate values in the drawing, it has been found more practicable to number the points of change and inflexion and to indicate their coordinates in a table adjacent to the drawing (Fig. 291b).

Perhaps the most advanced application of the Ferranti system is found in the special milling machine (Fig. 291c). The movable column weighing 9 tons has a longitudinal traverse of 25 ft. The vertical traverse of the headstock is 7 ft, and the axial displacement of the milling spindle 12 in. The gratings have 500 lines/in. and with two impulses per line (see page 198), the measuring accuracy is approximately 0·001 in. The gratings which measure the different movements are arranged in such a manner that in case of temperature changes they move with the casting to which they are fixed.

All movements are produced by hydraulic motors which are controlled by electro-hydraulic valves. The hydraulic motors for the longitudinal and vertical traverse are mounted on the moving parts and each slide is traversed by a gear drive in which two pinions are pressed in opposite directions under preload against the teeth of a rack fastened to the fixed part of the machine (see Fig. 210). The short axial traverse of the spindle is driven by a screw and recirculating ball nut system (see Fig. 203), two nuts being pre-loaded against each other in order to eliminate backlash.

Friction is practically eliminated by the use of hydrostatically lubricated slideways. The friction resistance between the heavy column and the bed at a traverse speed of about 10 in/min, has been found to be about 2 lb. The vertical clamping surface for the workpiece has an area 28 ft by 8 ft, and the maximum feed rate is 120 in/min.

An automatic boring and milling machine, which can be used not only in mass production but also for machining economically small quantities of workpieces, is the "Milwaukee-Matic" (Fig. 292a). The horizontal traverse of the table a_1 and the column a_2, the vertical setting of the spindle head b, the axial displacement of the spindle c, as well as the selection of a variety of cutting tools taken from a magazine d and transferred to and clamped in the spindle, are all controlled by a punched tape. The rotating tool magazine can carry 30 different cutting tools. For each change, one tool is taken out of the spindle and another one taken from the magazine, thus freeing one storage position. This means that altogether 31 different boring and milling tools can be used. It is not necessary for the tools to be arranged in a certain order in the magazine, as they are not selected for each operation on the basis of their position in the magazine, but in accordance with coding rings on the tool holders.



a

Points of change	Coordinates		
	x	y	z
101	1·7500	1·000	0·7500
102	1·7500	6·000	0·7500
103	2·000	2·4062	0·3750
104	2·000	4·5938	0·3750
105	2·2500	1·7812	0·3750
106	2·2500	2·1562	0·3750
107	2·2500	2·4062	0·6250
108	2·2500	4·5938	0·6250
109	2·2500	4·8438	0·3750
110	2·2500	5·2188	0·3750
111	2·6250	1·7406	0·3750
112	2·6250	1·7812	0·3750
113	2·6250	5·2188	0·3750
114	2·6250	5·2594	0·3750
115	2·8750	1·7406	0·6250
116	2·8750	5·2594	0·6250
117	2·8999	1·4918	0·3750
118	2·8999	5·0106	0·3750
119	3·2542	1·7786	0·6250
120	3·2542	5·2214	0·6250
121	3·2791	1·5298	0·3750
122	3·2791	5·4702	0·3750
123	3·4008	1·9811	0·3750
124	3·4088	5·0188	0·3750
125	4·5000	3·5000	0·7500
126	5·8782	2·2289	0·3750
127	5·8782	4·7711	0·3750
128	6·0620	2·0594	0·6250
129	5·0620	4·9406	0·6250
130	6·0869	1·8106	0·3750
131	6·0869	5·1894	0·3750
132	8·6499	2·0668	0·3750
133	8·6499	4·9332	0·3750
134	8·6250	2·3156	0·6250
135	8·6250	4·6844	0·6250
136	8·8750	2·3156	0·3750
137	8·8750	4·6844	0·3750
138	9·000	2·3782	0·3750
139	9·000	4·6218	0·3750
140	9·2500	2·3782	0·6250
141	9·2500	4·6218	0·6250
142	9·2749	2·1294	0·3750
143	9·2749	4·8706	0·3750
144	10·6250	2·5156	0·6250
145	10·6250	4·4844	0·6250
146	10·6499	2·2668	0·3750
147	10·6499	4·7332	0·3750
148	10·8750	2·5156	0·3750
149	10·8750	4·4844	0·3750
150	10·8750	2·5938	0·3750
151	10·8750	4·4062	0·3750
152	11·2500	2·5938	0·3750
153	11·2500	4·4062	0·3750
154	11·2500	2·9688	0·3750
155	11·2500	4·0312	0·3750
156	11·2500	3·2188	0·6250
157	11·2500	3·7812	0·6250
158	11·5000	3·2188	0·3750
159	11·5000	3·7812	0·3750
160	11·7500	2·0000	0·7500
161	11·7500	5·0000	0·7500
162	2·7500	3·5000	0·3750
163	4·5000	5·2500	0·3750

b

FIG. 291

The defining of the different dimensions for planning and coding is shown in Fig. 292b. It is even possible to machine alternately two different workpieces by means of two suitably punched tapes. As shown in Fig. 292c, this requires the use of two "readers". The operator inserts tape A into the left-hand reader, whereupon workpiece A, which is clamped to the left-hand table, is machined. During this operation, the operator inserts punched tape B in the right-hand reader and clamps workpiece B to the right-hand table. After completion of workpiece A, the left-hand table is moved away from the spindle to the left and the right-hand table with workpiece B is brought in front of the machine spindle and the machining operation started. If, after the machining of B another workpiece A is to be machined, the operator removes the finished machined workpiece A from the left-hand table and clamps a new one in position. During the machining of workpiece B, the tape in reader A has been brought back to its initial position and the machine is now ready for machining another workpiece A as soon as the machining of workpiece B has been completed.

The speed change for the spindle drive is also tape controlled and covers a range of 100 to 4000 rev/min. The table feed is driven by hydraulic servo motors via lead screws and pre-loaded recirculating ball nuts. The tables are guided on pre-loaded ball bearing slides.

The automatic devices which have been described, serve for controlling the working movements of tool or workpiece carriers (spindle head, table, slide, etc.), in accordance with the requirements specified on a drawing, a working programme, a template or a master. If the actual movement of

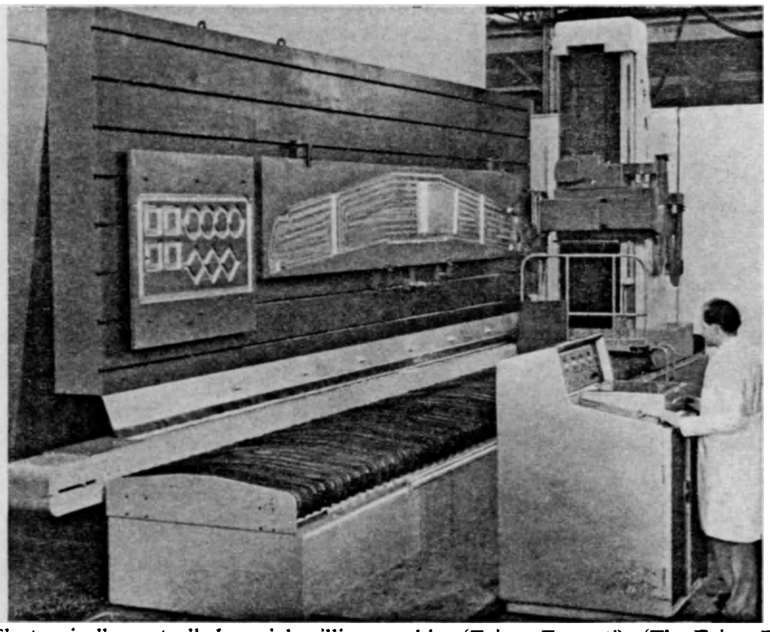


FIG. 291c. Electronically controlled special milling machine (Fairey–Ferranti). (The Fairey Engineering Co. Ltd., Stockport).

these parts is measured by independent instruments, compared with that required by the programme and corrected in case of differences, the accuracies of the various movements will only depend upon the type and quality of the measuring and control equipment.

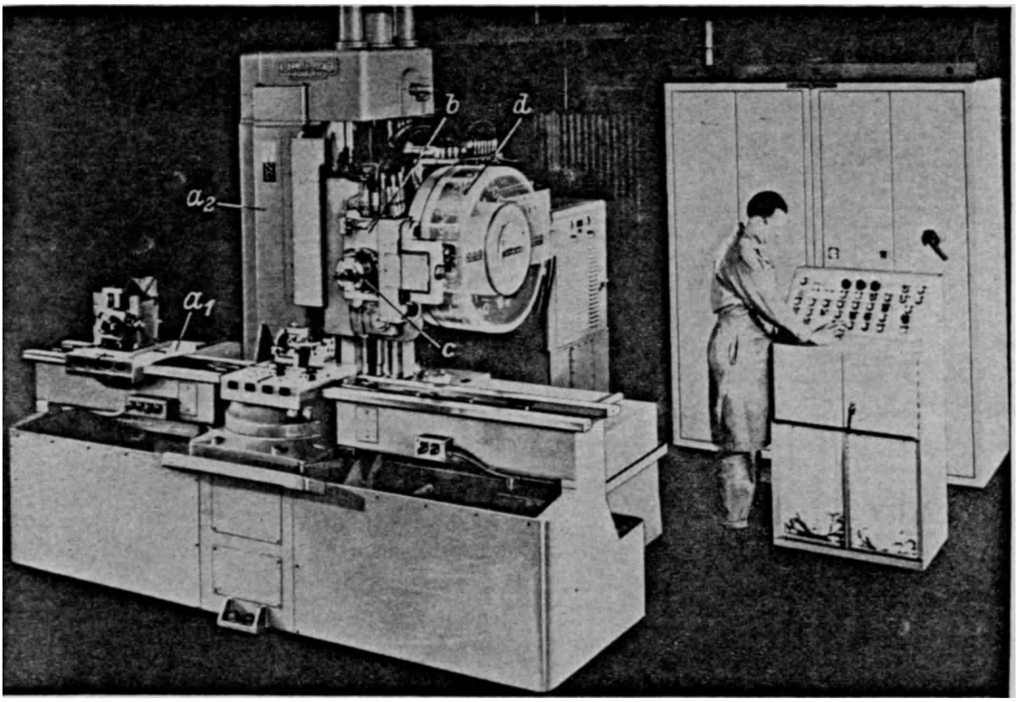


FIG. 292a. "Milwaukee-Matic" milling machine (Kearney & Trecker, Milwaukee, U.S.A.).

The final operational accuracy, however, not only depends upon the accuracy of the setting and feed movements, but also upon the accuracy with which the moving parts are guided. All this

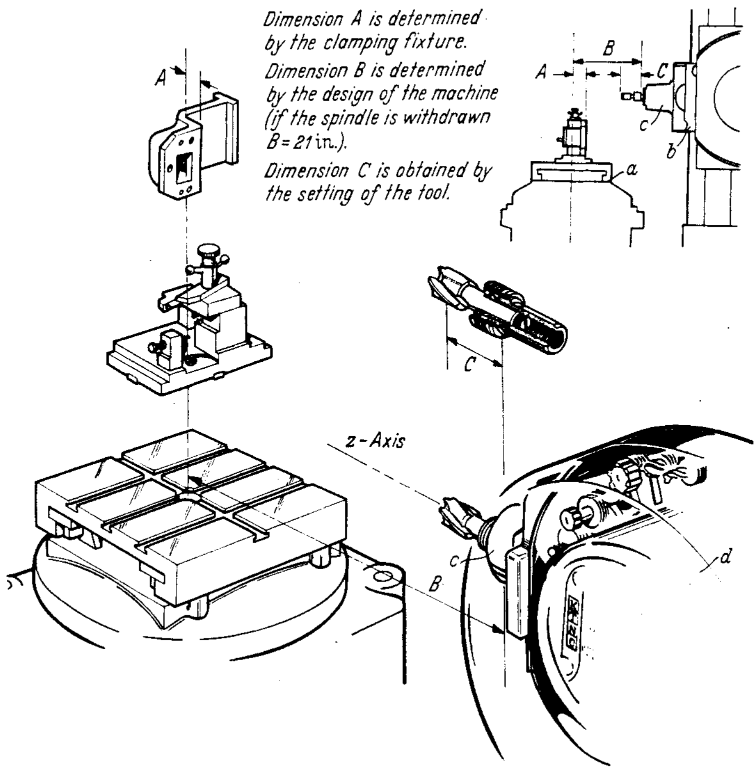


FIG. 292b. Determination of the setting dimensions for the machine (Fig. 292a).

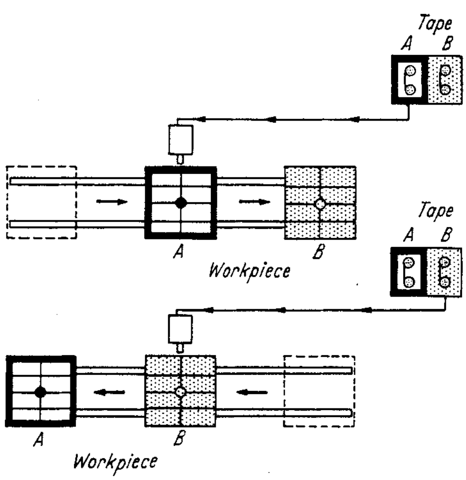


FIG. 292c

equipment measures with great accuracy the length of setting and feed movements, i.e. displacements in the direction of the working movements, but it does not cover any displacements at right angles to the working movements. Such displacements may be caused by manufacturing inaccuracies of slideways, by play between guiding and guided elements, by wear, etc. These are usually checked by means of acceptance tests* and, therefore, kept within certain limits which until recent times corresponded to the order of magnitude of setting and feed accuracies. With the introduction of optical and electronic control devices, however, this order of magnitude has been considerably reduced and a corresponding reduction of the permissible manufacturing limits for machine tools might become uneconomical in many cases. It has been suggested,[68] therefore, that the high accuracy obtainable with electronically controlled machines be ensured by means other than the increased accuracy of the slideways, guides, etc., in other words, avoiding the necessary reduction in tolerances of straightness and parallelism, by measuring continuously the instantaneous deviations of the slide movements from their theoretically required path, to transmit electronically the results of these measurements to the control mechanism, and to correct the control signals in such a manner

* Schlesinger's *Testing Machine Tools* was first published in 1927.

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

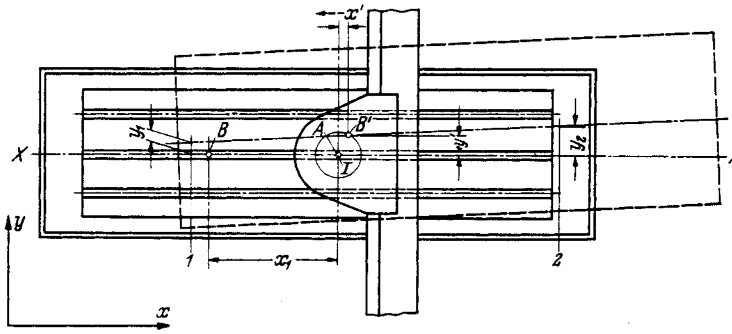


FIG. 293

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 *I* 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

TABLE 20

CALCULATED ERRORS OF A PROFILING MACHINE

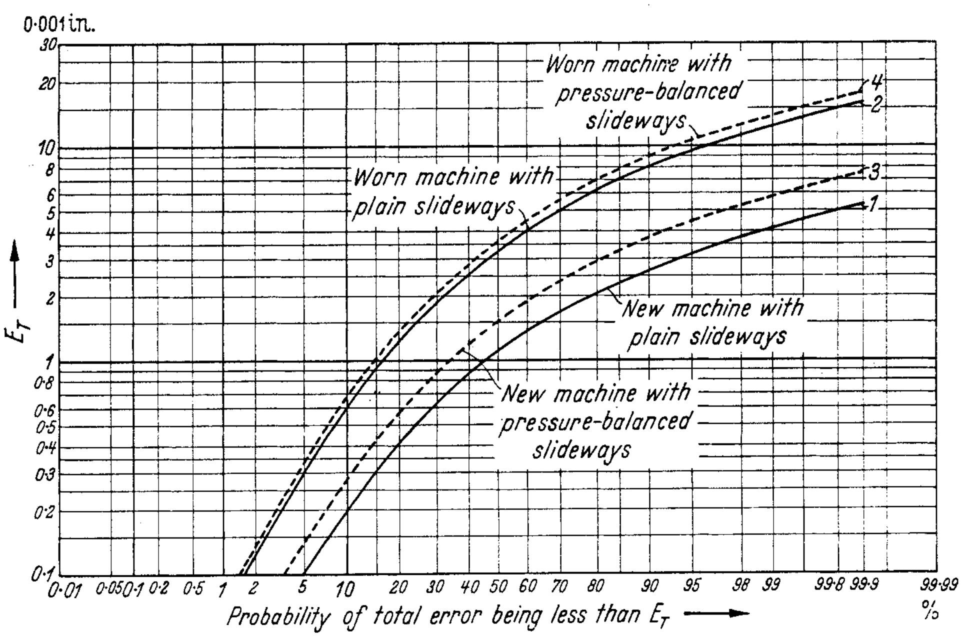
Table traverse : 4·0 ft
Cross ,, : 2·0 ft
Vertical ,, : 0·6 + 0·6 ft
Alignment Limits: 0·0008 in/ft

Machine Component and its Misalignment		Maximum Error (0·001 in.)		
		Longitudinal axis = $\pm E_x$	Cross axis = $\pm E_y$	Vertical axis = $\pm E_z$
Table	Yaw	1·61	3·22	0
	Roll	0	0·96	1·61
	Pitch	0·96	0	3·22
Cross-slide	Yaw	1·61	0·54	0
	Roll	0·96	0	0·40
	Pitch	0	0·96	1·61
Spindle-slide	Yaw	0·96	0	0·32
	Roll	0·30	0·30	0
	Pitch	0	0·96	0·30
Spindle	Tilt/Lift	0·25	0·25	0·22
Sum of Maximum Errors		6·65	7·19	7·68
Total Error = 0·7 × sum		4·65	5·03	5·37

Overall Error $E_o = \sqrt{E_x^2 + E_y^2 + E_z^2}$

$\therefore$ Maximum $E_o = \sqrt{4{\cdot}65^2 + 5{\cdot}03^2 + 5{\cdot}37^2} = 0{\cdot}0087$ in.

NOTE: All the maximum values of the errors along each axis cannot occur simultaneously; the factor 0·7 is the geometric mean between the sum of half maximum errors, which can occur simultaneously, and the sum of full maximum errors. Owing to differences in sign, there will be partial self-cancellation amongst the 7 components along each axis.



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.

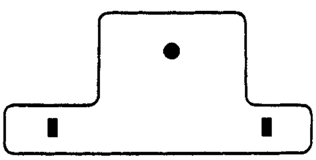


FIG. 294

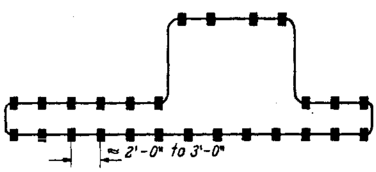


FIG. 295

(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 ($\frac{15}{16}$ in. dia.) for the tension screw.

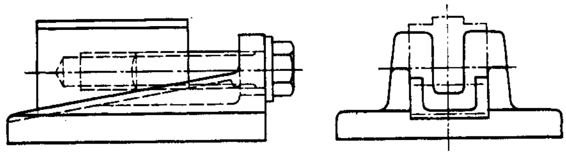


FIG. 296

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.

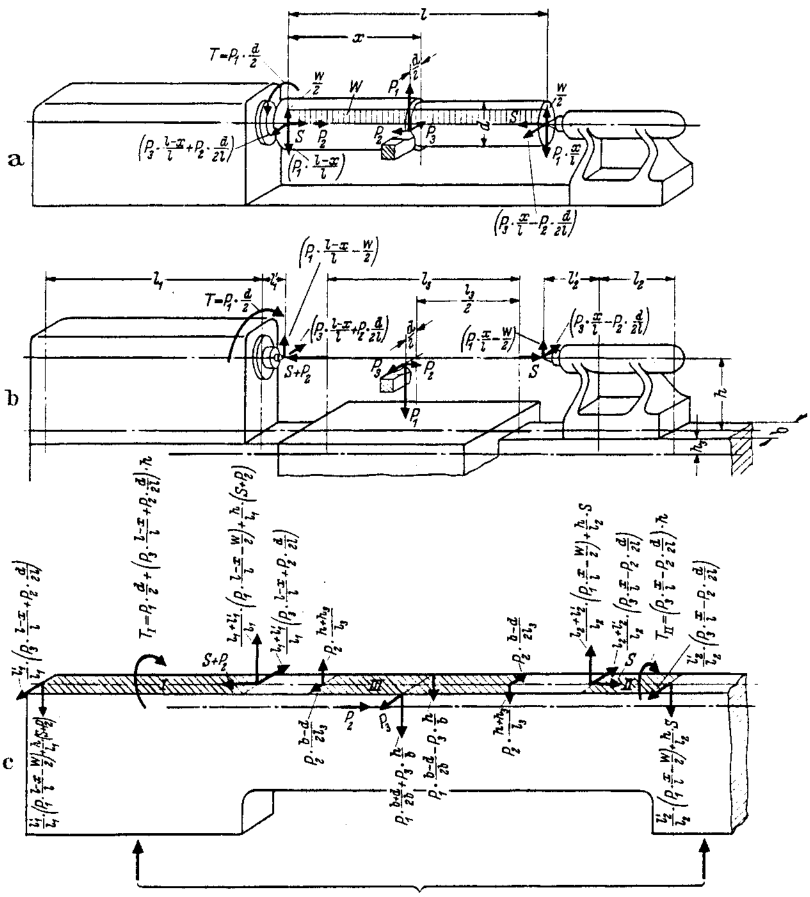


FIG. 297

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

Δd_2 and that due to torsion Δd_3. However, the effect of δ_1 upon the change in effective rake angles and the displacement of the cutting edge (danger of chatter) must not be neglected. The bed has to be designed, therefore, to be sufficiently stiff in bending and especially so in torsion.

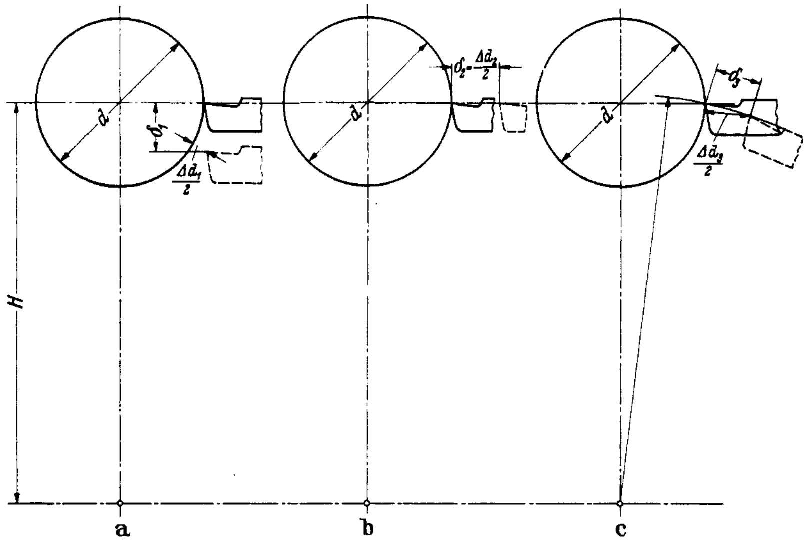


FIG. 298

(ii) **Drilling machine.** For the case of a drilling machine (Fig. 299), the conditions are relatively simple, because theoretically the column is loaded only by the axial force P acting at the drill point, which stresses it in tension and bending (the bending moment is equal to $P \times l$, where l is

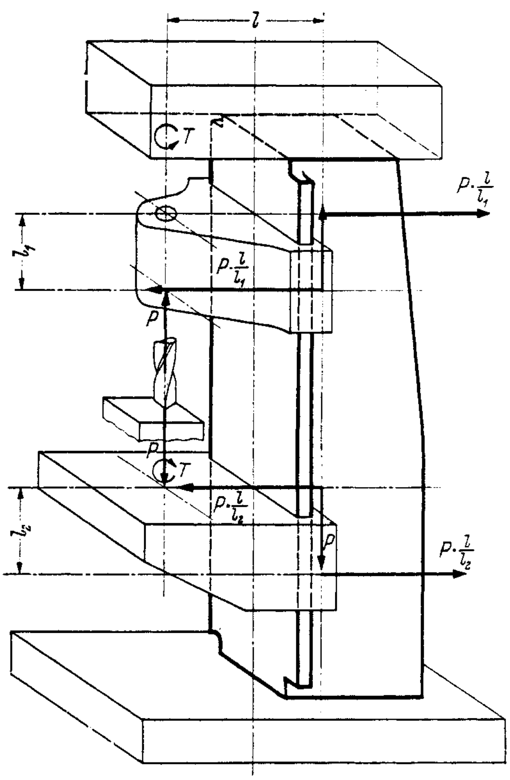


FIG. 299

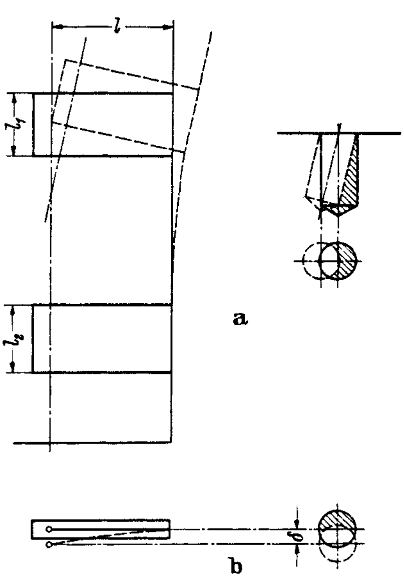


FIG. 300

the distance between the drill axis and the axis of the column), and by the torque T acting on the drill. The weights of the workpiece and the machine parts are not considered in this connexion. The elongation of the column, caused by the relatively low tensile force, can be neglected because this does not affect the position of the spindle axis. However, the inclinations of the axes of the spindle and of the drilled hole (Fig. 300a), caused by the bending moment, must be kept below permissible limits. The twist of the column is also relatively small, although it may cause a displacement (δ, Fig. 300b), which would affect the concentricity of the drilling operation (especially if the feed rate is low), and thus disturb the symmetry of the forces acting at the cutting edges (Fig. 301).

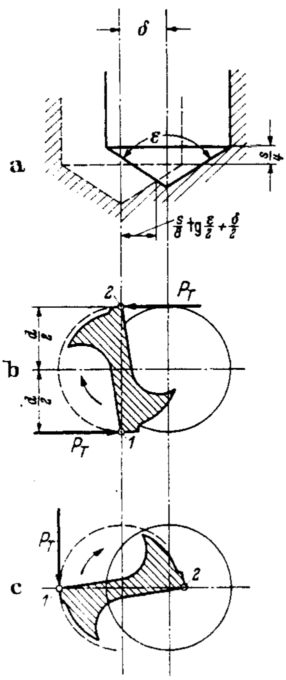


FIG. 301

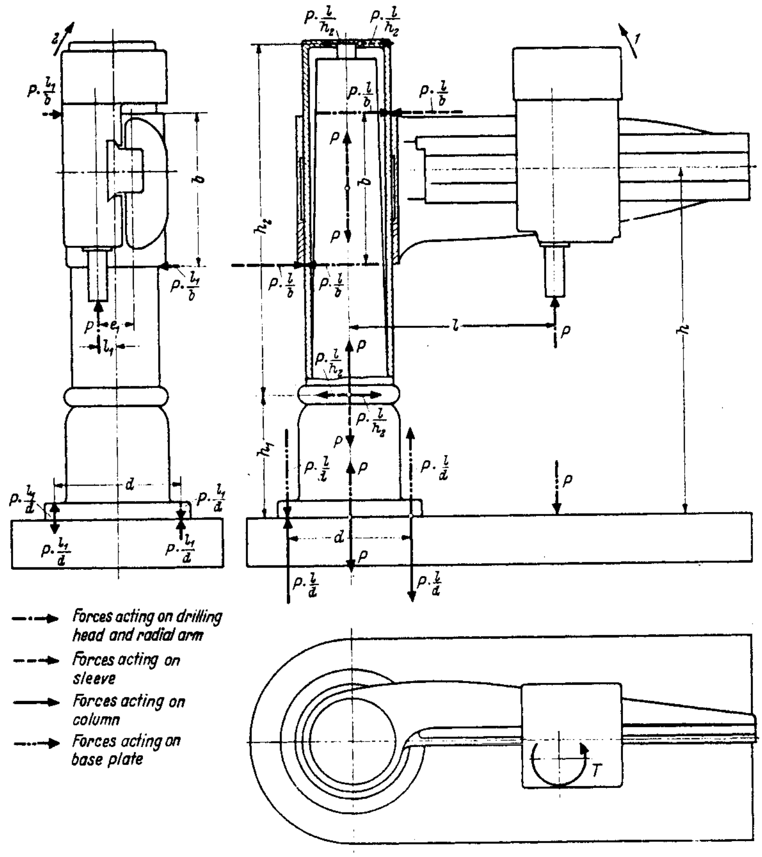


FIG. 302

Whilst in the position Fig. 301b, both cutting edges work symmetrically, this is not the case after a quarter revolution of the drill (Figs. 301a and 301c) and the drill cuts unsymmetrically. As these conditions vary periodically during each quarter revolution of the drill, undesirable vibrations may be created.[76]

The loading conditions of a *radial drilling machine* are shown in Fig. 302. The drilling spindle is usually arranged eccentrically to the radial arm axis. For this reason, the inclination of the spindle under the effect of the axial force P has to be considered in two planes (arrows 1 and 2). It is determined by the deflexions of the internal column, and the outer sleeve (bending moments $P \times l$ in the plane of arrow 1, Fig. 303 and $P \times l_1$ in the plane of arrow 2) and by the deflexion (arrow 1) and the twist in the plane of arrow 2 (torque $P \times e_1$) of the radial arm. In addition, it must not be forgotten that the deflexion of the radial arm under the weight of the spindle head varies for different positions of the head and that this cannot be neglected unless the slideway of the radial arm is machined in such a manner that, as long as no axial cutting force is applied to the spindle, the spindle axis remains vertical in any position of the spindle head.

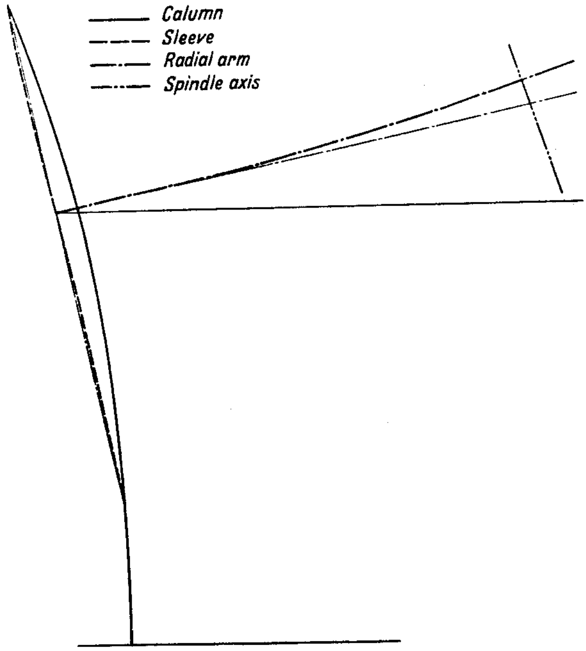


FIG. 303

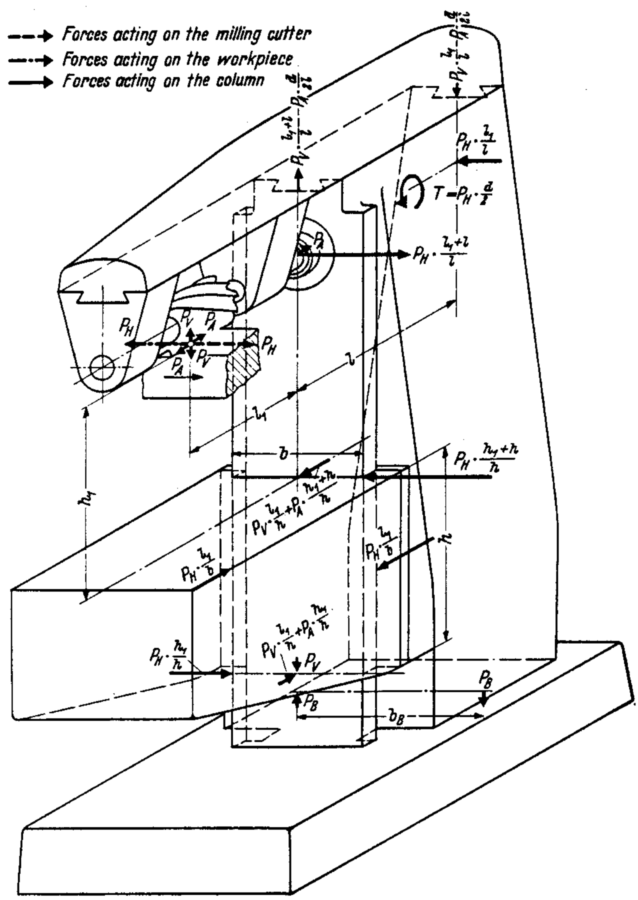


FIG. 304

(iii) **Knee-type milling machine.** (Fig. 304). The cutting force acting at any instant on the cutting edge is again resolved into three components (P_H, P_V, P_A, see Fig. 23b). The forces acting on the cutting edge (shown dotted) are transmitted via the milling arbor, the spindle and the overarm to the column, whilst the forces acting on the workpiece (shown chain-dotted) are transmitted via the table, the saddle and the knee to the column. The weights of the workpiece and the machine parts are again not considered. The point of application of the cutting force varies during the cutting operation, but this has little effect on the static conditions, because this positional variation is small in relation to the other dimensions (see page 11). As a simplification, it may also be assumed that during cutting the knee is clamped to the column and the vertical screw for raising or lowering the

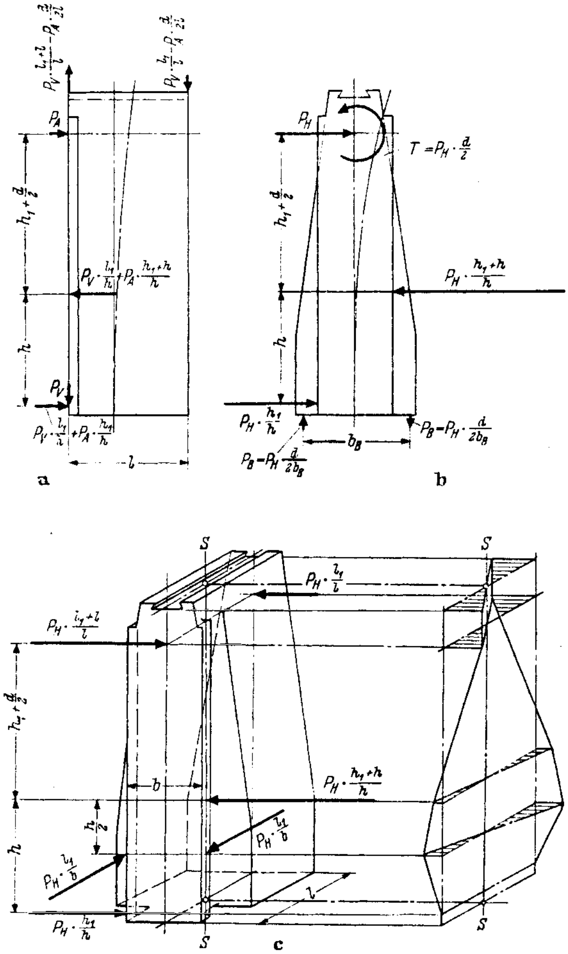


FIG. 305

knee is relieved of the cutting force; in other words, the knee transmits all forces directly to the vertical slideways of the column. For reasons of simplification, it may further be assumed that the force component P_V is transmitted to the column by the overarm and the components P_H and P_A by the spindle. The forces acting on the column can then be calculated in a similar manner as described for the case of the centre lathe, i.e. by establishing the equilibrium conditions for the force transmitting parts. The cutter diameter is d. The torque exerted on the spindle $T = P_H \times d/2$* is transmitted to the column via the spindle, the drive and the motor in the form of the bearing loads of the intermediate shafts and the supporting forces of the motor. Again, for reasons of simplification, it may be assumed that this torque is transmitted directly to the base ($P_B = T/b_B$, as shown in

* The horizontal component P_H may be assumed to be approximately equal to the tangential force component.

Fig. 304). Under the effect of the forces shown in Fig. 304, the column is bent in two planes (Figs. 305a and 305b) and twisted round its vertical axis *S–S* (Fig. 305c).

The columns of knee-type milling machines usually serve also as housings for gearboxes and control mechanisms and their cross-sections are rather complex. An accurate calculation of the deformations is difficult or impossible. Although only the trends of the deformations are shown in Fig. 305 (and these are not drawn to scale), they give an indication of how the column deforms under load. However, even if an accurate calculation of the deformations were possible, it would not have much practical value, because in the case of milling machines not only the magnitudes of static deformations, but also the variations of deformations during the milling operation and the danger of resonance, which would be particularly great at high working speeds (see page 55), are extremely important. In this connexion, it must not be overlooked that the stiffness of the machine depends not only upon that of the column, but also upon that of the other elements (knee, table, milling arbor and arbor support), etc., and also that of the joints (slides and slideways, clamping devices, etc.).

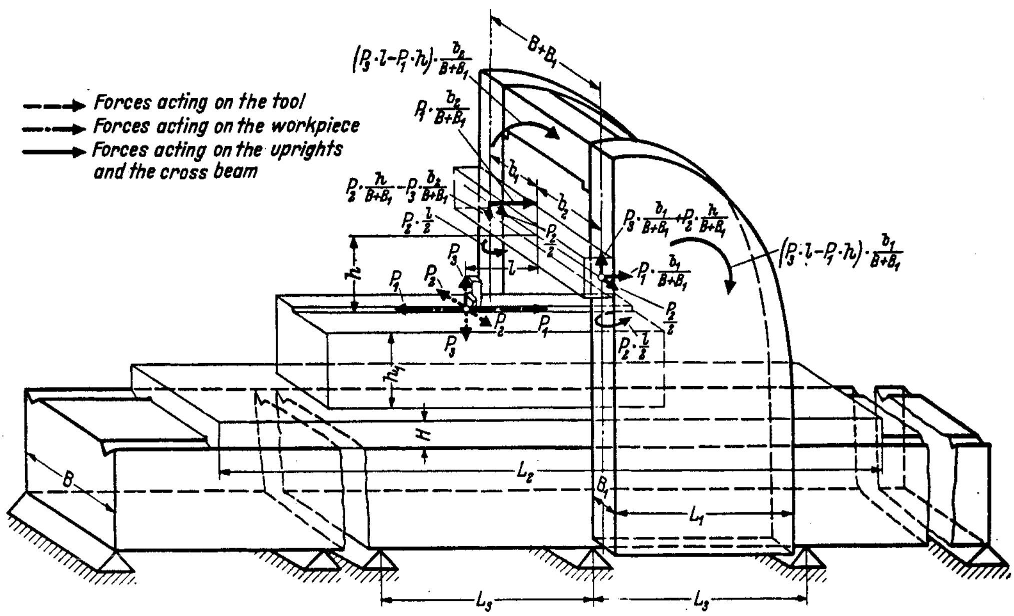


FIG. 306

From a general point of view, however, the designer of a milling machine column must remember that

(a) The bending forces acting in the plane (Fig. 305b) are considerably larger than those acting in the plane (Fig. 305a), and

(b) The torsional stiffness is of the greatest importance.

It is also essential that wide (in the plane shown in Fig. 305b) closed box sections, with a minimum of apertures between the level of the milling spindle and the knee, i.e. over the length $h_1 + d/2$ (Fig. 305c), must be provided (see page 46).

(iv) Planing machines. The forces acting on the bed and the structure of the planing machine are as follows (Fig. 306):

1. Weight W of table and workpiece. This may become rather large compared with the cutting forces.

2. The cutting force, which is again resolved into three components (P_1, P_2 and P_3). Additional cutting forces may have to be considered if several tool slides on the cross beam, and possibly on the uprights, are provided. Whilst the weights of the various parts are transmitted by the bed to the supports, the flow of the cutting forces is closed in the bed on to which they are transmitted on

the one hand by the workpiece on the table and on the other by the tool, the tool carrier, the cross beam and the uprights.

The depth H of the table cross-section is usually small compared with that of the bed section. It is, therefore, reasonable to assume that the table adjusts its shape to the shape of the bed and that its weight, together with that of the workpiece, is evenly distributed over the slideways along the length L_2. It may also be assumed that the weight of the workpiece is equally supported by the two table slideways, so that on each slideway of the bed a force $W/2$ is evenly distributed over the length L_2 of the table. The bed is supported at equal distances L_3 on adjustable blocks (see page 214).

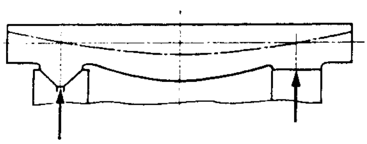


FIG. 307

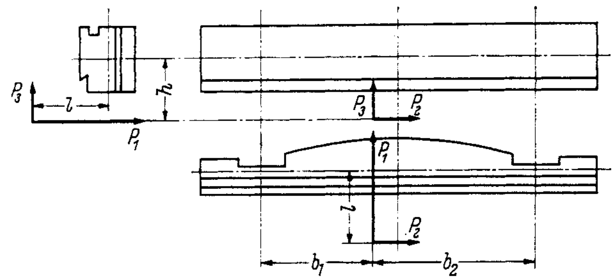


FIG. 308

The distance between these blocks is chosen so that the maximum deflexion of the bed calculated under the assumption of a beam freely supported on these supports, remains within permissible limits. The deflexion of the table in the transverse plane is not affected by the stiffness of the bed. It can be calculated easily as it is reasonable to consider the table as a beam which is freely supported on the two slideways (Fig. 307).

Compared with the weights of the bed and the table, the magnitude of the cutting force components is generally negligibly small. Even if the cutting force acts at a considerable height above the surface of the table, and thus exerts moments which tend to twist and bend the bed upwards, these moments are much smaller than those exerted in the opposite direction by the weight of the table and the bed. It will be appreciated, therefore, that it is desirable to design the bed of the planing machine as heavy as possible. From the point of view of stiffness, it would also be preferable to design the table as heavy as possible. However, it must not be forgotten that the weight of the table affects the reversing action (see page 142).

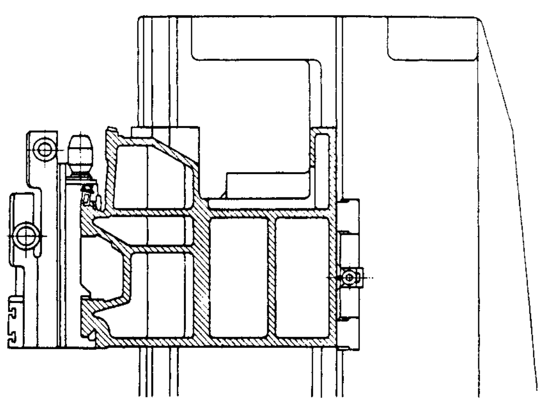


FIG. 309. Section through the cross beam of a heavy duty planing and milling machine (H. A. Waldrich, Siegen, Germany).

The force components acting at the tool edge are transmitted by the tool carrier to the cross beam and from there, on to the uprights and the bed. The cross beam is stressed in bending in the vertical plane (component P_3 and bending moment $P_2 \times h$), and in the horizontal plane (component P_1 and bending moment $P_2 \times l$). It is also stressed in torsion (torque $P_1 \times h - P_3 \times l$) (Fig. 308). The magnitude of the resulting deformations depends upon the shape and dimensions of the cross beam and upon its support conditions on the uprights. Leading designers of these machines obtain the necessary bending stiffness of the cross beam (Fig. 309) in the horizontal plane by providing a sufficient depth to its cross section, whilst the torsional stiffness is obtained by designing it as a closed box (see page 44). The provision of powerful clamping devices K, which are distributed in suitable positions over the vertical slideways (Fig. 310), further increases the stability of the cross beam. The forces acting on one upright are shown in Fig. 311a. The maximum loading occurs when the tool carrier is in its highest position and near to one upright ($b_1 = 0$ or $b_2 = 0$, see Fig. 306) because, in this case, this upright has to carry practically the whole cutting force, and the bending moment produced by P_1 reaches its maximum.

Finally, Fig. 311b shows the forces exerted by the table and the uprights upon the bed. Of particular importance in this connexion is the joint between the bed and the uprights (see page 50) which has to ensure that the relative position between the upright slideways and the bed slideways remains within permissible limits under load.

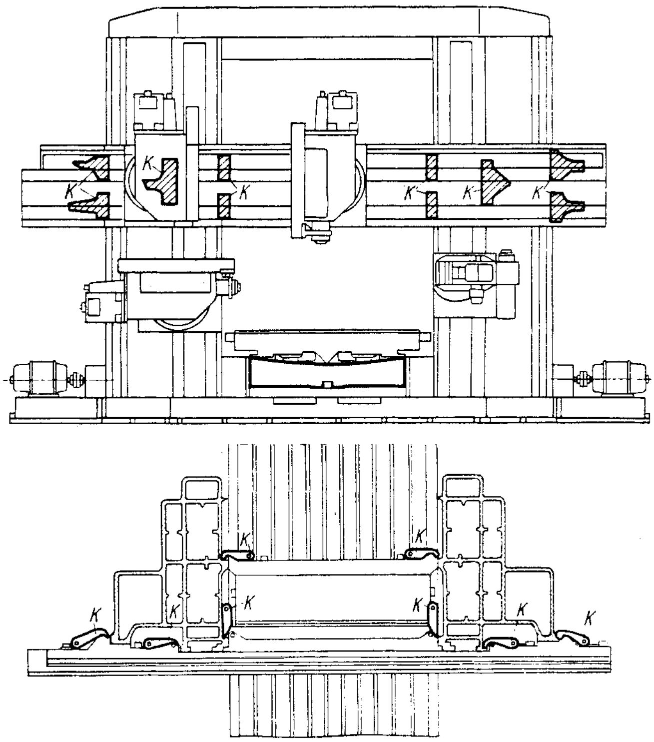


FIG. 310. Clamping of the cross beam for a heavy duty planing and milling machine (H. A. Waldrich Siegen, Germany).

(E) In general, either cast iron or welded steel is used. Both have their advantages and the designer has to consider in either case the technical and economic aspects, before deciding in favour of one or other material. These aspects, some of which have been mentioned before (see page 45), are:

(i) Material properties
  (a) Strength under tensile, compressive, static and dynamic loads
  (b) Stiffness against deformation under static loads
  (c) Vibration behaviour (damping)
  (d) Sliding properties in the case of slideways (wear).

(ii) Manufacturing problems
  (a) Maintaining accurate wall thicknesses
  (b) Combination of different wall thicknesses
  (c) Production of thin walls
  (d) Machining allowances
  (e) Internal stresses.

(iii) Economy
  (a) Cost of patterns or fixtures respectively
  (b) Economy in weight.

These will now be discussed in detail.

Points (i) (a), (b) and (c) have already been mentioned (see page 45).

With regard to (d), it should be remembered that the sliding properties of cast iron are usually superior to those of weldable steels. In welded steel structures it is, therefore, necessary to take special precautions. The saddle slideways of the milling machine described previously (see page 102 and Fig. 133) were made from 0·4 per cent carbon steel (see Fig. 340), so that they could be flame hardened after stress relieving heat treatment. They were ground on a slideway grinding machine and have shown an excellent performance in combination with the cast iron slides of the saddle. The vertical *T*-slot carrying slideways for the spindle head of the same machine are of cast iron and fixed to the column by bolts and dowel pins (see Fig. 341).

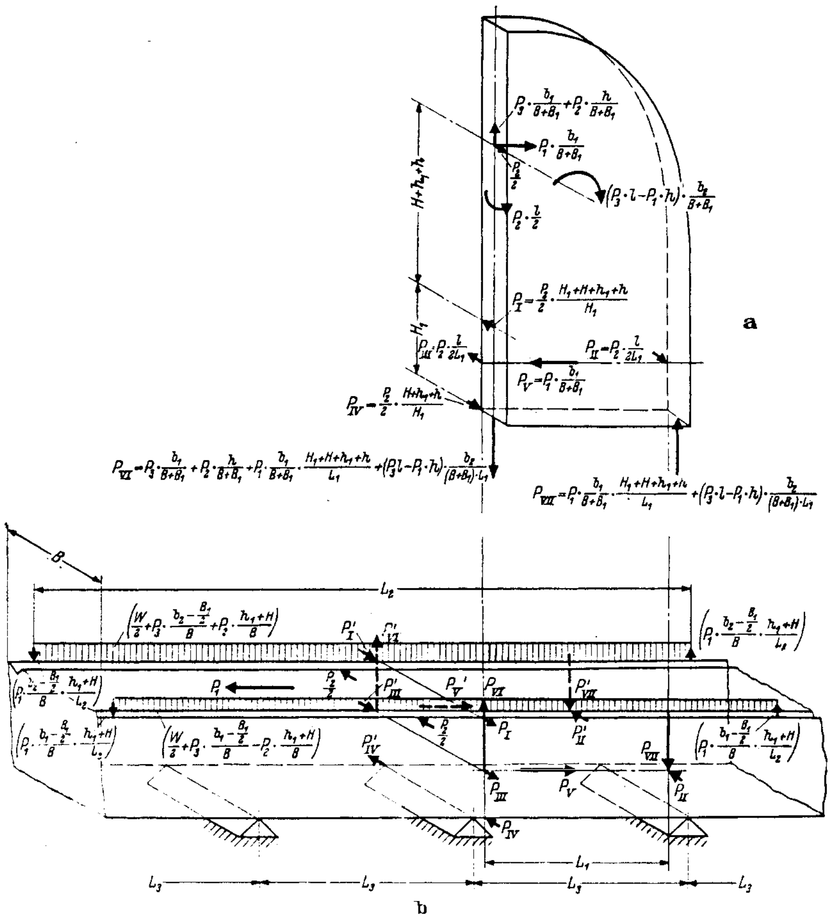


FIG. 311

(ii) (a). It is often difficult to prevent core displacements and therefore wall thickness variations in castings. In order to avoid the resultant weakening of the parts in question, it is necessary to increase the theoretically requisite wall thicknesses of castings by an amount equal to the tolerance of the core position. This means that the weight or the material consumption of a casting is greater than would appear necessary from theoretical considerations.

(b) In a weldment, it is not difficult to arrange for walls of different thicknesses to be joined. In the case of castings, abrupt changes in wall thicknesses can lead to casting faults and for this reason, it is again necessary to make the casting heavier than would appear necessary from theoretical considerations.

(c) It is often difficult to produce thin cast walls and, for this reason, it is again rarely possible to design castings which do not exceed the minimum weight theoretically required for reasons of stiffness or strength.

(d) The machining allowances for castings have to be such that no trace of the skin remains after machining. In order to prevent surface irregularities on castings, produced by fall of sand in the mould or other causes, necessitating rejection as a whole, machining allowances must often be considerably larger than in the case of steel sheets or steel plates, whose surface irregularities are much smaller.

(e) Internal stresses must be reduced or removed as far as possible in castings and weldments for machine tool structures. This can be achieved by suitable heat treatment (in castings, certain success can be achieved by natural ageing). Weldments are usually heated to 600 to 650°C (in the case of structures for highest precision requirements up to 750°C). They are kept at this temperature for about one hour per inch of thickness of the thickest wall and then left in the closed furnace until the temperature has dropped to about 250 to 200°C.

(iii) (a) If small quantities are to be produced, the cost of patterns may have a considerable effect upon the total manufacturing cost. On the other hand, the cost of welding fixtures, especially if a large number of identical weldments have to be produced, may also have to be considered.

Relatively simple beds or columns with regular shapes without protruding bearing bosses, complicated ribbing, etc., can be assembled easily and welded without fixtures. They are, therefore, more suitable for welded construction than complicated arrangements in which a large number of relatively small components would have to be assembled. Apart from the need for welding fixtures, the preparation of the component parts for such designs and particularly the actual labour cost for welding generally exceeds the cost of the patterns which make it possible to produce complex shapes almost automatically by the casting process.

Although all these considerations must be taken into account for each case separately, it can perhaps be stated generally that welded construction is more suitable for simple structures, whilst for complicated shapes, the cast design appears preferable.

(b) The question of weight and with it, that of possible economy in material, must be considered very carefully indeed. Reduction in weight may be advantageous or absolutely essential for technical reasons (see page 56). Economic considerations are important, because not only the cost of material but also that of transport and Customs may be based on the weight of the finished machine. Whilst the theoretically requisite minimum weight of a casting will usually have to be exceeded for manufacturing reasons (see page 223), it is possible to keep the weight of a welded steel structure low, although this may not affect the material consumption itself. It must not be forgotten that the material consumption is not proportional or equal to the finished weldment, because any scrap material, cut-outs, etc., have to be paid for when the plates are bought.

In castings, lightening holes are produced almost automatically by the casting process, because the molten material cannot fill the spaces which are already covered by the moulding sand and the cores. On the other hand, in the case of weldments, all such lightening holes have to be cut out of the plate material by additional operations. In other words, not only must the weight of the scrap metal be paid for, but also additional expenditure is necessary in order to cover labour cost, and overheads.

If, therefore, the finished weight of the weldment need not be low for technical reasons, any saving in weight obtained by such means is not economical. This can, of course, be remedied by suitable design of the structure, especially by using the scrap material elsewhere. As an example, the material cut out of the walls of the milling machine bed (Fig. 134) in order to provide the apertures for access to the motors, was used to form the doors. This not only led to a saving in material, but also eliminated the need for fitting the doors into the apertures, because the cut-out pieces were bound to fit into the corresponding apertures. The drilled hole at the start of the flame-cut was placed in the position of one of the hinges and was, therefore, covered during assembly.

The more complex the component parts of a weldment, the greater is the loss of material caused by scrap. It is, therefore, again most important to design simple shapes. The greater the number of component parts in a weldment, the more preparatory work is necessary and the more difficult is the supervision of material movements. The welded seams are also longer and the cost increases. For this reason, it is sometimes advantageous to reduce the labour cost even at the expense of increased material consumption.

The welding of a ball bearing housing into a thin wall (Fig. 312a), results in less material consumption than the use of a thicker wall (Fig. 312b). However, in the design, Fig. 312b, all the welding work and the separate manufacture of the housing *1* is eliminated. Which arrangement is the more economical one depends upon the general conditions of material and labour cost. In 1955, for instance, the saving of one ton of steel used in a weldment was justified in Germany, even if this saving in material required additional labour plus overheads for 80 man hours.[77] This is possibly the main reason for the considerable development and application of the cellular and shell construction methods in Germany. In Great Britain and the U.S.A., these types of design are not favoured because, for instance, in the U.S.A. the cost of one ton of steel corresponds only to the cost of labour plus overheads for 20 man hours.

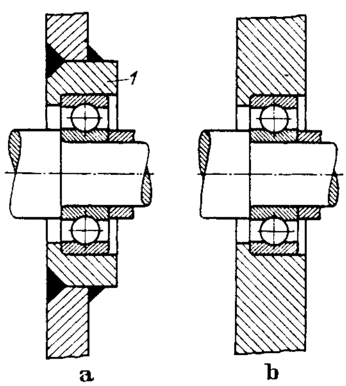


FIG. 312

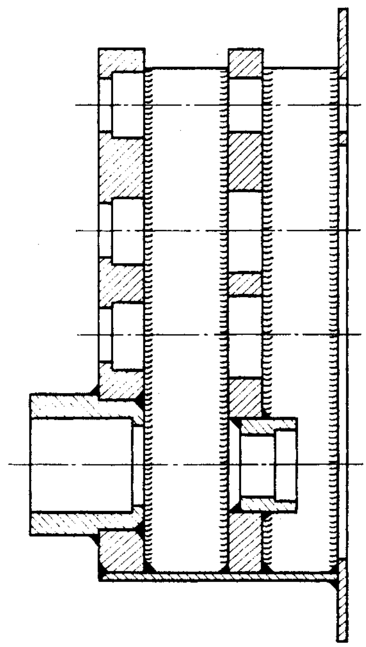


FIG. 313. Welded feed gearbox for the milling machine (Fig. 133).

If the application of welded construction appears uneconomical, even where technical reasons necessitate weight reduction, light alloy castings are sometimes used. As often happens during his work, the designer may find that a compromise produces the most favourable solution. If, for instance, in a certain case, the application of welded construction appears very suitable, yet on the other hand, certain parts of the structure are complex or otherwise unsuitable for welding and would excessively increase the total cost of the structure, a combination of steel castings and weldments may be the best solution. An example is the main spindle bearing housing of the milling machine (Fig. 142). This housing is a steel casting (carbon content below 0·25 per cent), which is welded into the spindle head. Similarly, two bearing housings are welded into the feed drive gearbox (Fig. 313) of the milling machine (Fig. 133), whilst the other ball bearing housings are machined directly into the walls of the gearbox.

Finally, it may be mentioned that in some cases, the available plant and equipment, a good foundry or a modern welding shop, may decide the issue. However, the designer must avoid an excessive influence of such considerations over and above technical or economic reasons.

(F) In many machines, especially those which work at high speeds and which, therefore, produce large quantities of chips per unit time, the problems of chip removal and chip disposal have to be borne in mind during the design of the beds. The chips must be removed from the actual cutting zone as quickly as possible and transported away from the machine, especially if the quantities are large. The designer must design the machine beds, therefore, in such a manner that the chips can fall freely away from the cutting zone and conveyors or other means may need to be provided for transporting the chips away from the machine (Fig. 314).

In the design of beds and columns, the problem of the slideway layout is of the same importance as that of the transmission of the various operational forces. The detail design of the elements which

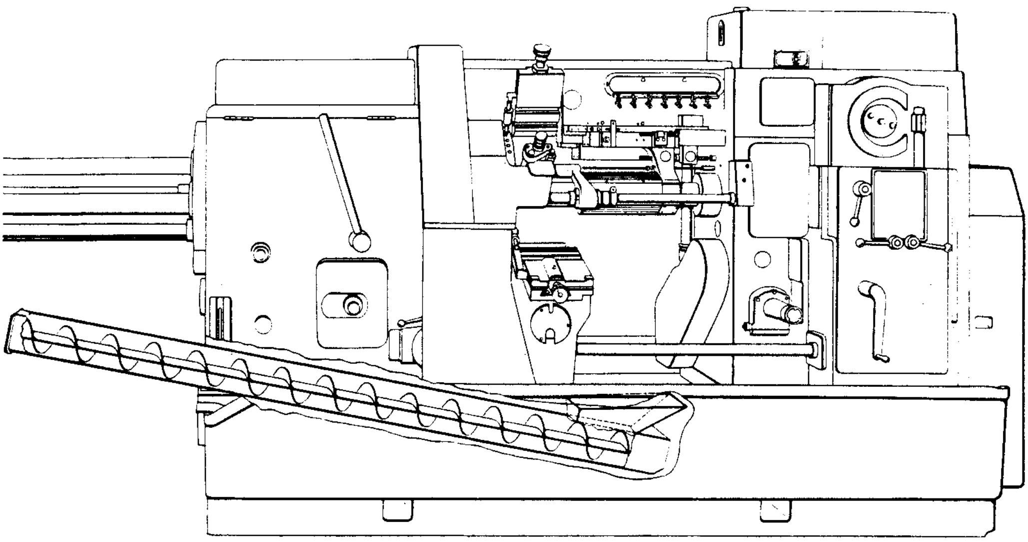


FIG. 314. Swarf conveyor in a five spindle automatic (Wickman, Ltd., Coventry).

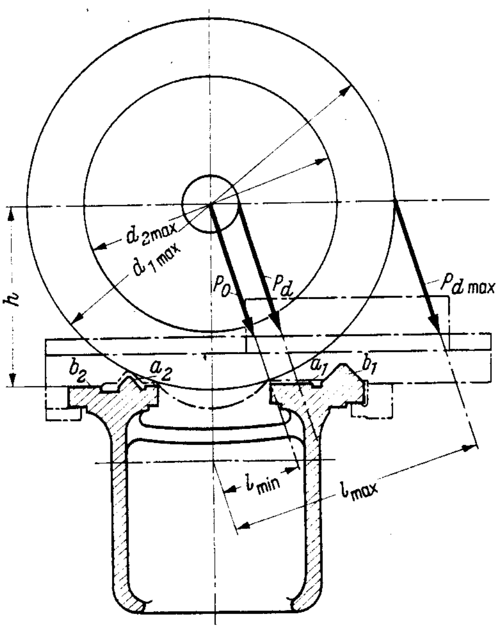


FIG. 315

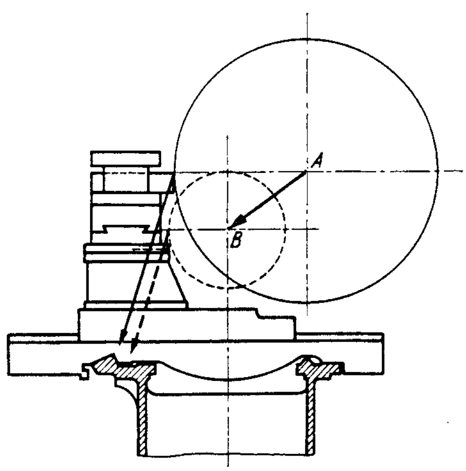


FIG. 316

guide the moving parts (slides, tables, tool carriers, etc.), with the required accuracy and which keep them in their desired relative positions under load, will be discussed in the next section. Only the correct layout of these elements on the structure makes it possible, however, to exploit fully the stiffness and strength properties of the structure. This means that the slideways must be arranged in such a manner that the operational forces are taken and transmitted in the best possible way and in the most favourable positions.

The workpiece diameter which has to be machined on a centre lathe, may vary between zero and a maximum value, the latter being determined by the dimensions of the machine (especially the height of centres h). The maximum diameter above the bed may be $d_{1_{max}}$, the maximum turning diameter above the cross-slide $d_{2_{max}}$ (Fig. 315). The leverage l, of the torque which twists the bed ($P \times l$), may then vary between l_{min} and l_{max} and the twisting torque depends, therefore, on the machined diameter and may vary between $T_{min} = P \times l_{min}$ and $T_{max} = P \times l_{max}$. In view of the fact that the twist of the bed forms a large part of the total deformation and has thus considerable influence upon the working accuracy (see page 216), the designers of the Austrian lathe "Neomat" (Heid, Vienna), have provided a movable head and tailstock (Fig. 316).[78] The position of the turning axis can be moved from A in the case of the maximum diameter to B in the case of a smaller diameter, thus keeping

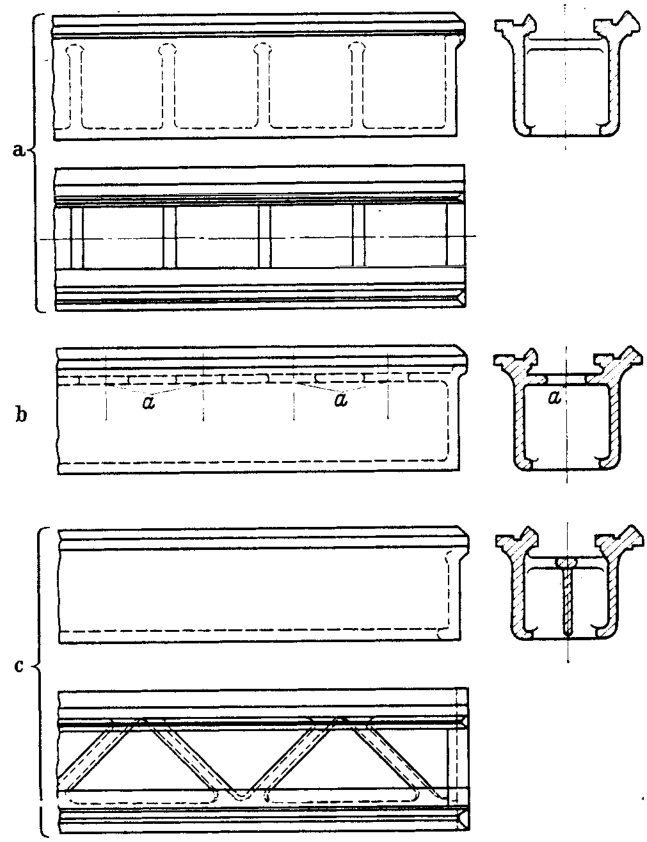


FIG. 317

the position of the line of action of the cutting force almost constant. It must be remembered, however, that in such a design, the mechanisms and the driving gear become rather complicated, and it may not be easy to maintain the correct alignment between head and tailstock within permissible limits in all positions. For this reason, most designers maintain the position of the turning axis and use torque-resistant cross-sections, in which the deformations can be kept within permissible limits, even under the highest possible torques. If beds are designed with two shears, which carry the slideways (a_1 and a_2 for head and tailstock, b_1 and b_2 for the saddle, see Fig. 315), the space between the shears is usually left open for the chips to fall freely. Whilst it is usual for the shears to be sufficiently stiff against bending in the vertical plane, they have to be stiffened by suitable ribbing against bending in the horizontal plane and against torsion. It will be obvious that vertical stiffener ribs (Fig. 317a) would neither increase the stiffness against bending, nor that against torsion. Horizontal stiffeners (Fig. 317b) would have to be provided with apertures a in order to allow the chips to fall through the bed. Such stiffeners increase the stiffness against bending but not that against torsion. The best stiffening effect is obtained with the so-called Peters arrangement (Fig. 317c),[79] in which the ribs are arranged diagonally. Maximum torsional stiffness is obtained with closed box sections (see page 44) and many high power lathes show interesting design solutions to the problem of combining closed box sections with the possibility of free fall of the chips. The bed of the machine (Fig. 318),[80] carries the saddle A on two box sections B and C between which the

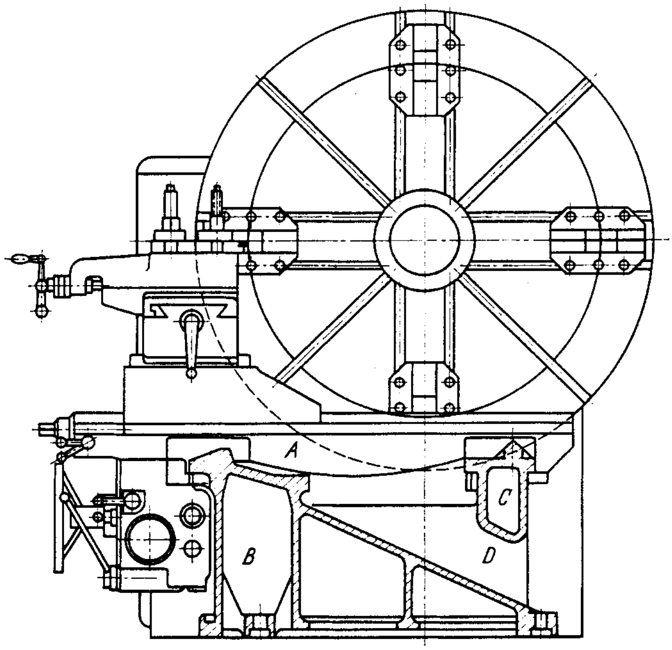


FIG. 318. Section through the bed of a heavy duty centre lathe (Heylige staedt & Co., Giessen, Germany).

chips can escape by way of the sloping channel D. A similar arrangement is found in the Jones and Lamson Machine (Fig. 319), where the slideways a are made of hardened steel and bolted to the bed casting for easy replacement in case of wear.

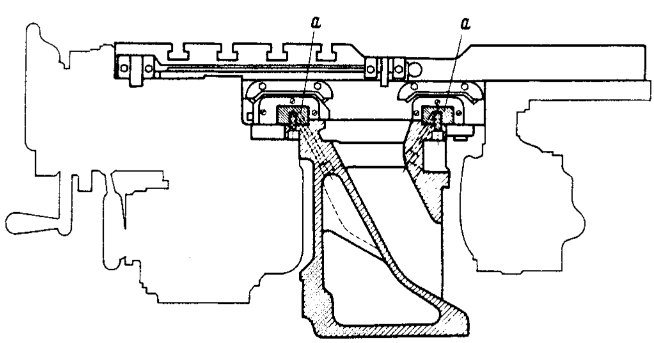


FIG. 319. Section through the bed of a lathe (Jones & Lamson Machine Co., Springfield, Vermont, U.S.A.).

The bed of the heavy roll turning lathe (Fig. 320)[81] consists of three box sections, the rear one of which (a) supports the head and tailstock, whilst the front one b supports the saddle. The centre box c serves as a joining element and carries on its rear slideway c_1 the head and tailstock and on the front one c_2 the saddle. The flow of operational forces is, therefore, closed via the foundation which has been provided with a channel for chip removal d and a transport channel (e), thus saving in overall height of the machine.

The slope of the various walls, although facilitating the falling of the chips, does not solve the problem of chip removal completely as long as the slideways are exposed to the swarf and can, therefore, be damaged by chips which may have been work hardened, especially during work at

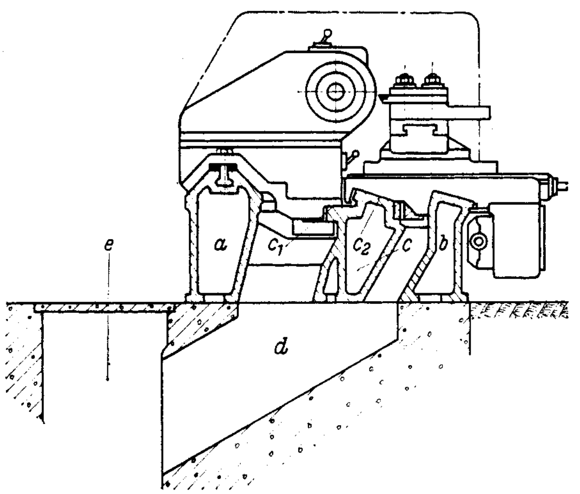


FIG. 320. Bed of the "Waldrich" roll turning lathe.

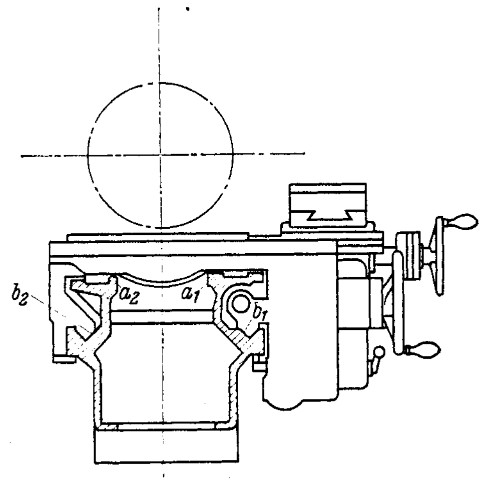


FIG. 321. Bed of the "Schaerer" lathe (Industrie Werke, Karlsruhe, Germany).

high cutting speeds. A certain protection can be provided by the use of slideway covers (see page 253). A sloping arrangement of the slideways may prevent the chips from collecting on the surfaces and thus giving rise to any damage.

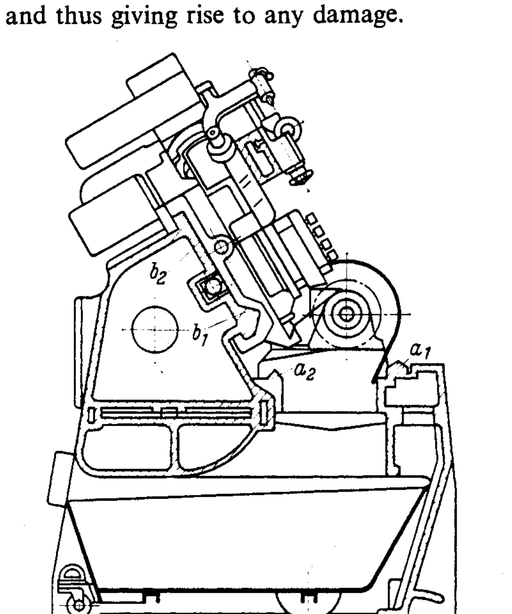


FIG. 322. Bed of the Copying Lathe "Heycomat" (Heyligenstaedt, Giessen, Germany).

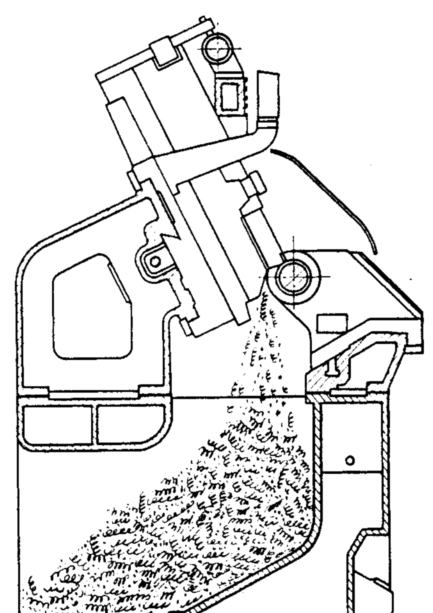


FIG. 323. Bed of the semi-automatic "Schaerer" copying lathe (Industrie-Werke, Karlsruhe, Germany).

So far as wear is concerned, the saddle slideway has to be specially considered, as the headstock is not moved, once it has been fixed to the bed, and the tailstock is only shifted when turning operations are not in progress (see page 246).

An original design solution is shown on the bed of the Schaerer lathe (Fig. 321), in which the slideways for the headstock and tailstock (a_1 and a_2) are arranged in the usual manner whilst the slideways for the saddle (b_1 and b_2) are below the level of the former and are covered by the top portion of the bed. In order to avoid weakening of the saddle through intermediate members and bolted joints, the saddles of these machines are cast in one piece with the apron which also carries one of the sliding surfaces.

The best protection against damage through falling chips can be obtained by arranging the slideways completely outside the fall of chips and swarf. This has been achieved in the machine (Fig. 322),[81] in which a heavy box section above the turning axis carries the saddle slideways b_1 and b_2

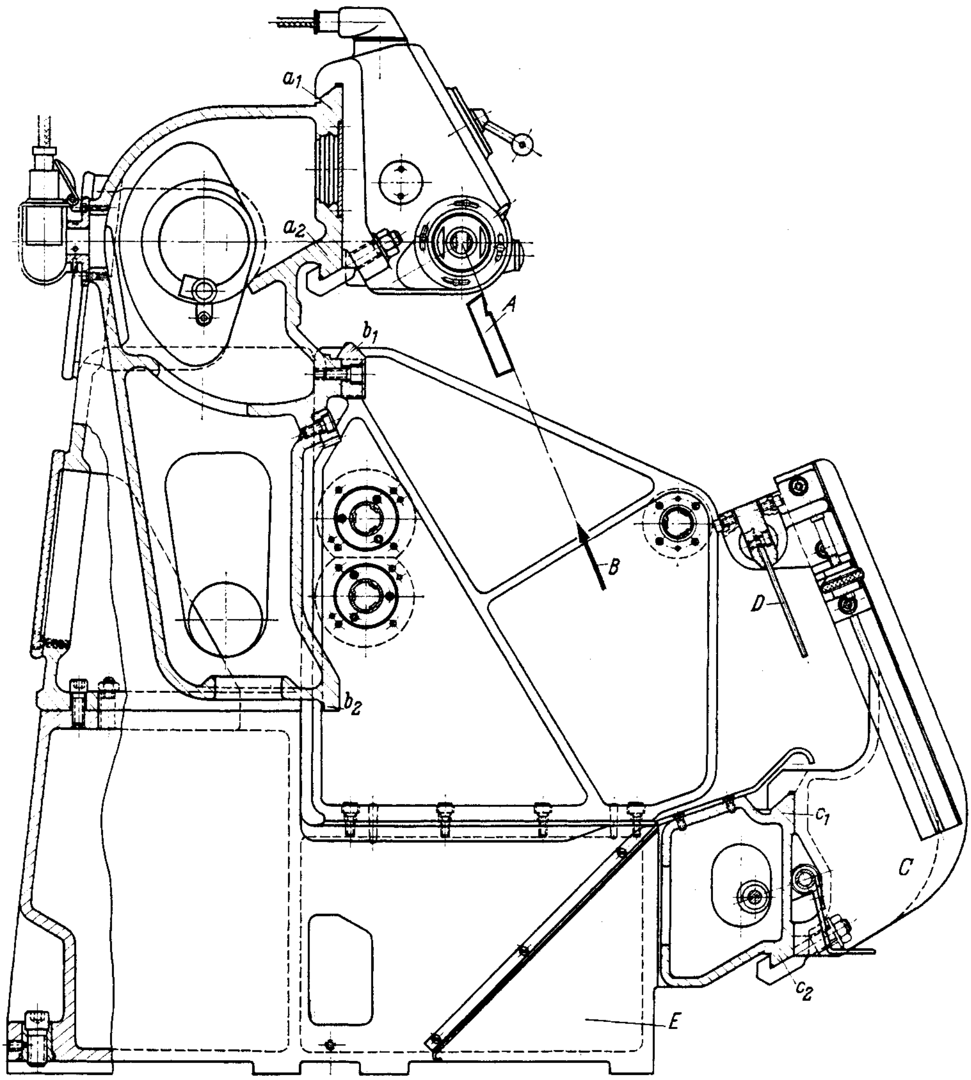


FIG. 324. Bed of a copying lathe (Georg Fischer, Schaffhausen, Switzerland).

and one of the slideways (a_2) for the head and tailstock. Slideway a_2 serves also for connecting the two boxes, the lower one of which carries the other head and tailstock slideway a_1. Below these two box sections a chip removal channel is provided, together with space for a swarf barrow, thus solving at the same time the problem of removing the swarf from the machine.

The complete separation of the slideways (those aligning the workpiece and those carrying the saddle) has been achieved in the copying lathe (Fig. 323), where the saddle slideways lie again outside the chip fall, whilst the tailstock slideways slope at such an angle that the danger of chip deposition is small. At the same time, in this arrangement, easy clamping and removing of the workpiece are possible and the operator can watch comfortably both the cutting tool and the tracer. The separation of the guiding elements for tool and workpiece carrier requires stiff connecting elements which can be provided by the bed (Figs. 322 and 323) or by the guided parts (Fig. 320). This difficulty has been avoided in the design of the bed for the Fischer copying lathe (see Fig. 272 and Fig. 324). All slideways are carried by the main cross-section of the bed, and this is of almost elliptical shape, so that high stiffness against torsion is obtained. The position of the ellipse relative to that of the cutting tool *A*, which is fed into the workpiece in the direction of arrow *B*, ensures maximum stiffness in the direction of the radial component P_3 of the cutting force and, therefore, against bending in the plane which is most important from the point of view of the machining accuracy see (page 216).

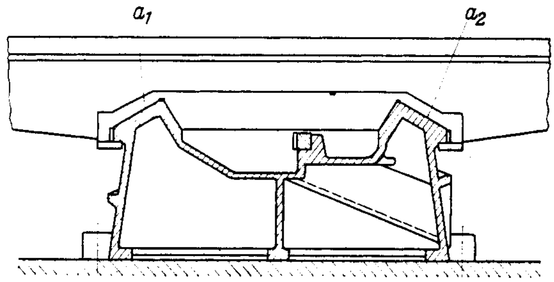


Fig. 325

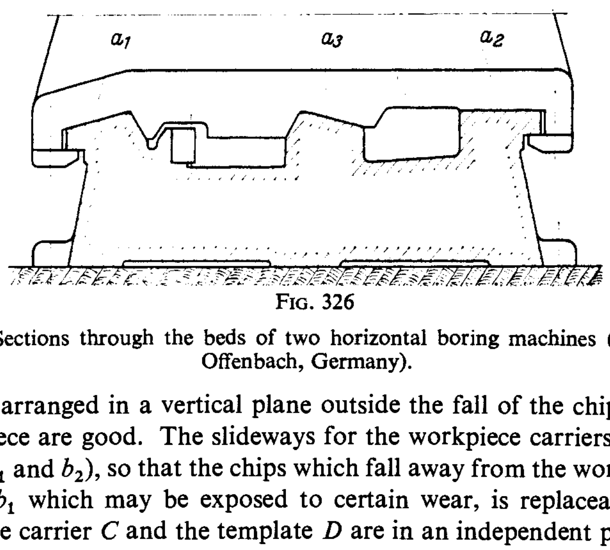


FIG. 326

FIGS. 325 and 326. Sections through the beds of two horizontal boring machines (Collet & Engelhard, Offenbach, Germany).

The slideways are arranged in a vertical plane outside the fall of the chips, and accessibility to both tool and workpiece are good. The slideways for the workpiece carriers (a_1 and a_2) are above those for the saddle (b_1 and b_2), so that the chips which fall away from the workpiece cannot damage them. The slideway b_1 which may be exposed to certain wear, is replaceable. The slideways c_1 and c_2 for the template carrier *C* and the template *D* are in an independent position where they are not affected by the deforming effects of the cutting force.

In contrast to the conditions encountered with the centre lathe, the cutting forces which arise during the work on *horizontal boring machines* are usually considerably smaller than the weights of the various parts carried and guided by the bed. In addition to the two slideways a_1 and a_2 in a typical cross-section of a horizontal boring machine bed (Fig. 325), a third one a_3 has been added on the bed (Fig. 326), in order to ensure that a wider saddle does not deflect excessively in the middle. Both bed sections are of an inverted channel shape and stiffened by diagonal webs.

An interesting design is the bed of the horizontal boring machine of H. W. Kearns (Fig. 327), both the saddle and the outer stay being guided on an outer and an inner set of flat slideways. The necessary ease of movement is ensured by roller bearings on the outer ways a_1 and a_2, whilst the inner ways b_1 and b_2 enhance the stiffness of the supported parts. As the inner slideways b_1 and b_2 are raised the longitudinal feed screw for the saddle can be completely immersed in oil, thus reducing friction.

In some designs, the cross traverse slideways of the saddle *A* are so long that the cross-slide *B* never overhangs during its traverse, even in its two extreme positions (Fig. 328). Other designers

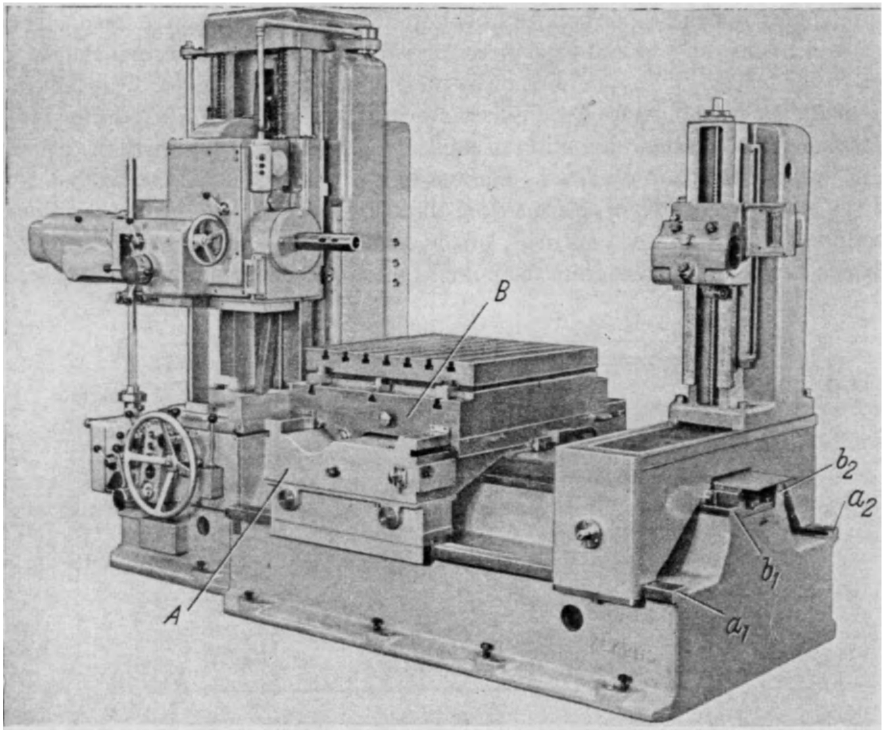


FIG. 327. Horizontal boring machine "Optimetric" (H. W. Kearns & Co. Ltd., Altrincham).

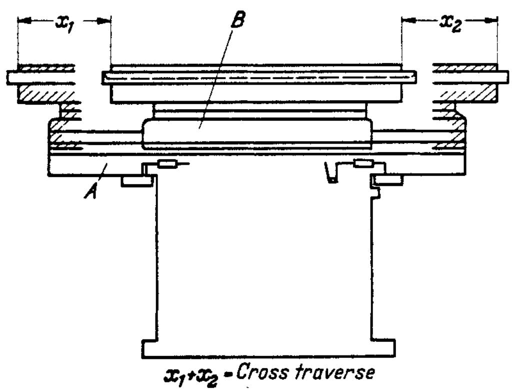


FIG. 328

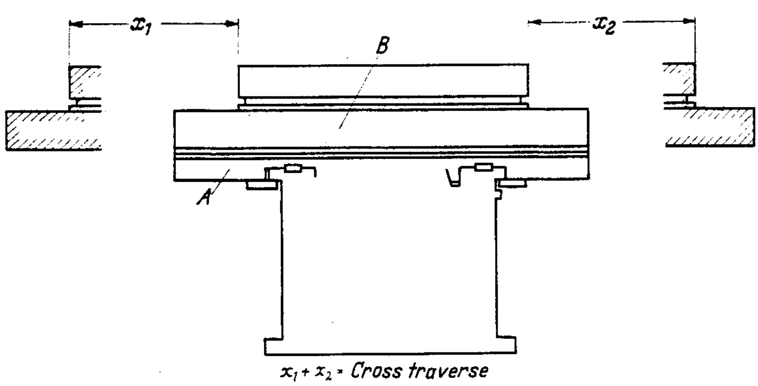


FIG. 329

arrange for the cross slide to be so stiff that a certain overhang is permissible and a considerably longer traverse possible (Fig. 329). A design of the latter type is shown in Fig. 330, where the saddle is guided on four outer (a_1'–a_1'' and a_2'–a_2'') and two inner ways (b_1 and b_2). However, the difficulties of manufacturing such an overdetermined slideway design must not be overlooked!

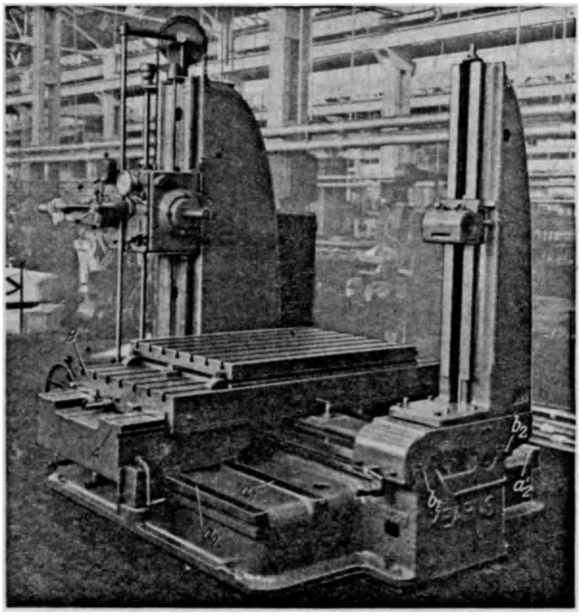


FIG. 330. Horizontal boring machine (H. W. Kearns & Co. Ltd., Altrincham).

In order to obtain the static stiffness and vibration rigidity required for high quality grinding operations, the beds of *grinding machines* can be designed to be either extremely heavy or extremely light (see page 56). Figure 331 shows an orthodox heavy design of a grinding machine bed. The slideways for the longitudinal traverse of the table (a and b) are arranged on heavy box sections A. These are completely closed except for some core print apertures reinforced by cross ribs and are therefore extremely stiff against bending and torsion. The connexion of the part B of the bed, which carries the grinding wheel head, with the bed A, is strengthened by diagonal webs.

In contrast to the heavy bed of this machine, the bed of the "Diskus" grinding machine (Fig. 332) is designed in the lightweight cellular construction (see page 44). In this case, vibrational rigidity is ensured by combining high static stiffness with light weight (thickness of the cell walls is $\frac{5}{32}$ in.), this resulting in a high natural frequency (see page 56).

Figure 333 shows in schematic form the arrangements of such cells in the column of a surface grinding machine with a vertical grinding spindle. The practical execution can be seen from Fig. 334.

Some problems encountered in the design of *radial drilling machine structures* have been discussed earlier (see page 217). It may be repeated once again that the stiffness against bending of the column and the torsional stiffness of the radial arm are of vital importance. In the machine (Fig. 335), the main column is stiffened over its full length by two crossed ribs, whilst the radial arm (Fig. 336) is designed as a diagonally webbed closed box section and is thus sufficiently stiff against torsion. Outside the closed box section of the radial arm, a channel is formed by the two horizontal outer walls (see Fig. 336), embracing the space in which the electric control and switching devices are housed.

The most severe working conditions are perhaps applied to the structures of high power *milling machines*, in which large cutting forces are usually combined with high frequencies of force pulsations. For this reason, the cross beam of the planing and plano-milling machine shown in Fig. 310 is equipped with an additional slideway (a, Fig. 337), for the case of milling heads being used instead of ordinary planing tool slides. This additional slideway, which remains unused for planing slides (see Fig. 309), serves for supporting the milling head against the heavy horizontal forces which may

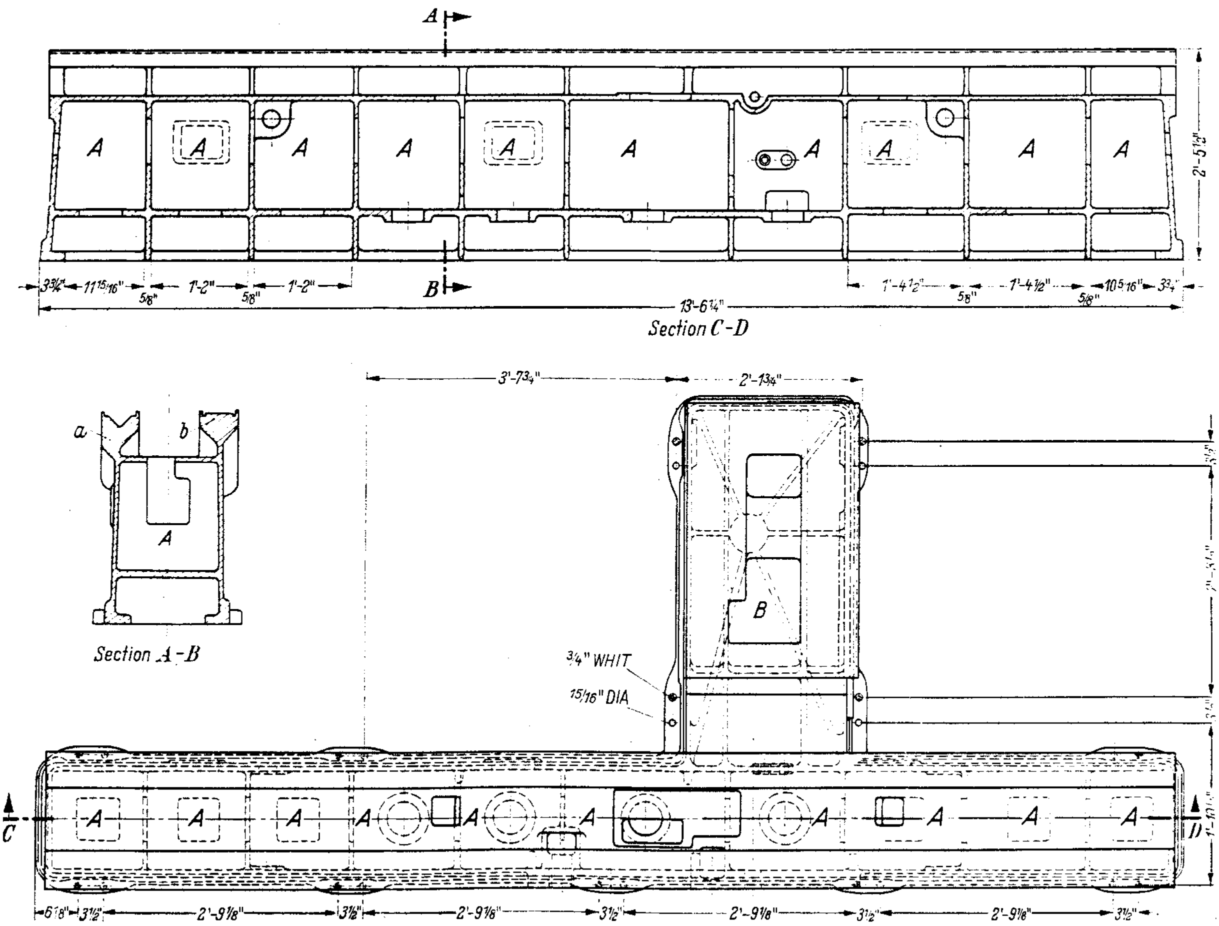


FIG 331. Bed of a cylindrical grinding machine (The Churchill Machine Tool Co. Ltd., Manchester).

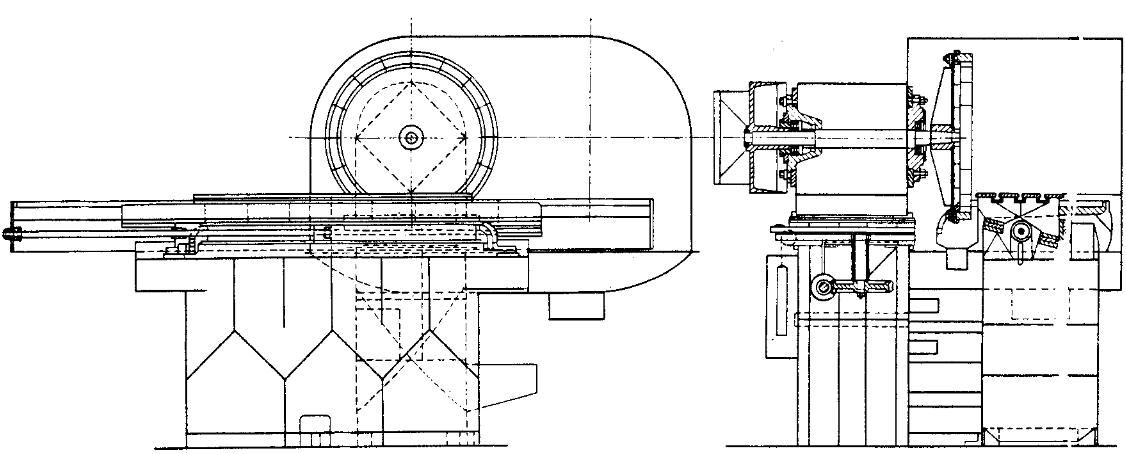


FIG. 332. Surface grinding machine (Diskus, Frankfurt a.M., Germany).

be encountered during milling operations. In order to facilitate the fitting of the head on three (otherwise overdetermined) ways, the milling head is supported on the additional slideway *a* by spring loaded rollers.

In the design of milling machine structures, the closed box section is preferably employed and the number of apertures usually reduced to a minimum. Unavoidable apertures are arranged outside the highly stressed zones.

Particular importance is to be given to the load transmission between slideways and the structure. In the case of the milling machine bed (Fig. 338), the centrally arranged feed screw is immersed in oil between two main ways which form a large "V". Possible tilting couples are counteracted by two narrow flat slideways which are arranged as far apart as

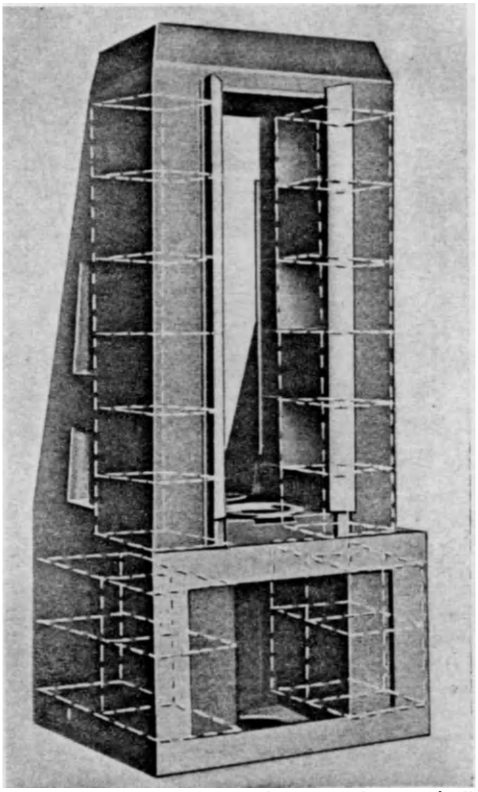


FIG. 333. Column of a surface grinding machine (cellular construction) (Diskus, Frankfurt a.M., Germany).

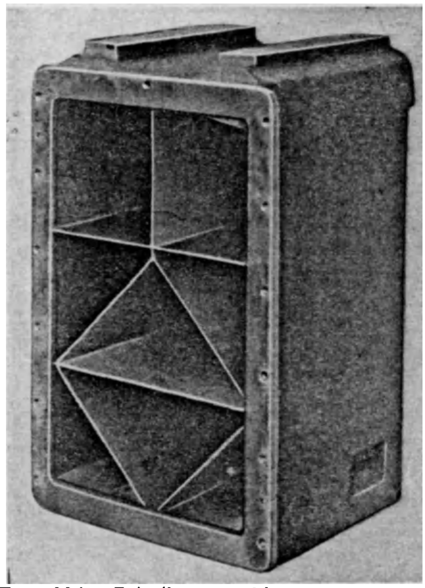


FIG. 334. Grinding machine structure in cellular construction (Diskus, Frankfurt a.M., Germany).

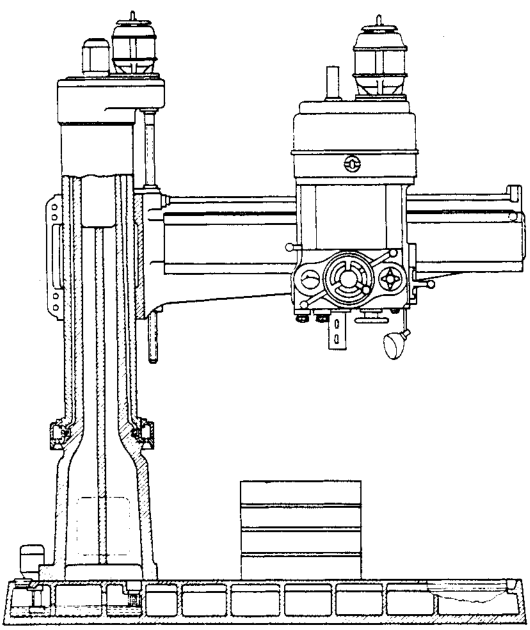


FIG. 335. Radial drilling machine (Raboma, Berlin, Germany).

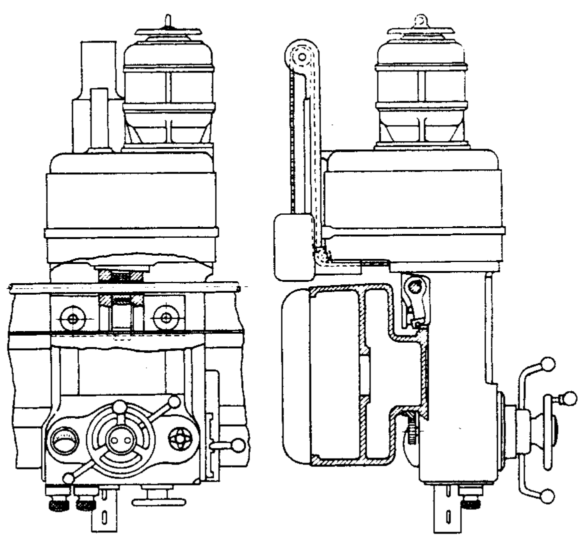


FIG. 336. Spindle head on the radial arm of a "Raboma" radial drilling machine.

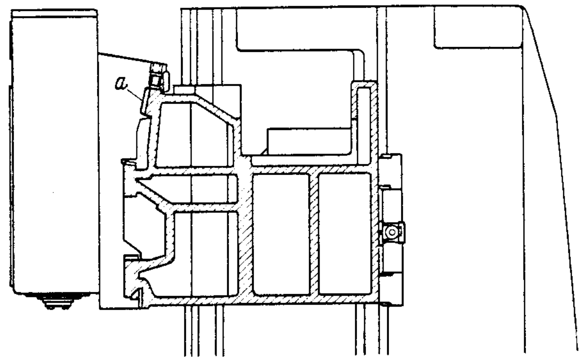


FIG. 337. Milling head slide of the heavy duty planing and milling machine (Fig. 310) (H. A. Waldrich, Siegen, Germany).

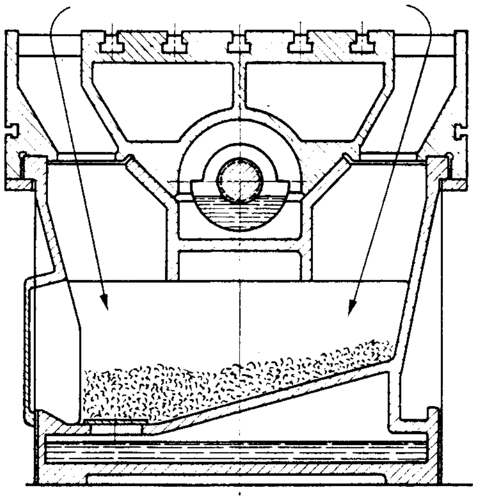


FIG. 338. Section through a milling machine bed (Heller, Nürtingen, Germany).

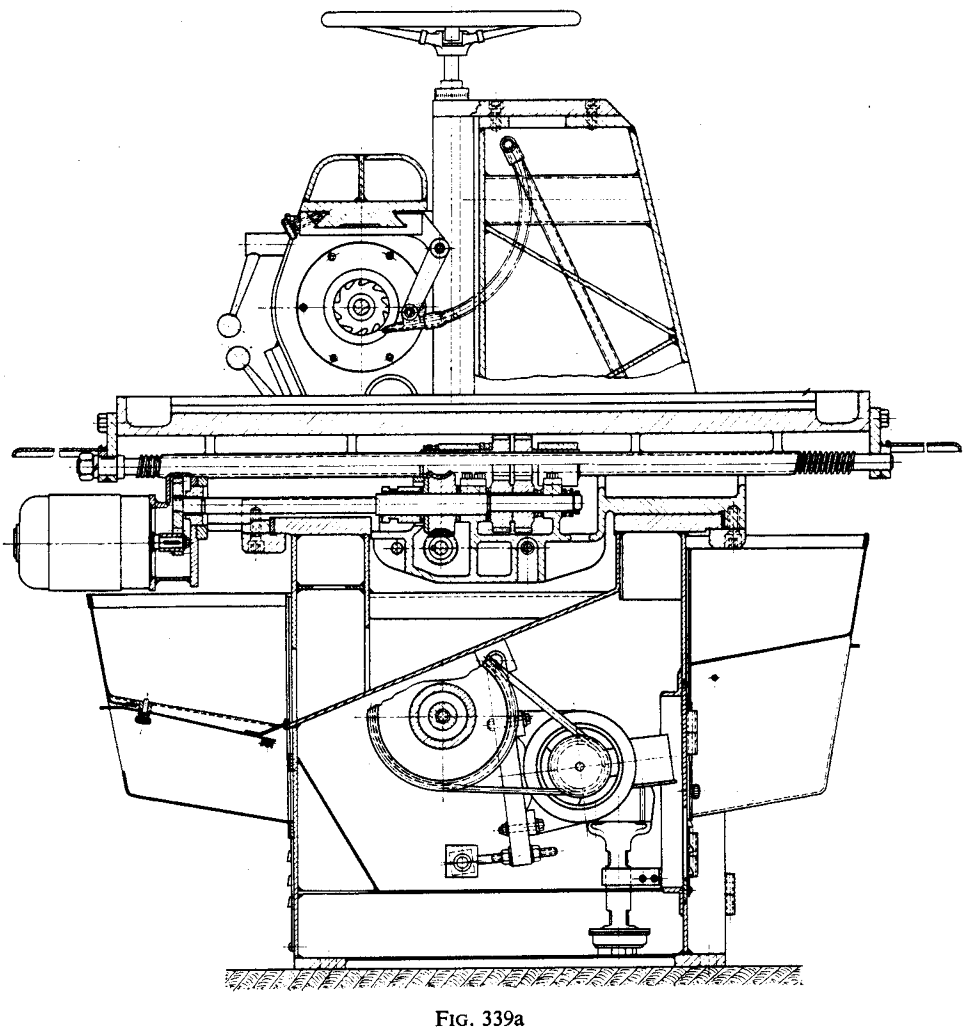


FIG. 339a

possible (large leverage), so that the forces acting on these ways are relatively small. The design of the cross-section of the bed and the table results in high stiffness without interfering with the free fall of chips and swarf.

A welded design of a milling machine structure may be shown for the example of the machine (Fig. 339, see also Figs. 133 and 134). It may be stressed that the welded design was not applied in this case unconditionally, but only when it was considered advantageous both from technical and economic considerations. The bed, the upright, the spindle head and the outer stay are welded. The saddle would be too complicated for welded design because of the many control and driving mechanisms which have to be built into it (see page 224). It is therefore designed as an iron casting, whilst the arbor supports are light alloy castings and thus of light weight and easy to handle. The bed (Fig. 340) is formed mainly by three $\frac{3}{8}$ in. thick plates *1*, *2* and *3*, which are bent into the required shapes and supported by a base frame formed by five 1 in. × $3\frac{1}{2}$ in. flats *4*. By bending the three $\frac{3}{8}$ in. plates, five costly welded seams are avoided. A sloping plate *5* built into the otherwise closed boxes allows chips and coolant to get away freely through the apertures *6* into the troughs (see Fig. 133). As mentioned earlier (see page 223), the slideways *7* and *8* are made of 0·4 per cent carbon steel, flame hardened and ground.

The outer wall of the upright (Fig. 341) is formed by a $\frac{3}{8}$ in. thick bent steel plate *1* and stiffened by diagonal webs *2*.

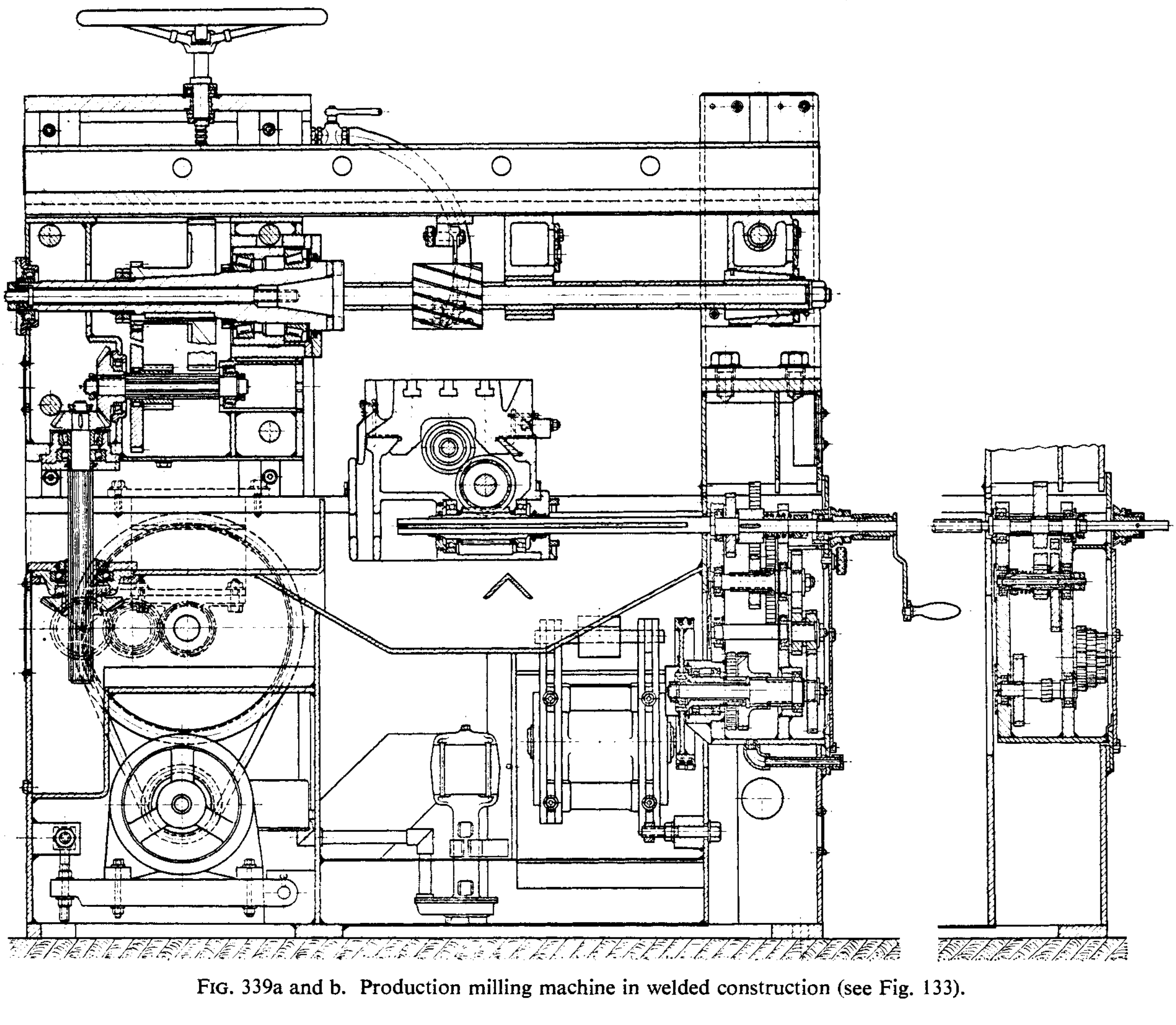


FIG. 339a and b. Production milling machine in welded construction (see Fig. 133).

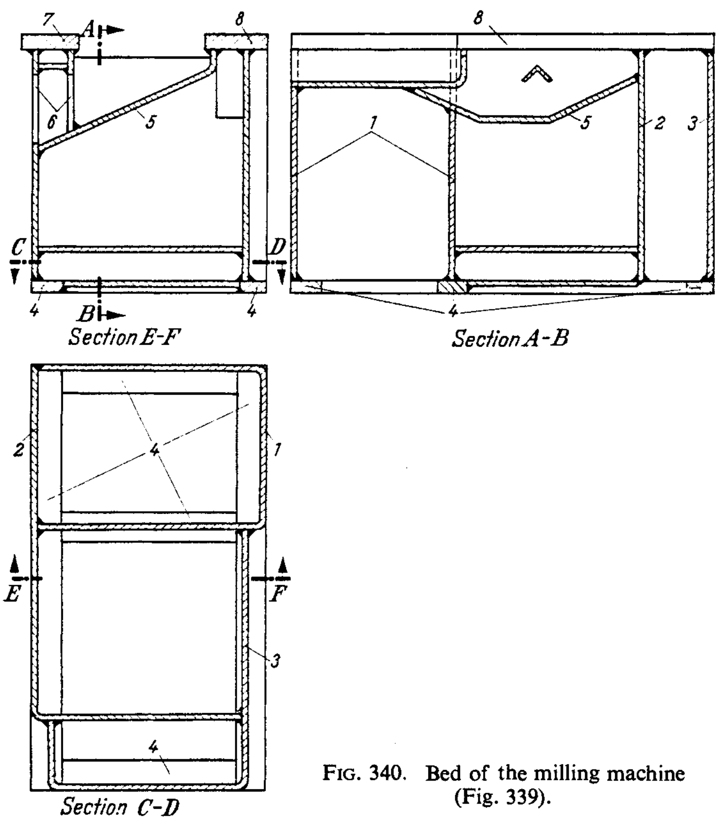


FIG. 340. Bed of the milling machine (Fig. 339).

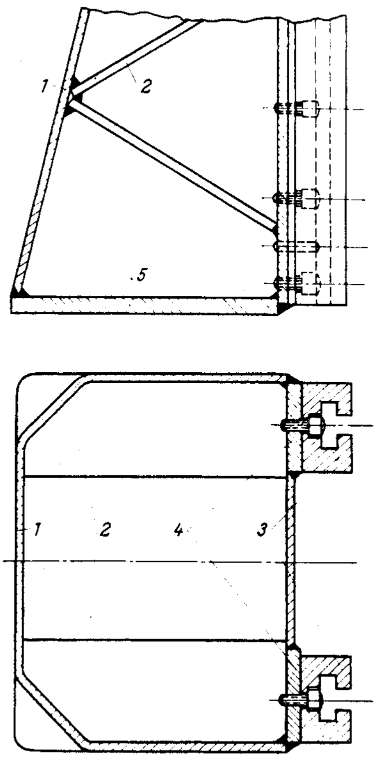


FIG. 341. Upright of the milling machine (Fig. 339).

Again as mentioned earlier (see page 223), the slideways *3* for the spindle head are made of cast iron, bolted and dowelled to the machined faces of the pads *4*. The base plate *5* is 1 in. thick as it has to be machined and fitted to the bed.

Good accessibility of the welded seams is an essential point which must not be overlooked during the design of a welded structure. An example is the outer stay (Fig. 342) of this milling machine. The Tee-slot for the clamping bolt of the overarm can be produced by rolled sections. However, if a channel is used (Fig. 342a), one of the three welds *1*, *2* or *3* would be inaccessible during assembly. By replacing the channel by two angles (Fig. 342b), this difficulty can be avoided and the box section closed by weld *4* as a final step in the assembly operation.

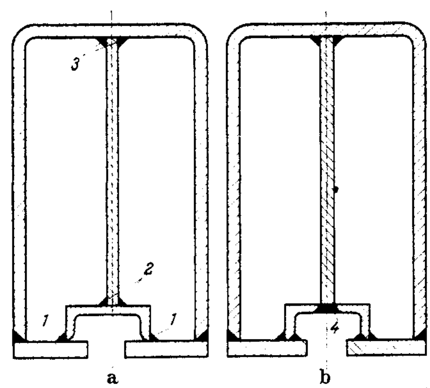


FIG. 342. Section through the outer stay of the milling machine (Fig. 339).

Finally, it may be mentioned that difficulties would also be encountered in the weld assembly of the bed (Fig. 340), unless it is assembled and welded upside down, in other words, by starting with the slideways and finishing with the base frame. The three side walls *1*, *2* and *3* must first be welded to the slideway plates *7* and *8*, the sloping plate *5* must then be inserted and the flats *4* finally welded to the side walls, otherwise the internal fillet welds, which connect plates *7* and *8* with plates *1*, *2* and *3*, would be inaccessible. These internal fillet welds are, however, very important because they ensure the strength of the joints which is essential in order to transmit the operational forces from the saddle to the bed section. The designer must always bear in mind the conditions which prevail during the assembly of a welded structure, and exact instructions must be given to the workshop in order to ensure that the finished structure will satisfy the designer's requirements.

2. SLIDEWAYS

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

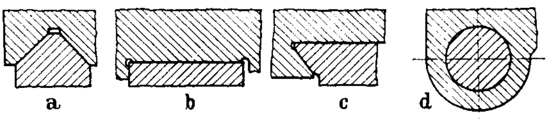


FIG. 343

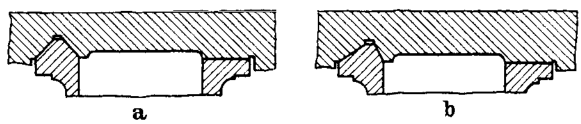


FIG. 344

(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.

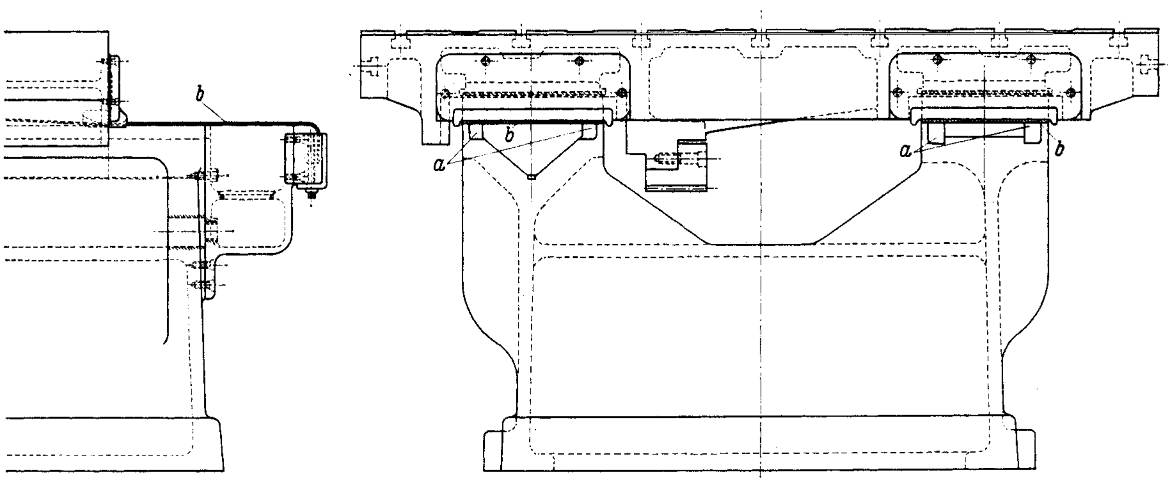


FIG. 345. Protected slideway for a grinding machine table (The Churchill Machine Tool Co. Ltd., Manchester). *a*—Oil channels; *b*—Protective cover.

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.

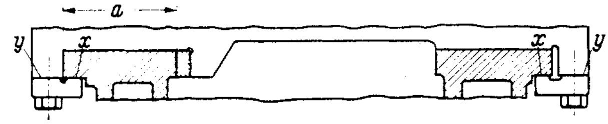


FIG. 346

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.

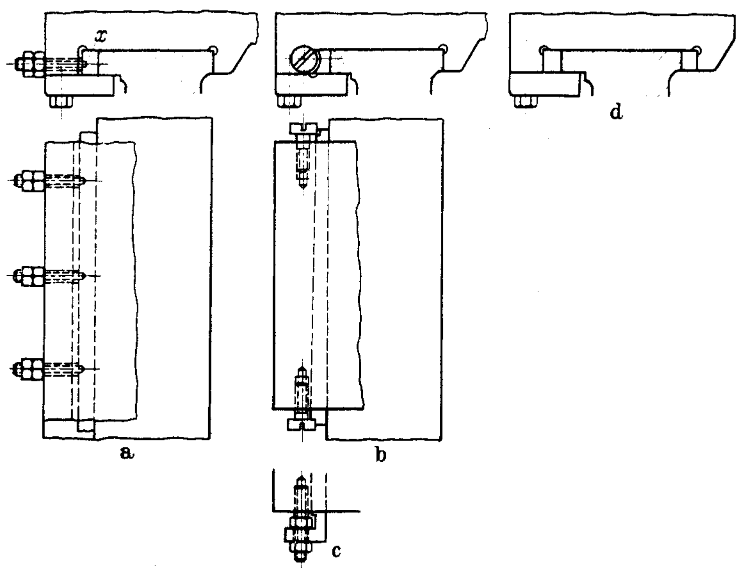


FIG. 347

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 $\frac{3}{8}$ 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).

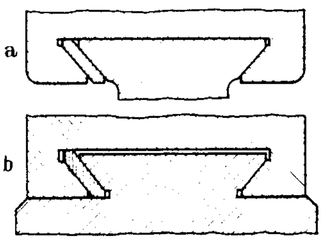


FIG. 348

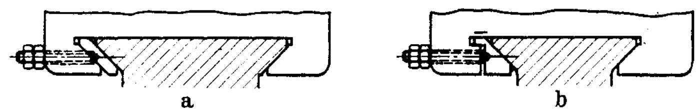


FIG. 349

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).

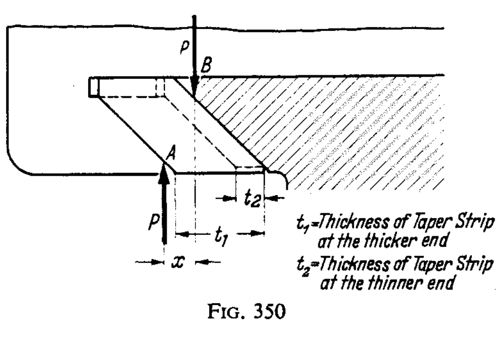


FIG. 350

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 ($P \times x$).

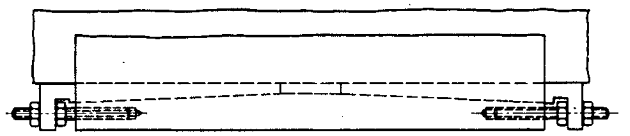


FIG. 351

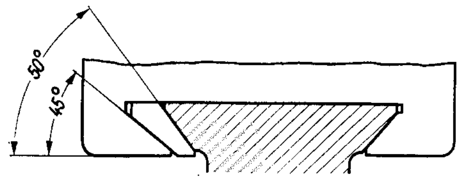


FIG. 352

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).

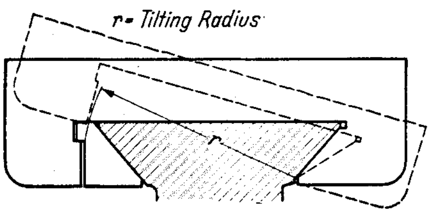


FIG. 353

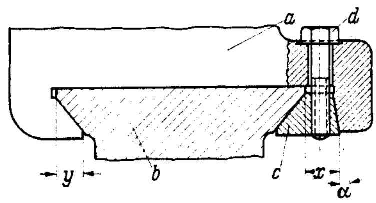


FIG. 354

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.

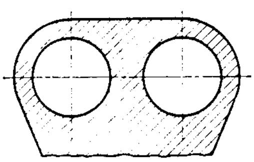


FIG. 355

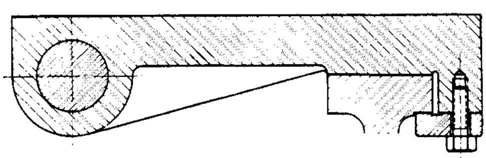


FIG. 356

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 instruments (Fig. 356), results in a restraint-free guiding device, as long as the flat surface is arranged radially to the cylindrical surface. Such an arrangement is, however, limited in length, because above a certain length stiffness considerations would require the diameter of the cylinder to be excessive. Moreover, larger diameters again require greater bearing lengths, thus increasing the difficulties. The design shown in Fig. 357,[83] in which a cylinder is supported over its full length, is a solution to this problem; however, the manufacture of the cylindrical element is not easy.

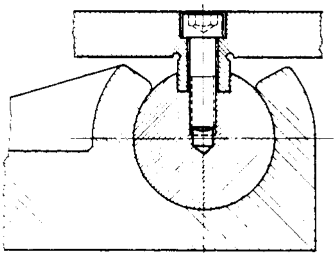


FIG. 357

(B) If the guiding and the guided surfaces are in direct contact, wear may be caused by several factors. Such wear is not always evenly distributed over the full length of the fixed part, its distribution depending upon the use of the slideway by the shorter, guided component in accordance with its relative position during various operations. This results in different wear conditions which, together with the manufacturing inaccuracies and deformations of the machine elements under load, reduce the working accuracy of the machine. The following factors influence the wear of slideways:

(1) The material properties of the fixed and moving elements
(2) The surface condition of the slideways
(3) The pressure exerted by the moving parts on the slideways
(4) Dirt on the slideways.

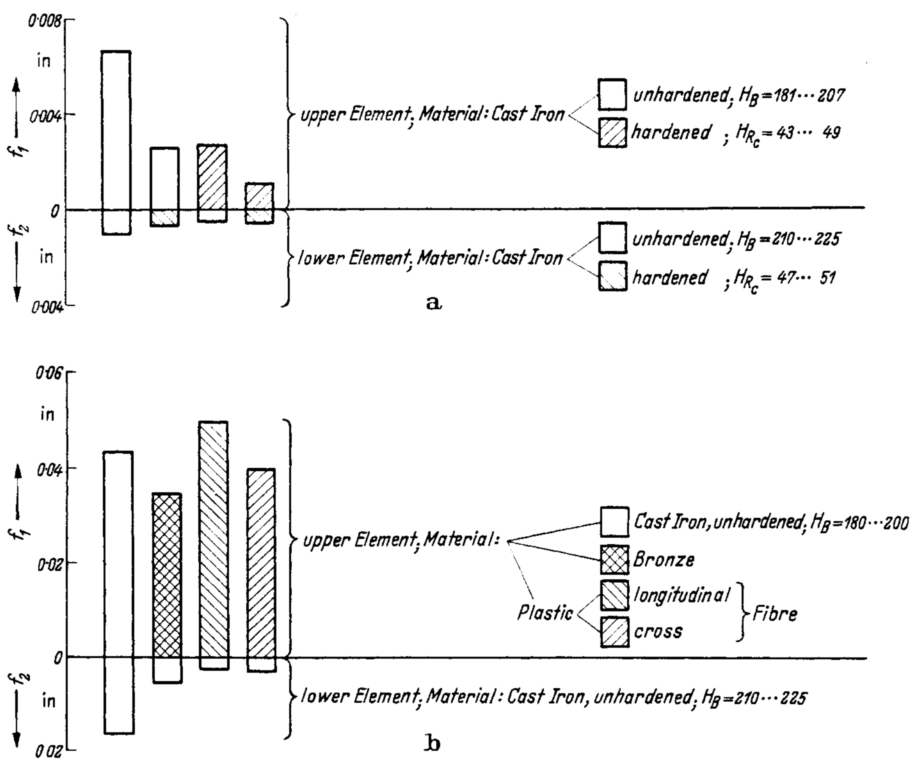


FIG. 358

These factors have been studied by P.E.R.A.,[84] Lapidus,[82] Saljé[85] and others. Figure 358a shows the wear measured by Lapidus for various combinations of hardened and non-hardened cast iron surfaces. The experiments were carried out under a surface pressure of about 140 lb/in^2 and a sliding velocity of 23 ft/min. The wear was measured after a total length of about 10,000 ft had been traversed. If the wear of the upper (moving) element is f_1, that of the lower (generally fixed) element f_2, then the total displacement in the direction normal to the sliding surface, i.e. the total wear $f_1 + f_2$ is greatest if two elements of non-hardened cast iron are sliding on each other, the larger amount of wear f_1 occurring on the upper part. If, for reasons of manufacturing difficulties, (for instance, hardened surfaces cannot be scraped and have to be ground) only one surface was hardened, Lapidus found that the total wear was less in the case of the sliding surface of the upper element being hardened and that of the lower one remaining unhardened.

The minimum wear occurs if both surfaces are hardened. If the upper element is made of bronze or plastic material, the wear of its sliding surface is little affected. However, the wear of the lower surface (cast iron 210–255 Brinell) is considerably less[86] (Fig. 358b). The use of plastic sliding surfaces, of course, reduces considerably the danger of seizing.

In order to obtain hard surfaces, the slideways of cast beds or saddles can be flame hardened (see page 223). If special steel strips are provided (see Fig. 319), these can be hardened and ground or hard surfaced and bolted to the machine parts in question[87] (see also page 228).

With increasing wear, the surface roughness of sliding surfaces decreases (Fig. 359).[85] After a

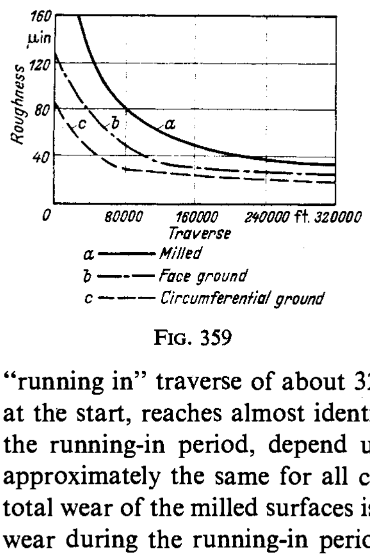


FIG. 359

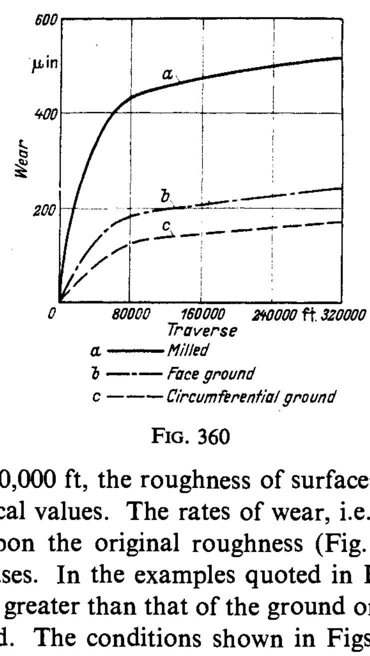


FIG. 360

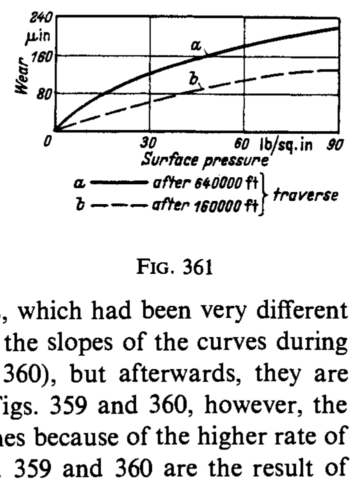


FIG. 361

"running in" traverse of about 320,000 ft, the roughness of surfaces, which had been very different at the start, reaches almost identical values. The rates of wear, i.e. the slopes of the curves during the running-in period, depend upon the original roughness (Fig. 360), but afterwards, they are approximately the same for all cases. In the examples quoted in Figs. 359 and 360, however, the total wear of the milled surfaces is greater than that of the ground ones because of the higher rate of wear during the running-in period. The conditions shown in Figs. 359 and 360 are the result of experiments with fixed and moving cast iron parts, carried out at a sliding velocity of 22 ft/min and a specific surface pressure of 155 lb/in^2.

The effect of the surface pressure as found by Saljé[85] can be seen from Fig. 361. This is the result of tests at a sliding velocity of 33 ft/min, with cast iron slideways and sliding parts. As a general rule, it can be stated that sliding surfaces should be dimensioned in such a manner that the mean specific pressure does not exceed 55 to 85 lb/in^2. In this connexion, it should be remembered that the forces exerted and the resulting specific pressures are not always constant and depend upon the position of the cutting edge relative to the slideways.

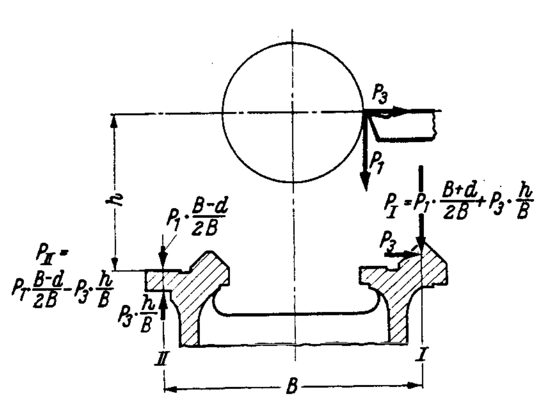


FIG. 362

Figure 362 shows the forces which act on the saddle slideways of a centre lathe and which are caused by the cutting force components P_1 and P_3. Their magnitudes depend upon the instantaneous position of the cutting edge, i.e. upon the turning diameter d. In the case of this example a Vee and a flat slideway result in the horizontal component P_3 being taken entirely by the front Vee slideway I. On this slideway, therefore, the vertical component is

$$P_I = P_1 . \frac{B+d}{2B} + P_3 . \frac{h}{B}$$

On the rear slideway II, only a vertical component acts, which according to the value of d, may be directed upwards $(-)$ or downwards $(+)$.

$$P_{II} = P_1 . \frac{B-d}{2B} - P_3 . \frac{h}{B}$$

The variation of these supporting forces with the turning diameter is shown in Fig. 363. In order to determine the specific surface pressures, it is necessary to find the forces acting at right angles to the

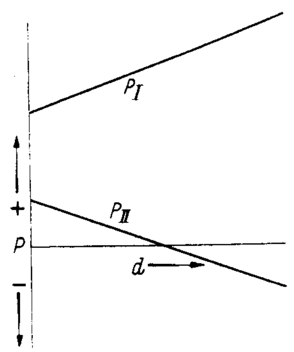


FIG. 363

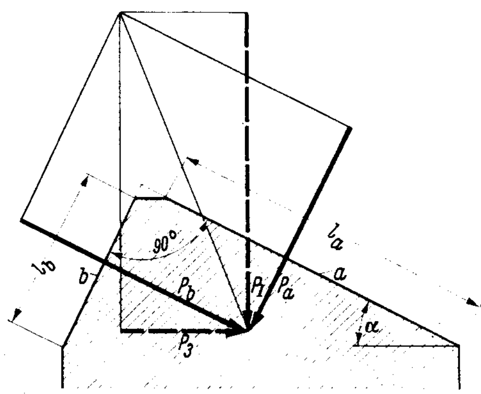


FIG. 364

sliding faces (Fig. 364). If the angle between the two sliding faces a and b is 90° and if sliding surface a is at a slope α to the horizontal, the conditions are:

$$P_a = P_I \cos\alpha - P_3 \sin\alpha$$

and

$$P_b = P_I \sin\alpha + P_3 \cos\alpha$$

By substitution, the following equations are obtained:

$$P_a = P_1 . \frac{B+d}{2B} . \cos\alpha - P_3\left(\sin\alpha - \frac{h}{B}\cos\alpha\right)$$

$$P_b = P_1 . \frac{B+d}{2B} . \sin\alpha + P_3\left(\cos\alpha + \frac{h}{B}\sin\alpha\right)$$

The normal forces acting on the guiding surfaces vary, therefore, with the turning diameter and depend upon the value of angle α. In order to prevent a lifting of the saddle, P_a must not be allowed to become negative (P_b is always positive). This means that in the case of the smallest turning diameter $d = 0$, the limiting condition is:

$$\frac{P_1}{2} . \cos\alpha > P_3 . \left(\sin\alpha - \frac{h}{B} . \cos\alpha\right)$$

Under the unfavourable assumption that $P_3 \approx 0{\cdot}4P_1$ and if $h/B = 0{\cdot}6$, $\tan\alpha$ must be less than 1·85 or $\alpha < 60°$ approximately.

Suppose the length of the sliding part is L. Under the assumption of a uniformly distributed load, the surface pressures are proportional to the normal forces P_a and P_b and inversely proportional to the width of the surfaces (l_a and l_b) (see Fig. 364). The specific pressure on surface a is, therefore:

$$p_a = \frac{P_a}{L.l_a}$$

and that on surface b:

$$p_b = \frac{P_b}{L.l_b}$$

Hence

$$\frac{p_a}{p_b} = \frac{P_a}{P_b}.\frac{l_b}{l_a}$$

With

$$\frac{l_b}{l_a} = \tan\alpha$$

$$\frac{p_a}{p_b} = \frac{P_a}{P_b}.\tan\alpha$$

The pressure on surface II is

$$p_{II} = \frac{P_{II}}{L.l_{II}} \quad \text{(see Fig. 365)}$$

$$\frac{p_a}{p_{II}} = \frac{P_a}{P_{II}}.\frac{l_{II}}{l_a}$$

It may be assumed that

$$l_{II} = l_a$$

hence,

$$\frac{p_a}{p_{II}} = \frac{P_a}{P_{II}}$$

The slideways are very rarely uniformly used and worn over their full length. If the wear was uniformly distributed over the full length of the slideways, this would have little effect on the working accuracy of the centre lathe. For this reason, the order of magnitude of the wear is less important than its irregularities measured over the full length of the slideways, because these result in a deviation of the path of the saddle from its originally intended shape. If, for instance, the greater amount of wear occurs in the middle of the bed a barrel-shaped workpiece would result, even if the workpiece and the bed were both infinitely stiff.

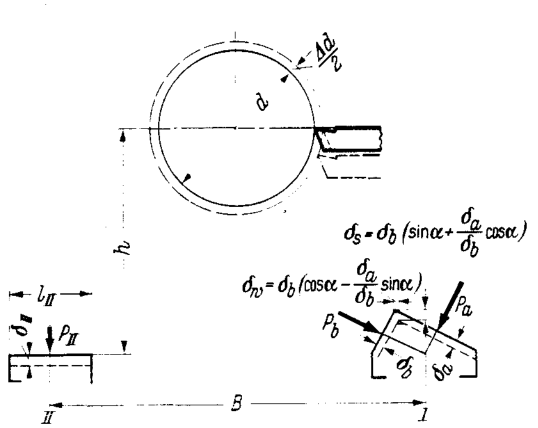


FIG. 365

The vertical deviation of the cutting edge from its intended straight line path has a relatively small influence upon the working accuracy but the diametral error caused by the horizontal displacement is twice the value of the displacement itself. For a lathe saddle the horizontal deviation caused by wear has been investigated by Lapidus.[82] It is due to:

(i) a horizontal displacement of the saddle δ_w (Fig. 365), and

(ii) a rotating movement of the saddle caused by the unequal wear in the vertical direction on the front (δ_s) and rear (δ_{II}) slideway. The horizontal deviation caused by (ii) is approximately

$$\frac{h}{B}(\delta_s - \delta_{II})$$

The total displacement of the cutting edge, which is equal to half of the diametral error Δd is therefore:

$$\frac{\Delta d}{2} = \delta_w + \frac{h}{B}(\delta_s - \delta_{II})$$

and the diametral error is:

$$\Delta d = 2\left[\delta_w + \frac{h}{B}(\delta_s - \delta_{II})\right]$$

If

$$\frac{\delta_a}{\delta_b} = \xi_1$$

and

$$\frac{\delta_a}{\delta_{II}} = \xi_2$$

the following equation can be obtained:

$$\Delta d = 2\delta_b\left[\cos\alpha - \xi_1 \sin\alpha + \frac{h}{B}\left(\xi_1 \cos\alpha + \sin\alpha - \frac{\xi_1}{\xi_2}\right)\right]$$

The diametral error caused by wear can only become zero for a certain relationship between ξ_1 and ξ_2. This relationship exists if:

$$\Delta d = 0$$

i.e. if

$$\cos\alpha - \xi_1 \sin\alpha + \frac{h}{B}\left(\xi_1 \cos\alpha + \sin\alpha - \frac{\xi_1}{\xi_2}\right) = 0$$

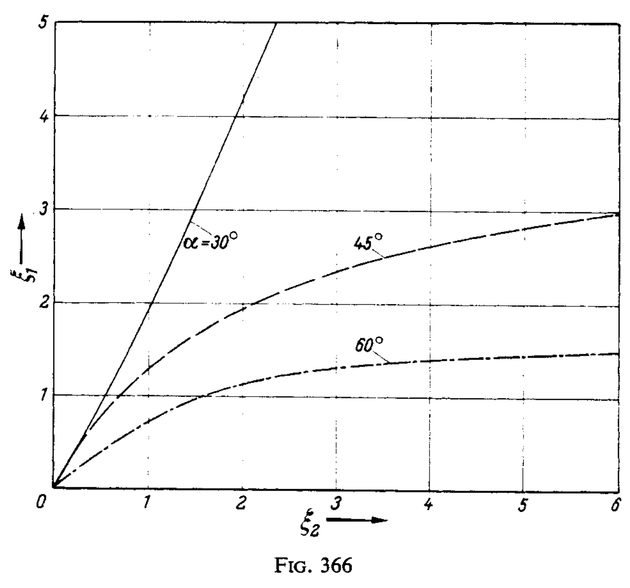


FIG. 366

The diametral error can therefore be zero only if:

$$\xi_1 = \frac{\xi_2[\cos\alpha + (h/B)\sin\alpha]}{\xi_2[\sin\alpha - (h/B)\cos\alpha] + h/B}$$

This equation is shown graphically in Fig. 366 for the case of $h/B = 0{\cdot}6$ and for $\alpha = 30°, 45°$ and $60°$.

As a first approximation, the wear can be assumed to be proportional to the specific surface pressure. The ratios of the wear values are thus equal to those of the specific pressures:

$$\xi_1 = \frac{p_a}{p_b}$$

$$\xi_2 = \frac{p_a}{p_{II}}$$

with

$$\frac{p_a}{p_b} = \frac{P_a}{P_b} . \tan\alpha \qquad \text{(see page 246)}$$

and

$$\frac{p_a}{p_{II}} = \frac{P_a}{P_{II}} \qquad \text{(see page 246)}$$

$$\xi_1 = \frac{P_a}{P_b} . \tan\alpha$$

$$\xi_2 = \frac{P_a}{P_{II}}$$

P_a, P_b and P_{II} vary with the turning diameter; Fig. 367 shows values for ξ_1 and ξ_2 which, for

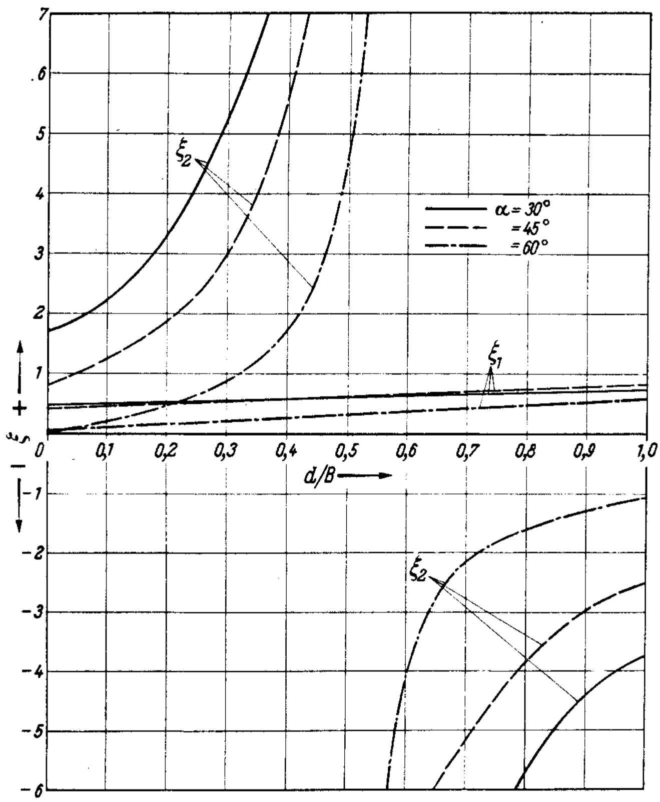


FIG. 367

the example of a centre lathe ($h/B = 0{\cdot}6$) and a ratio $P_3/P_1 = 0{\cdot}4$ are plotted as a function of the ratio (diameter/width) for $\alpha = 30°$, $45°$ and $60°$. The relationship of ξ_1 and ξ_2 can be calculated from the foregoing equations:

$$\xi_1 = \frac{P_a}{P_b}.\tan\alpha$$

$$\xi_2 = \frac{P_a}{P_{II}}$$

Hence

$$\frac{\xi_1}{\xi_2} = \frac{P_{II}}{P_b}.\tan\alpha$$

With

$$P_{II} = P_1.\frac{B-d}{2B} - P_3.\frac{h}{B}$$

$$P_b = P_1.\frac{B+d}{2B}.\sin\alpha + P_3\left(\cos\alpha + \frac{h}{B}.\sin\alpha\right) \qquad \text{(see page 245)}$$

and for

$$\frac{h}{B} = 0{\cdot}6 \quad \text{and} \quad P_3 = 0{\cdot}4P_1$$

$$\frac{\xi_1}{\xi_2} = \frac{0{\cdot}52 - d/B}{1{\cdot}48\sin\alpha + 0{\cdot}8\cos\alpha + (d/B)\sin\alpha}.\tan\alpha$$

Hence

$$\xi_1 = \xi_2\tan\alpha.\frac{0{\cdot}52 - d/B}{1{\cdot}48\sin\alpha + 0{\cdot}8\cos\alpha + (d/B)\sin\alpha.}$$

The actual relationship between ξ_1 and ξ_2 does not satisfy the requirements shown in Fig. 366, because it depends upon the turning diameter and can even become negative if d/B is greater than 0·52.

It is possible to reduce the effect of wear upon the diametral error by using two Vee slideways (Fig. 368). In the equation for the diametral error (see page 247), the contribution of the purely horizontal deviation is $\delta_w = 2\delta_b(\cos\alpha - \xi_1\sin\alpha)$, whilst the contribution of the tilting deviation of the saddle is:

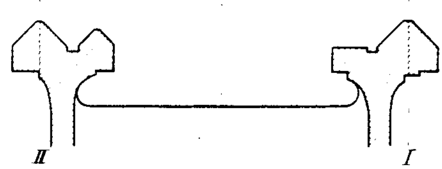


FIG. 368

$$\delta_w' = 2\delta_b.\frac{h}{B}\left(\xi_1\cos\alpha + \sin\alpha - \frac{\xi_1}{\xi_2}\right)$$

$$\frac{\delta_w'}{\delta_w} = \frac{h}{B}.\frac{\xi_1\cos\alpha + \sin\alpha - \xi_1/\xi_2}{\cos\alpha - \xi_1\sin\alpha}$$

For the case of a 45° Vee slideway ($\sin\alpha = \cos\alpha \approx 0{\cdot}7$)

$$\frac{\delta_w'}{\delta_w} = \frac{h}{B}.\frac{(1 + \xi_1 - 1{\cdot}4\xi_1/\xi_2)}{1 - \xi_1}$$

and for

$$\frac{h}{B} = 0{\cdot}6$$

$$\delta_w' = 0{\cdot}6.\frac{(1 + \xi_1 - 1{\cdot}4.\xi_1/\xi_2)}{1 - \xi_1}.\delta_w$$

If two Vee slideways are provided (Fig. 368), not only P_1 but also P_3 is distributed over both Vees and the additional wear δ_{II} of the rear slideway *II* results in the values $\delta_s - \delta_{II}$ and $\delta'_w = (h/B)(\delta_s - \delta_{II})$ being reduced. This is the reason why some designers are still using this arrangement for high precision centre lathes although it is kinematically unsound (see Fig. 321).

It can be shown that the effect of the turning diameter upon the diametral error caused by wear may be considerably reduced by a suitable design arrangement of the slideways.[88] Such an arrangement (Fig. 369) has also the advantage that the slideways are outside the fall of the chips. As a

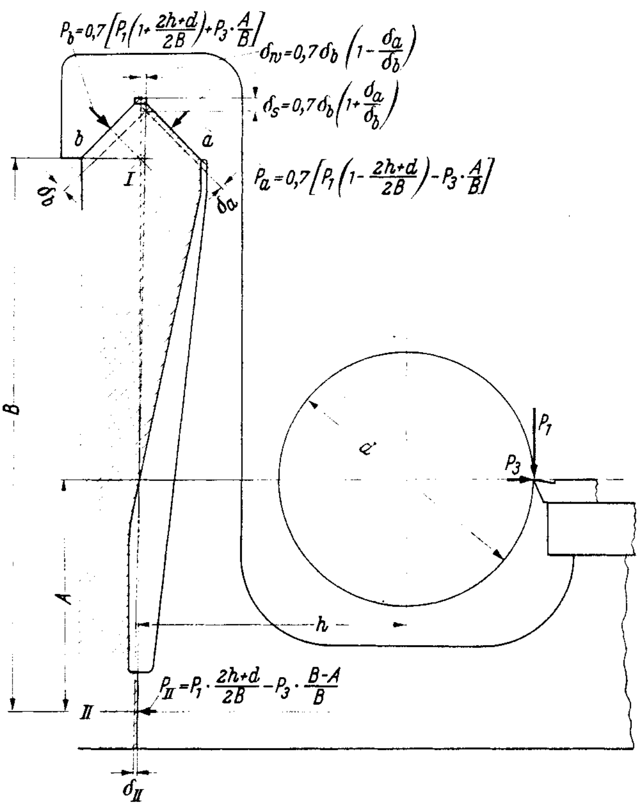


FIG. 369

detailed discussion of these conditions can be found in Seybold's publication, the conditions will be shown only approximately in this Chapter. The horizontal deviation of the cutting tool edge, which determines the working accuracy, can be assumed to be

$$\frac{\Delta d}{2} = \delta_w.\frac{A}{B} - \delta_{II}.\frac{B-A}{B}$$

and the resulting diametral error

$$\Delta d = \frac{2}{B}.[A.\delta_w - (B-A).\delta_{II}]$$

If, again, the wear is assumed to be proportional to the specific surface pressure (see page 248), and if it may be further assumed that the guiding surfaces *a*, *b* and *II* are of equal length and width, then:

$$\frac{p_a}{p_b} = \frac{\delta_a}{\delta_b} = \frac{P_a}{P_b} = \xi_1$$

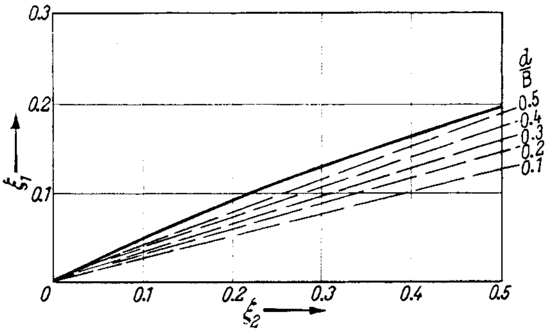


FIG. 370

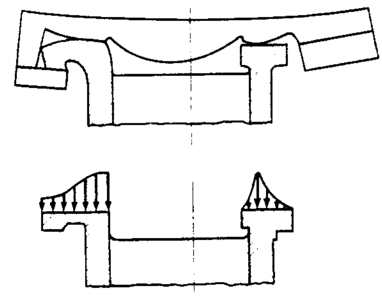


FIG. 371

$$\frac{p_a}{p_{II}} = \frac{\delta_a}{\delta_{II}} = \frac{P_a}{P_{II}} = \xi_2$$

$$\Delta d = \frac{2}{B}[0{\cdot}7A\delta_b(1 - \xi_1) - (B - A)\delta_{II}]$$

$$= \frac{2\delta_b}{B}\left[0{\cdot}7A(1 - \xi_1) - (B - A)\frac{\xi_1}{\xi_2}\right]$$

In order to satisfy the condition $\Delta d = 0$, it is necessary that:

$$0{\cdot}7A(1 - \xi_1) = (B - A)\frac{\xi_1}{\xi_2}$$

Hence:

$$\xi_1 = \frac{0{\cdot}7A\xi_2}{0{\cdot}7A\xi_2 + B - A}$$

$$= \frac{\xi_2}{\xi_2 + 1{\cdot}4B/A - 1{\cdot}4}$$

This relationship is shown as a full line in Fig. 370 for the case of $B/A = 2{\cdot}5$ (see also Fig. 366).
With

$$P_a = 0{\cdot}7\left[P_1\left(1 - \frac{2h + d}{2B}\right) - P_3 \, . \frac{A}{B}\right]$$

$$P_b = 0{\cdot}7\left[P_1\left(1 + \frac{2h + d}{2B}\right) + P_3 . \frac{A}{B}\right]$$

$$P_{II} = P_1 . \frac{2h + d}{2B} - P_3 . \frac{B - A}{B}$$

and for $h/B = 0{\cdot}5$ and $P_3 = 0{\cdot}4P_1$ (see page 249):

$$\xi_1 = \frac{P_a}{P_b} = \frac{1 - d/B - 0{\cdot}8A/B}{3 + d/B + 0{\cdot}8A/B}$$

$$\xi_2 = \frac{P_a}{P_{II}} = \frac{0{\cdot}7 - 0{\cdot}7d/B - 0{\cdot}56A/B}{0{\cdot}2 + d/B + 0{\cdot}8A/B}$$

Hence

$$\frac{\xi_1}{\xi_2} = \frac{(1 - d/B - 0{\cdot}8A/B).(0{\cdot}2 + d/B + 0{\cdot}8A/B)}{(3 + d/B + 0{\cdot}8A/B).(1 - d/B - 0{\cdot}8A/B).0{\cdot}7}$$

In the case of $B/A = 2{\cdot}5$

$$\xi_1 = \xi_2 \cdot \frac{1{\cdot}4(0{\cdot}52 + d/B)}{3{\cdot}32 + d/B}$$

The straight lines shown dotted in Fig. 370 represent this relationship for values of $d/B = 0{\cdot}1$, $0{\cdot}2$, $0{\cdot}3$, $0{\cdot}4$ and $0{\cdot}5$. With increasing ratio d/B the conditions approximate the requirements for $\Delta d = 0$. In other words, the detrimental effect of slideway wear upon the working accuracy decreases with increasing turning diameter. Although, therefore, in this case the requirements for $\Delta d = 0$ are almost completely satisfied in the case of $d/B = 0{\cdot}5$, even in this design the effect of the turning diameter is not fully eliminated.

The preceding considerations assumed that the specific surface pressure was evenly distributed over the guiding surfaces. Such an assumption is correct only if the guiding and guided parts of the machine are sufficiently stiff. For the example of a lathe saddle Saljé[85] has shown the effect of deformations upon the pressure distribution (Fig. 371). If some of the elements are insufficiently stiff, pressure peaks result which can reach such values that seizing may occur. The danger of such pressure peaks can be reduced by the application of narrower and therefore stiffer slideways.

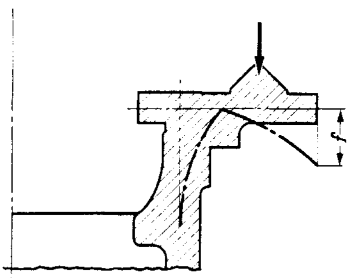


FIG. 372. f—Deflexion of the slideway.

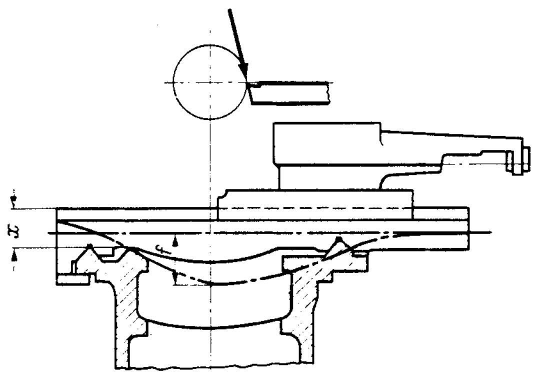


FIG. 373. f—Deflexion of the saddle; x—Minimum depth of the saddle section.

However, the reduction in width must not result in excessively high mean surface pressures. The importance of maximum possible stiffness again becomes evident in this case. In particular, the danger encountered in an overhanging front slideway (Fig. 372)* and of a weak cross-slide section (Fig. 373)* may be emphasized. The minimum depth x of the cross-slide section above the slideway has to be small for reasons of stability, but it is critical from the point of view of stiffness.

Dirt on the slideway surfaces increases the wear. Lapidus[82] found that the use of scrapers resulted in a wear reduction of more than 60 per cent. The slideway surfaces must, therefore, be carefully protected against foreign matter (chips, swarf, dirt, etc.).† In practice, three solutions of this problem may be encountered. They are (i) the protection of the slideway surfaces, (ii) the use of seals and scrapers which prevent the entry of foreign matter between the guiding and the guided surfaces, and (iii) the insertion of a replaceable intermediate member (steel tape) between the guiding and the guided surface.

If overhang of the moving parts does not result in inadmissible deflexions, the fixed slideway (A, length L_A) can be designed shorter than the moving part (B,

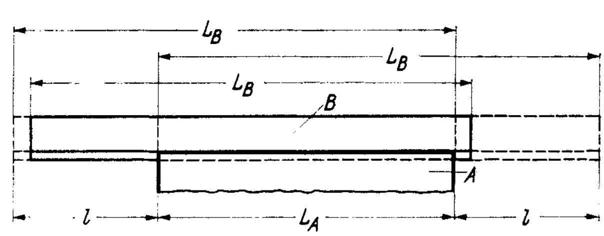


FIG. 374

* See ref. 28 Part I.
† Summary of Methods of Slideway Protection for Machine Tools, P.E.R.A. Research Report No. 38.

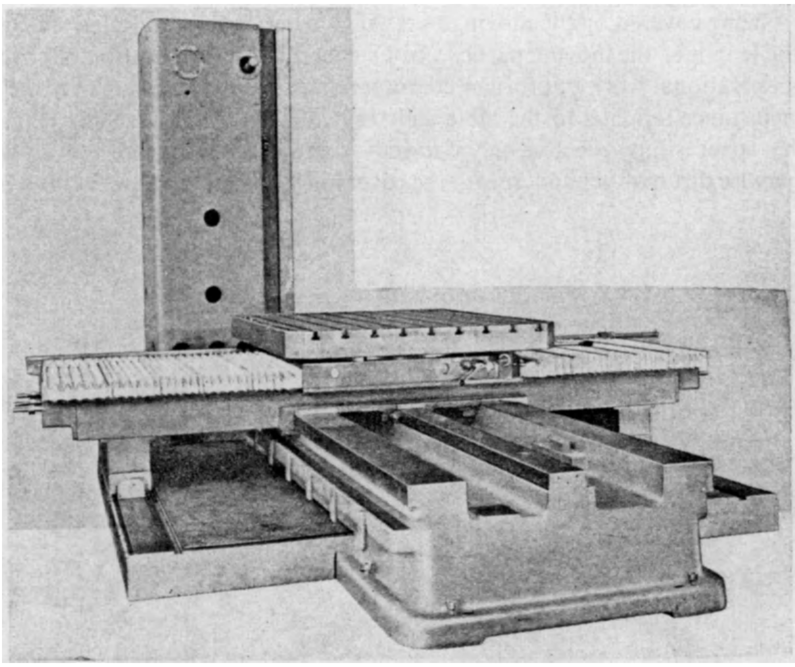


FIG. 375. Slideway protection of a Plauert–Wetzel horizontal boring machine (Vereinigte Werkzeugmaschinenfabriken, Frankfurt a.M., Germany).

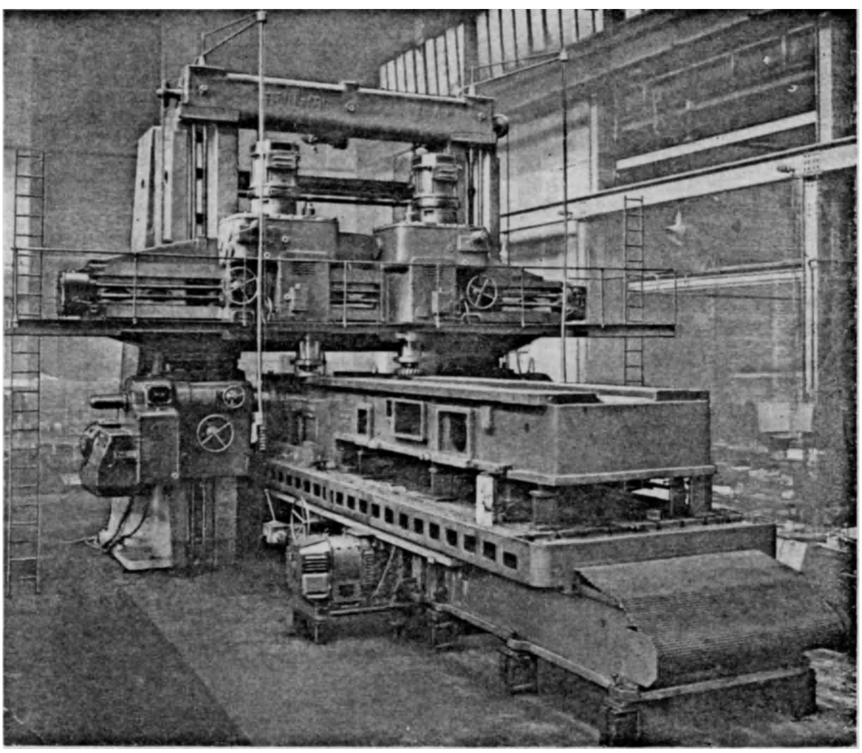


FIG. 376. Plano-milling machine with covered slideways (Kendall & Gent Ltd., Manchester).

length L_B) 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.

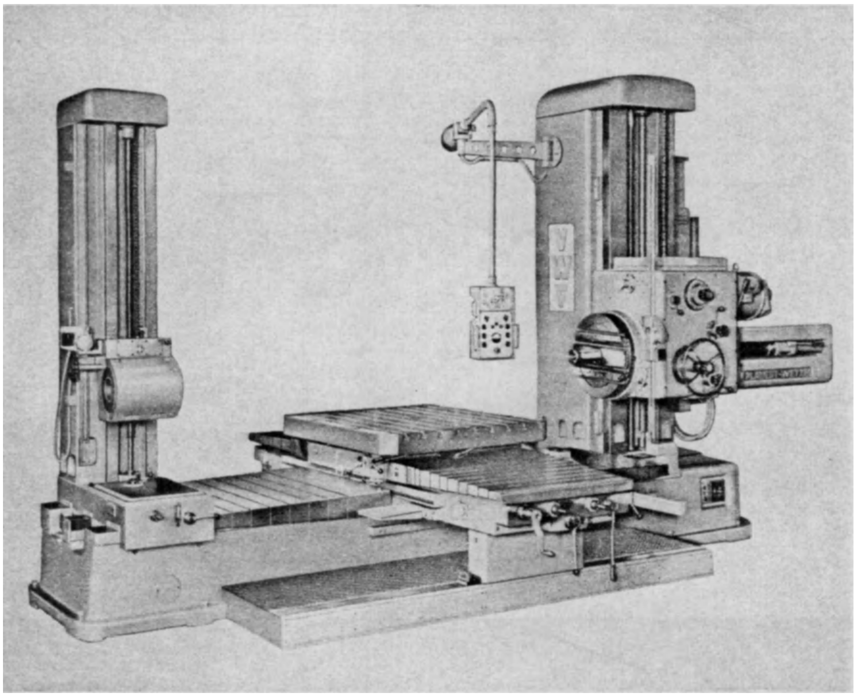


FIG. 377. Protection of slideways by means of telescopic cover plates on a Plauert–Wetzel horizontal boring machine (Vereinigte Werkzeugmaschinenfabriken, Frankfurt a.M., Germany).

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.

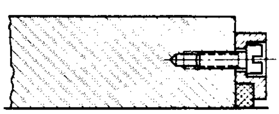


FIG. 378

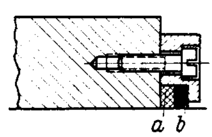


FIG. 379

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

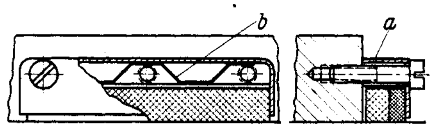


FIG. 380

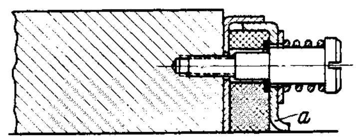


FIG. 381

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.

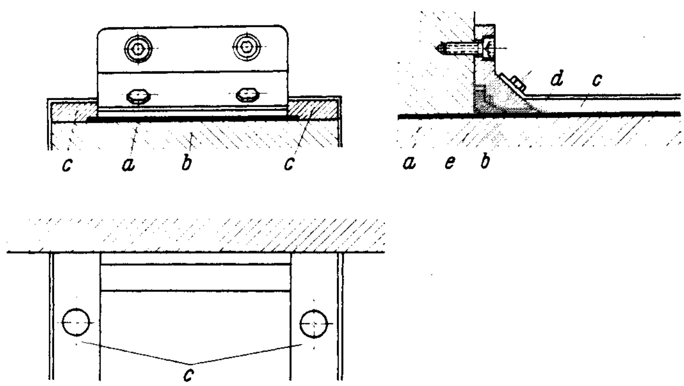


FIG. 382. Arrangement of scrapers on flat slideways (Scharmann). *a*—hardened steel tape; *b*—slideway casting; *c*—lateral steel strips; *d*—scraper fitted without play between *c*; *e*—"hydrofit" seal scraper.

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 $v < v_a$. The friction coefficient μ increases, of course, again with further increasing velocity $v > v_a$ (Fig. 383).[90] Over the range $v < v_a$, 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

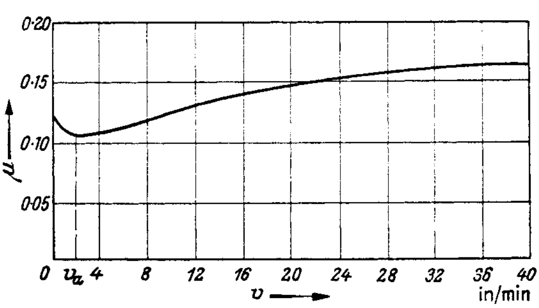


FIG. 383. Friction coefficient between the slideway and the saddle of a horizontal boring machine, as a function of the saddle speed.

friction (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

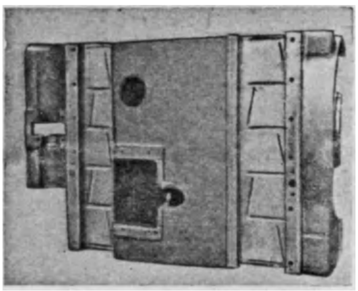


FIG. 384. Arrangement of oil grooves in the sliding surfaces of a horizontal boring machine (H. W. Kearns & Co. Ltd., Broadheath). Depth of groove ⅛ in.; Width of groove ⅜ in.

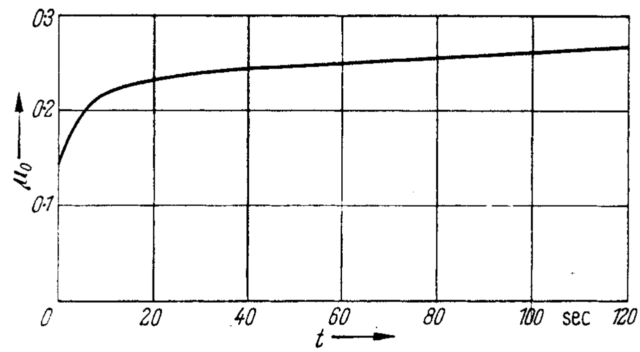


FIG. 385

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.

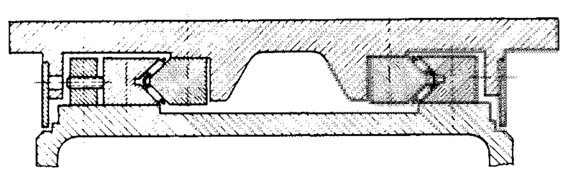


FIG. 386. Accurate setting made possible by roller-slides of a centreless grinding machine (Herminghausen).

(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

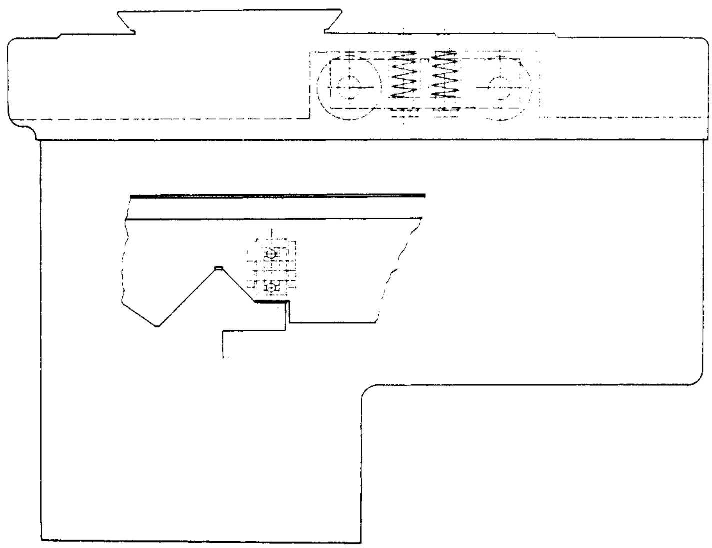


FIG. 387. Spring loaded rollers carry the lathe saddle, whilst the cutting force compresses the springs and is thus taken by the Vee-slide (Dean, Smith & Grace Ltd., Keighley).

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).

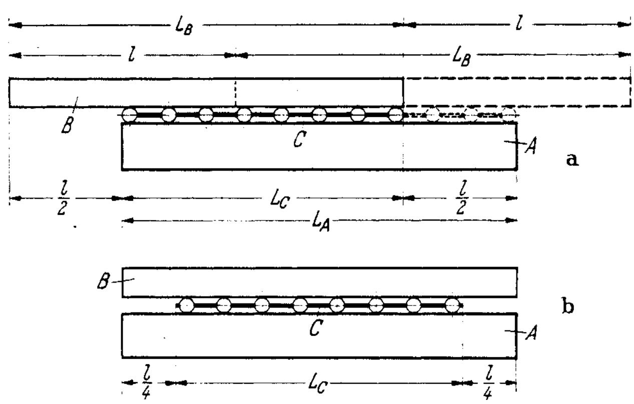


FIG. 388

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

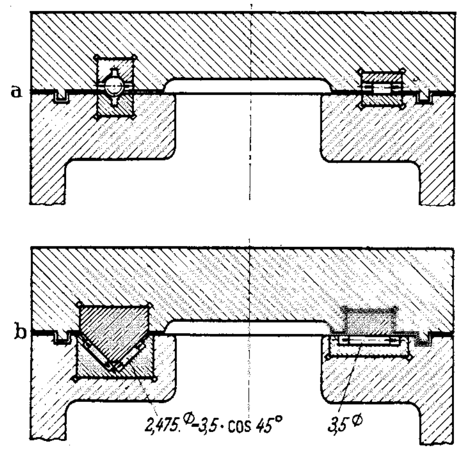


FIG. 389. Open arrangement. *a*) Ball and roller slide (Robert Kling, Wetzlar, Germany); *b*) Needles of different diameters (Industrie Werke Schaeffler, nr. Nürnberg, Germany).

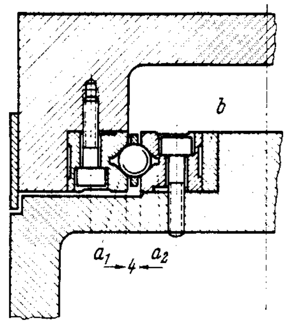


FIG. 390. Ball slide (W. Schneeberger A.G., Bern, Switzerland) a_1 and a_2—hardened and ground races; *b*—Adjusting strip.

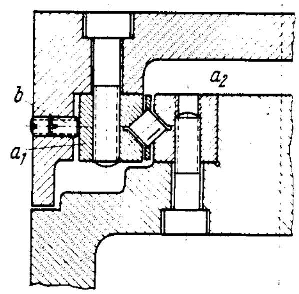


FIG. 391. Roller slide (W. Schneeberger A.G., Bern, Switzerland). a_1 and a_2—hardened and ground races; *b*—Adjusting screw.

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)

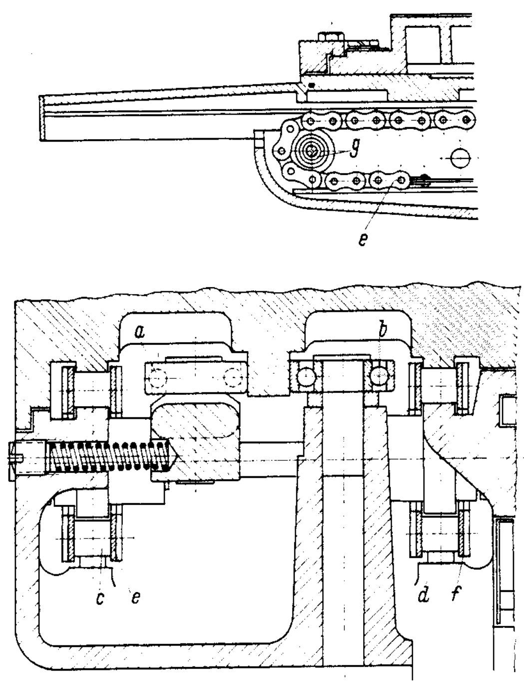


FIG. 392. Ball and roller bearing slides with precision roller chain (Ludw. Loewe and Co. A.G., Berlin).

is fixed to the moving slide (*A*, Fig. 393). The rollers run on the two surfaces of the main slideway *b*, which also lie at 90° to each other and are fixed to the bed *B*. The stroke of such a sliding arrangement is unlimited. Play adjustment can be obtained by accurate setting of the rail piece *a* on the moving slide *A*.

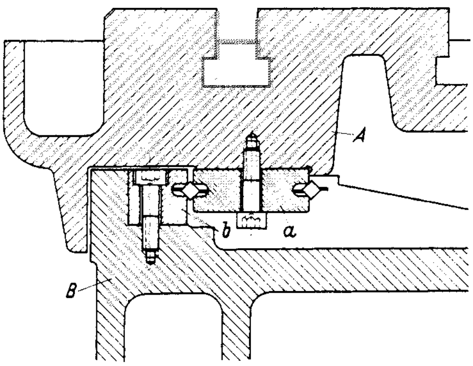


FIG. 393. Roller slide for long traverses (SKF).

(ii) *Hydrostatically lubricated slideways*[94] can be designed either with a compressible medium such as compressed air or with a practically incompressible medium such as oil for transmitting the load between the guiding and the guided surfaces. The stiffness, i.e. the minimum possible variation of the film thickness under a varying load is decisive and in cases in which variable cutting forces are encountered an incompressible lubricant would appear superior. In addition, with the use of compressed air as a lubricant, it is necessary to keep the humidity of the air to a minimum in order to prevent corrosion of the sliding surfaces.

Although some very interesting work has been carried out in the field of air lubricated slideways,[95] oil lubrication appears to be preferable in cases of high and pulsating loads. In this case the oil film carries the load exerted on the slideway, i.e. the weight of the moving slide plus the cutting force. A state of fluid friction is thus ensured in which the frictional resistance increases with increasing oil viscosity and sliding speed. As long as metallic contact between the two sliding surfaces is prevented,

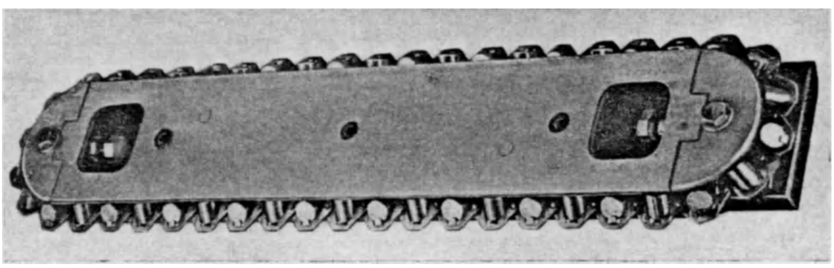


FIG. 394. Guide rail for SKF cross roller chain.

wear cannot take place. It is, of course, necessary to make certain that any oil leaking away from the sliding surfaces is immediately replenished, that no dirt either from external sources or carried in the oil can get between the sliding surfaces and that any sliding movement can be initiated only when the required oil pressure between the sliding surfaces is reached. The following parameters affect the design of such slideway arrangements:[96]

(a) Load
(b) Shape of the sliding surface
(c) Size of the sliding surface
(d) Distance between the sliding surfaces (oil film thickness)
(e) Oil pressure supplied by the pump
(f) Oil consumption
(g) Manufacturing problems of sliding surfaces, oil pockets, etc.
(h) Economic considerations.

The simplest arrangement of a hydrostatic slideway is shown in schematic form in Fig. 395. The pump supplies the oil under pressure p_0 and this pressure is kept constant by an overload valve. The oil pressure is reduced (resistance R_0) to p_1 and under this pressure it is supplied to the oil pocket *a*. From there, the oil is compressed between the sliding surfaces (upper surface *b*, lower surface c), from which it escapes eventually to atmospheric pressure. Inside the oil pocket a, the pressure may be assumed to be uniform. However, between the two sliding surfaces, where the gap (oil film thickness h) resists the oil flow, the pressure drops from p_1 to atmospheric pressure, and the load carrying capacity can be determined only if the pressure distribution between the sliding surfaces is known. A load coefficient f_A can be assumed for calculating the load carrying capacity P of a

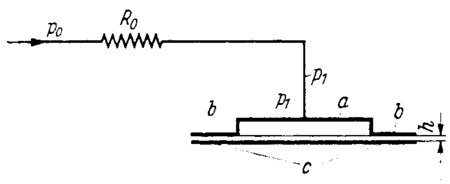


FIG. 395

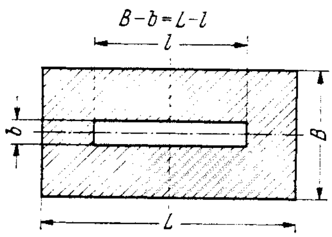


FIG. 396

sliding surface $A = B \times L$ (see Fig. 396), so that $P = p_1 \times A \times f_A$. The load coefficient is, therefore $f_A = A_{eff}/A$ where A_{eff} represents the effective magnitude of an area which would carry the load P if the pressure p_1 was evenly distributed. Typical values of load coefficients, for the case of a rectangular sliding surface with a central oil pocket (Fig. 396) are shown in Fig. 397.[97] The shape of the sliding surfaces and the gap h between them determine the resistance R_1 which is exerted against the oil flowing under the pressure p_1 to atmosphere. If the oil film is compressed by an amount Δh under a load P, the following expression can be established:

$$P = \frac{p_0 . A . f_A . R_1/R_0}{(1 - \Delta h/h)^3 + R_1/R_0}$$

If the weight of the moving part carried by the slideways is W, and if the oil film thickness under the load W serves as a starting point for the calculation ($\Delta h = 0$), then:

$$W = \frac{p_0 . A . f_A . R_1/R_0}{1 + R_1/R_0}$$

and

$$\frac{P}{W} = \frac{1 + R_1/R_0}{(1 - \Delta h/h)^3 + R_1/R_0}$$

It is known that maximum stiffness is obtained if $R_1 = R_0$, i.e. $R_1/R_0 = 1$. Hence

$$W = \frac{p_0 . A . f_A}{2}$$

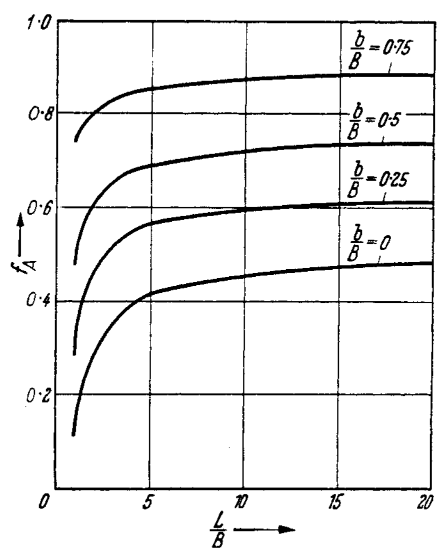


FIG. 397

This shows that the maximum stiffness is obtained if $p_0 = 2p_1$, i.e. if the supply pressure is twice the working pressure. If the value of h decreases by Δh under the additional working load P, the resistance R_1 and the ratio R_1/R_0 increase and, with them, the pressure p_1 and the load carrying capacity.

If the oil film thickness is very small, a deviation from the previous expression caused by hydrodynamic effects can be observed and the load carrying capacity increases rapidly (Fig. 398).[97] This means that even at very heavy loads, metallic contact between the two sliding surfaces is almost impossible as long as the oil pressure between them can be maintained.

The working conditions of machine tools require high stiffness and, therefore, as low values of Δh as possible. The value $\Delta h/h$ depends upon the oil film thickness h under the weight W and upon the permissible compression Δh of the oil film under the action of the operational forces. The value h must be so chosen that even when the oil film thickness is reduced to $h - \Delta h$, metallic contact does not occur in spite of manufacturing inaccuracies. If, for instance, $h - \Delta h$ shall not be less than 0·00024 in. and if Δh must not be greater than 0·00008 in., then $\Delta h/h$ must be less than 0·25.

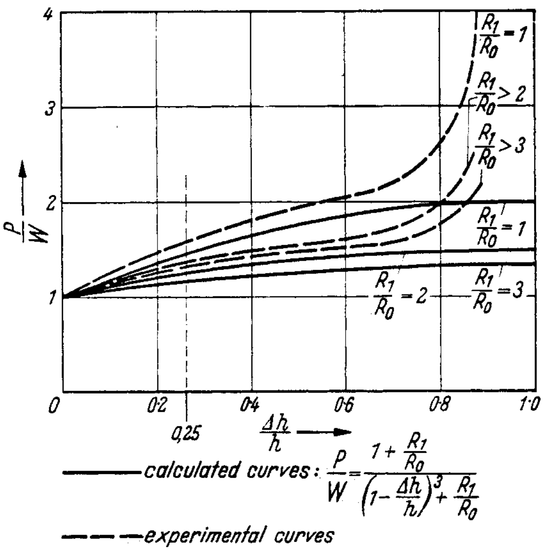


FIG. 398

The arrangement shown in Fig. 395 can, therefore, be applied only if P/W is not more than about 1·6, i.e. if the operational forces are not more than 160 per cent of the weight of the slide (see Fig. 398). If the operational forces are greater, and if pulsating forces occur, a hydrostatically lubricated holding strip (Fig. 399) is added, an arrangement in which pressure balancing occurs in such a manner that the oil film thickness remains almost constant. In accordance with the equation shown on page 260, the resulting vertical force acting downwards on the slideway is (Fig. 399):

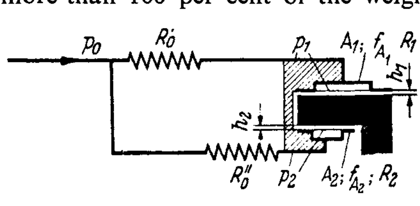


FIG. 399

$$P = \frac{p_0 . A_1 . f_{A_1} . R_1/R_0'}{(1 - \Delta h_1/h_1)^3 + R_1/R_0'} - \frac{p_0 . A_2 . f_{A_2} . R_2/R_0''}{(1 - \Delta h_2/h_2)^3 + R_2/R_0''}$$

If the oil film thickness is equal on both sides $h_1 = h_2 = h$, and because $\Delta h_1 = -\Delta h_2 = \pm\Delta h$ the following equation is obtained:

$$P = \frac{p_0 . A_1 . f_{A_1} . R_1/R_0'}{(1 - \Delta h/h)^3 + R_1/R_0'} - \frac{p_0 . A_2 . f_{A_2} . R_2/R_0''}{(1 + \Delta h/h)^3 + R_2/R_0''}$$

In order to obtain maximum stiffness, it may be assumed that:

$$\frac{R_1}{R_0'} = \frac{R_2}{R_0''} \approx 1$$

Hence:

$$P = \frac{p_0 . A_1 . f_{A_1}}{(1 - \Delta h/h)^3 + 1} - \frac{p_0 . A_2 . f_{A_2}}{(1 + \Delta h/h)^3 + 1}$$

Under the weight W of the slide $\Delta h = 0$, and:

$$W = \frac{p_0 . A_1 . f_{A_1}}{2} - \frac{p_0 . A_2 . f_{A_2}}{2}$$

$$= \frac{p_0}{2}(A_1 . f_{A_1} - A_2 . f_{A_2})$$

Hence:

$$\frac{P}{W} = \frac{2A_1 . f_{A_1}}{(A_1 . f_{A_1} - A_2 . f_{A_2})[(1 - \Delta h/h)^3 + 1]} - \frac{2A_2 . f_{A_2}}{(A_1 . f_{A_1} - A_2 . f_{A_2})[(1 + \Delta h/h)^3 + 1]}$$

$$= 2\left[\frac{1}{[1 - (A_2 f_{A_2}/A_1 f_{A_1})] . [(1 - \Delta h/h)^3 + 1]} - \frac{1}{[(A_1 f_{A_1}/A_2 f_{A_2}) - 1][(1 + \Delta h/h)^3 + 1]}\right]$$

With
$$\frac{A_1 \cdot f_{A_1}}{A_2 \cdot f_{A_2}} = \varphi$$

$$\frac{P}{W} = 2\left[\frac{1}{(1 - 1/\varphi)[(1 - \Delta h/h)^3 + 1]} - \frac{1}{(\varphi - 1)[(1 + \Delta h/h)^3 + 1]}\right]$$

This relationship is shown in Fig. 400, in which the variation of R_1/R_0' and R_2/R_0'' as a function of $\Delta h/h$ is not taken into consideration. This variation is small, however, within the permissible limits ($\Delta h/h < 0{\cdot}25$, see page 261).

It is clearly apparent from Fig. 400 that the arrangement of Fig. 399 can carry far heavier loads satisfactorily than the arrangement shown in Fig. 395.

The calculations previously shown referred to stiffness under static load. It can be shown, however, that the stiffness $P/\Delta h$ increases with the frequency of load pulsations and, therefore, as a general rule, it is possible to assume that any arrangement of a required static stiffness will be satisfactory under dynamic conditions (see Fig. 403c).

In Fig. 401,* the relationships between the load carrying capacity P, the stiffness $\Delta P/\Delta h$, the oil flow Q, the oil pressure p and the

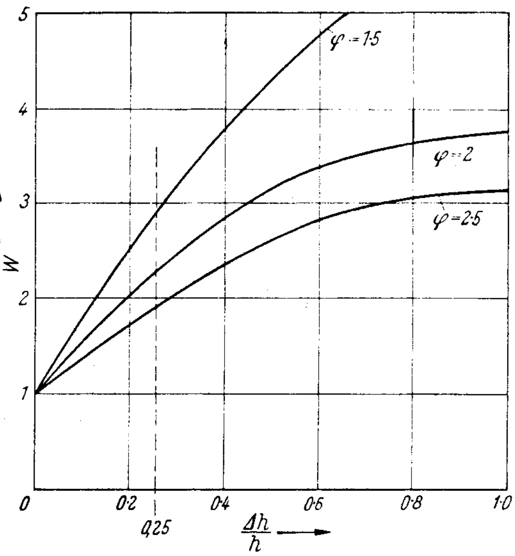


FIG. 400

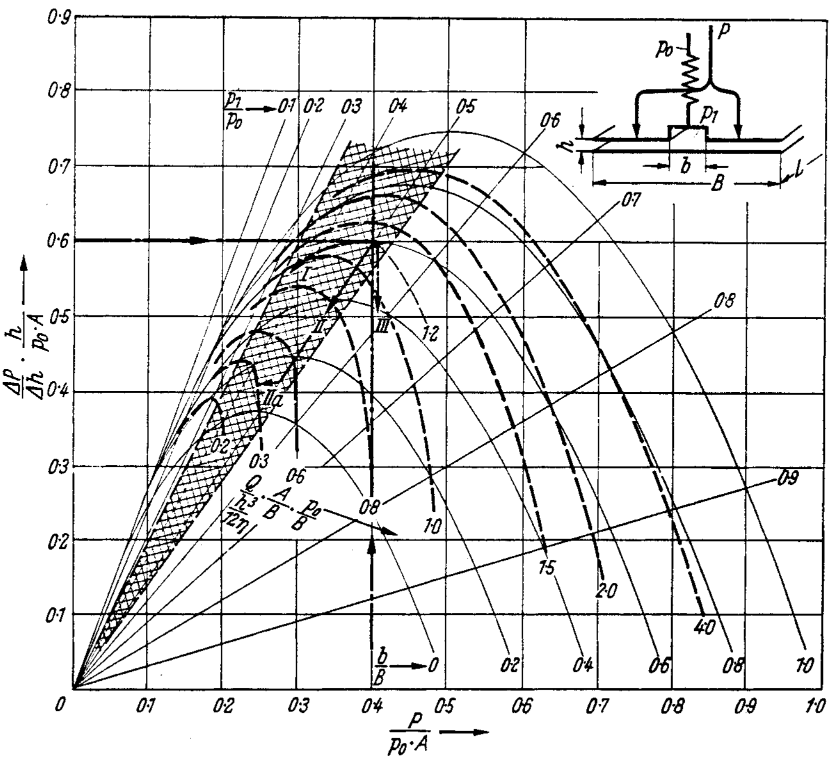


FIG. 401

* This Nomogram has been developed by Professor J. K. Royle.

dimensions of the slideway bearings (see sketch in the top right-hand corner of Fig. 401), are given non-dimensionally. The nomogram represents the following equations:

$$\frac{Q}{h^3/12\eta . A/B . p_0/B} = \frac{p_1/p_0}{1 - b/B}$$

$$\frac{(\Delta P/\Delta h)h}{p_0 . A} = \frac{3}{2}\left(1 + \frac{b}{B}\right)\frac{p_1}{p_0}\left(1 - \frac{p_1}{p_0}\right)$$

$$\frac{P}{p_0 . A} = \frac{1}{2}\left(1 + \frac{b}{B}\right)\frac{p_1}{p_0}$$

where η is the absolute viscosity of the oil.

An example may indicate the use of the nomogram. A rectangular slideway bearing (area $A = 12{\cdot}5$ in^2) is to carry a load of 2000 lb. The oil film is to have a static stiffness of 6,000,000 lb/in. The supply pressure of the oil is $p_0 = 400$ lb/in^2. In order to obtain optimum relations between the stiffness (vertical coordinate) and the load carrying capacity (horizontal coordinate), it is necessary to work near the maxima of the parabolas, i.e. in the cross-hatched zone of the nomogram. In this zone the ratio

$$\frac{\Delta P}{P} \bigg/ \frac{\Delta h}{h}$$

lies between 1·4 and 2·0. If a value of 1·5 is chosen, the following relationships exist:

$$\frac{\Delta P}{\Delta h} = 6{,}000{,}000 \text{ lb/in.}$$

$$P = 2000 \text{ lb}$$

$$\frac{6{,}000{,}000}{2{,}000} h = 1{\cdot}5$$

$$h = 0{\cdot}0005 \text{ in.}$$

which appears reasonable.

$$A . p_0 = 12{\cdot}5 \times 400 = 5000 \text{ lb}$$

Hence

$$\frac{P}{A . p_0} = \frac{2000}{5000} = 0{\cdot}4$$

$$\frac{\Delta P . h}{\Delta h . p_0 . A} = \frac{6{,}000{,}000 \times 0{\cdot}0005}{400 \times 12{\cdot}5} = 0{\cdot}6$$

For these values a width ratio $b/B = 0{\cdot}6$ and a pressure ratio $p_1/p_0 = 0{\cdot}5$ are shown in the nomogram. The value of 1·2 representing the oil flow may appear a little high. It can be reduced by:

(a) following the $b/B = 0{\cdot}6$ line to the left (arrow *I*); or

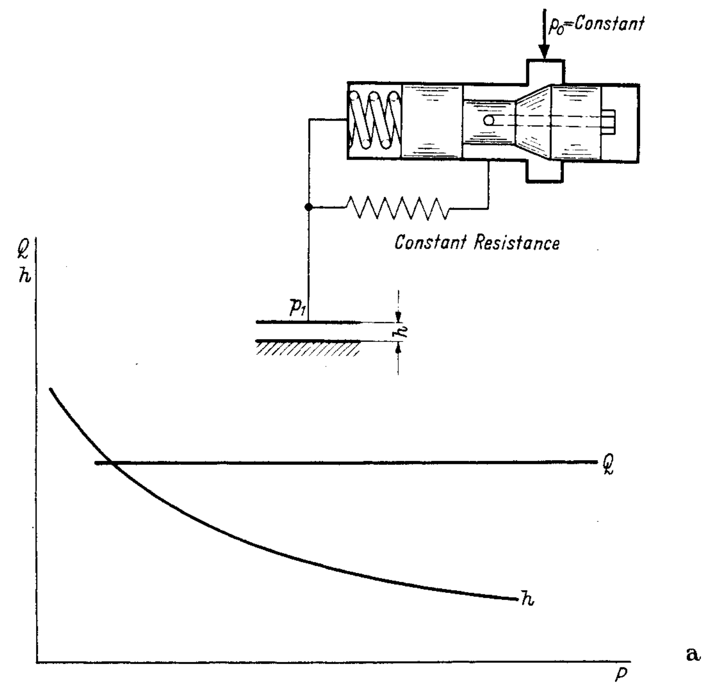
(b) following the $p_1/p_0 = 0{\cdot}5$ line to the bottom left (arrow *II*); or

(c) moving vertically downwards (arrow *III*).

The third alternative (*c*) would reduce the stiffness below the permissible value. The alternative (*a*) would not result in a considerable reduction of the oil flow. However, by using alternative (*b*), it is possible to reduce the oil flow considerably, as long as the bearing area can be increased sufficiently to maintain the required load carrying capacity without affecting the stiffness. It may even be permissible to allow a slight reduction of the load carrying capacity (arrow II_a) without affecting the stiffness ($\Delta P/\Delta h . h/p_0 A = 0{\cdot}45$), and this would further reduce the oil flow.

In practice, a compromise will perhaps have to be found.

It is also possible to devise some hydraulic servomechanism,[98] in which (see also page 157) the pressure in the oil film is used for operating valves which in turn adjust the resistances to suit the prevailing conditions.



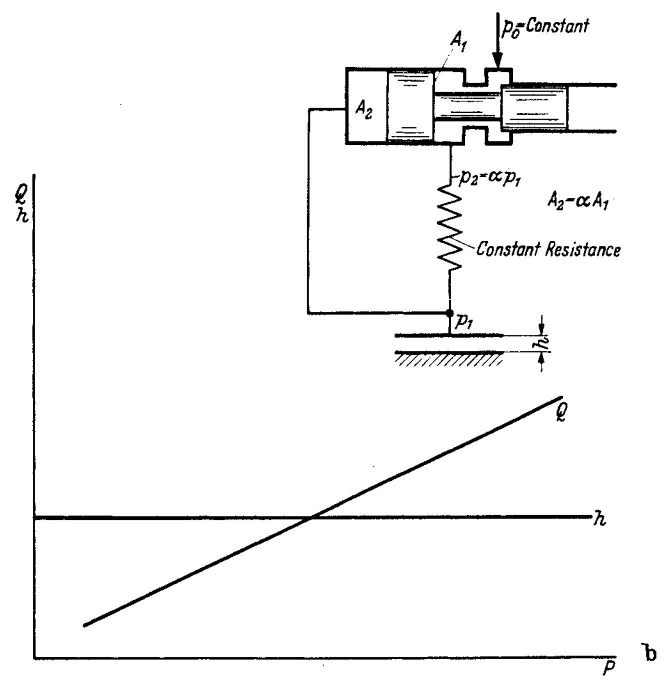
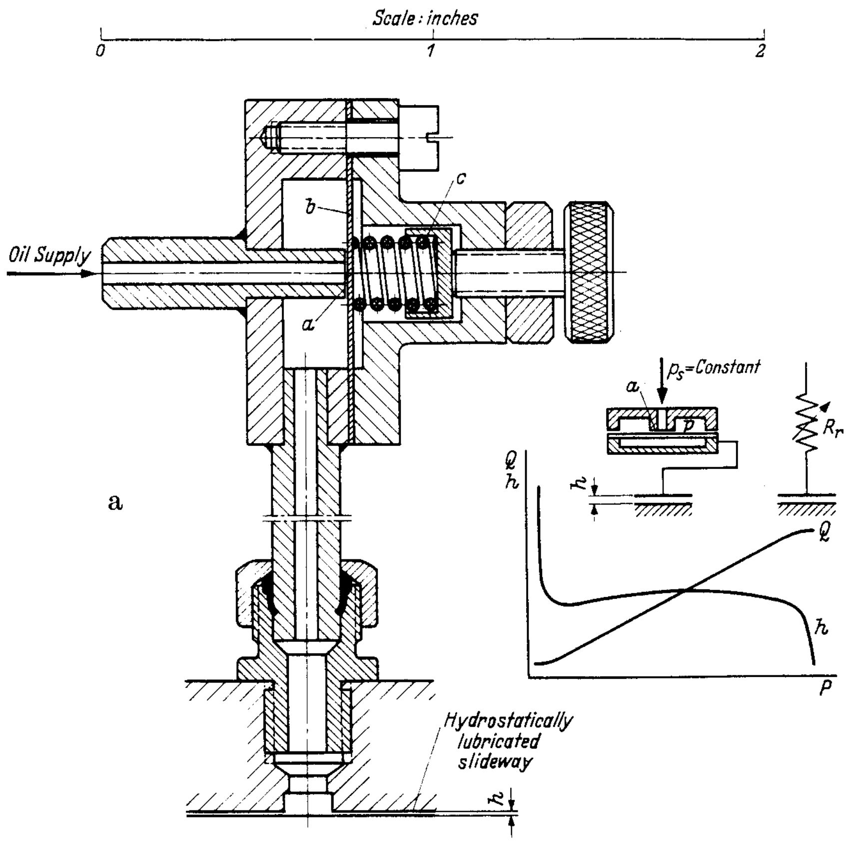


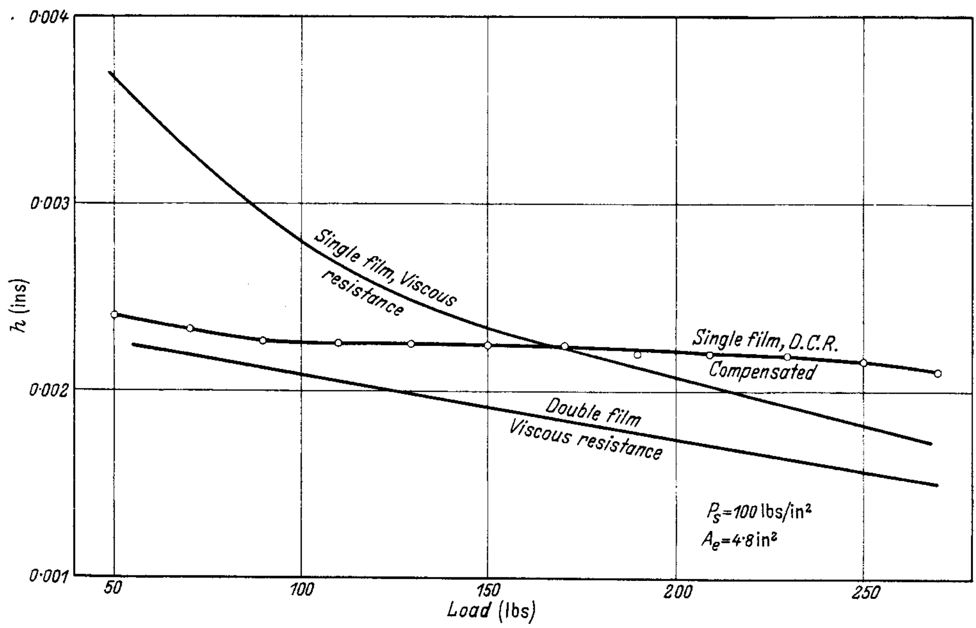
FIG. 402

In the constant flow method (Fig. 402a) a spool valve is controlled by a spring which ensures a constant pressure drop across a fixed resistance. In a method* which has been patented (Fig. 402b)

*J. K. ROYLE, British Patent.



a



b

FIG. 403a and b.

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),

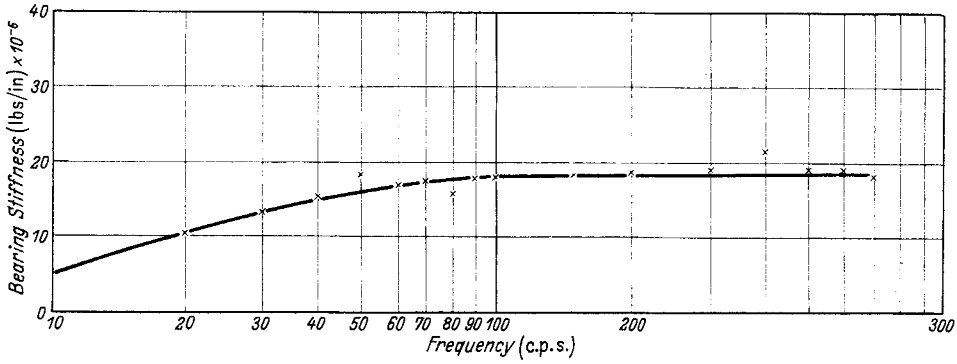


FIG. 403c.

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

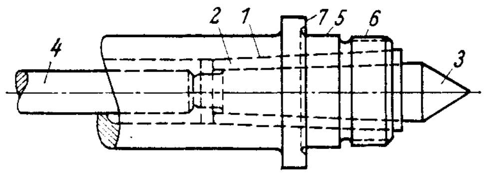


FIG. 404

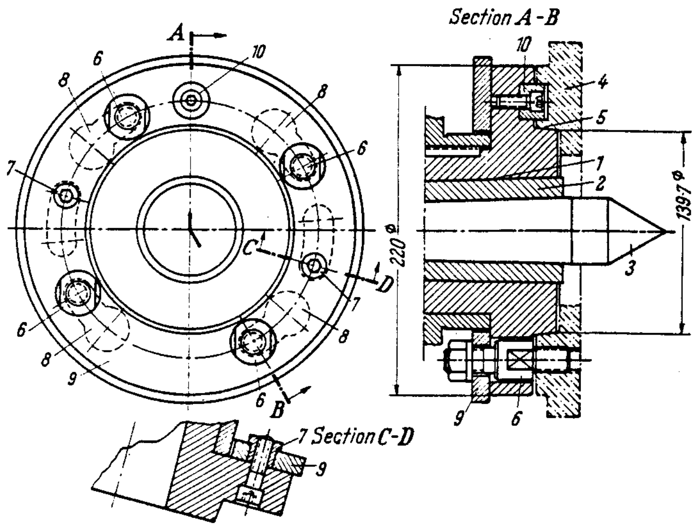


FIG. 405a. Spindle nose according to DIN 55022 with driving plate (Schaerer lathe).

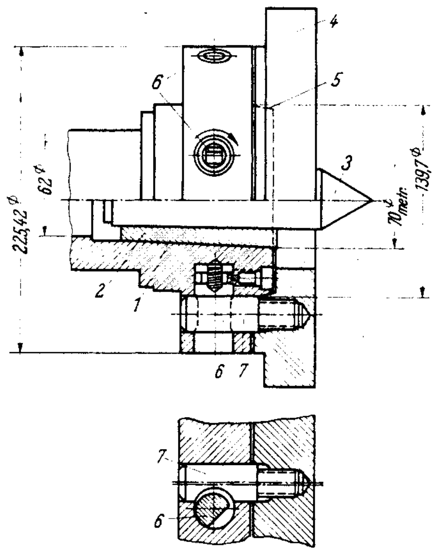


FIG. 405b. Spindle nose with short taper and "camlock" for the driving flange (Schaerer lathe).

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

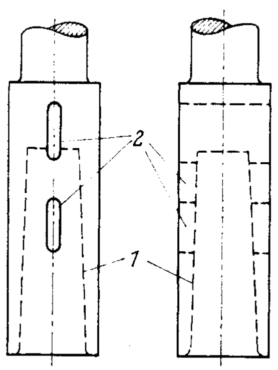


FIG. 406

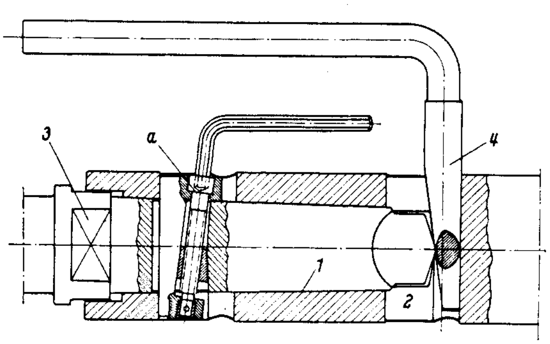


FIG. 407. Patented spindle nose of a horizontal boring machine (Collet & Engelhard, Offenbach, Germany).

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

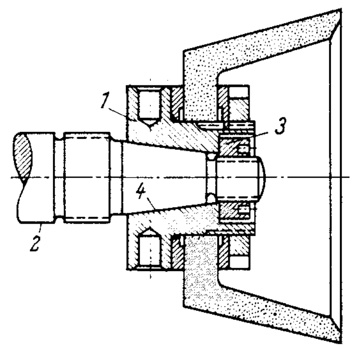


FIG. 408

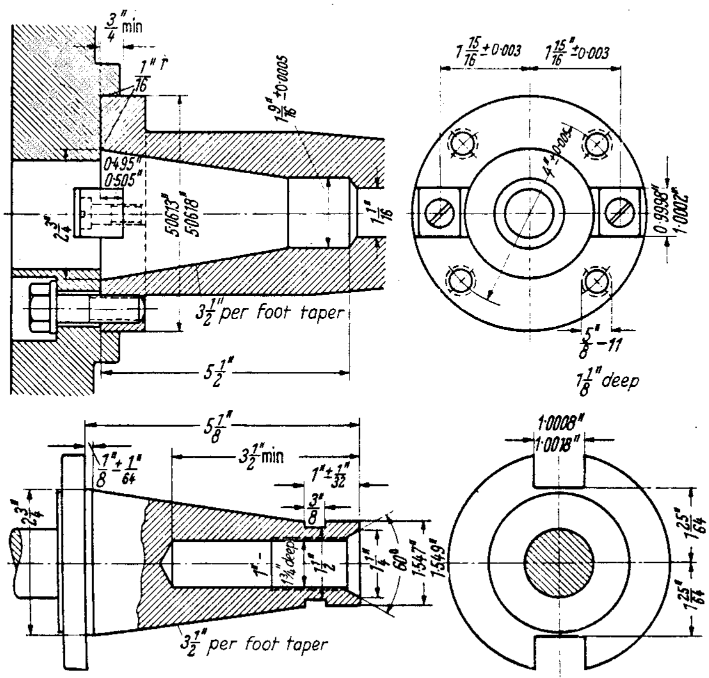


FIG. 409. Milling spindle taper.

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*.

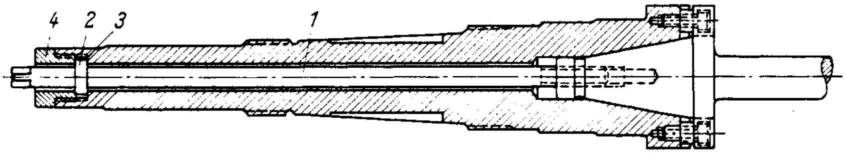


FIG. 410. Milling spindle.

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),[100] developed by the Cincinnati Milling Machine Company, and consisting mainly of a

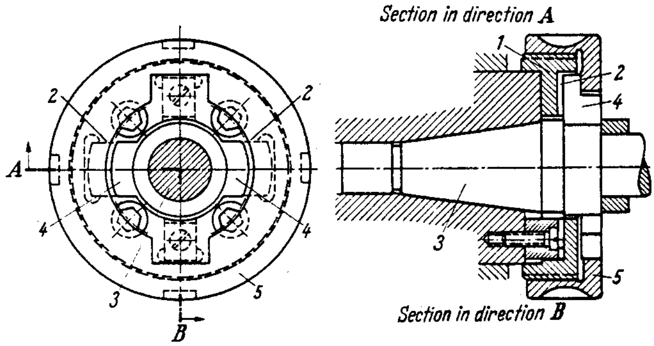


FIG. 411. Milling spindle nose. *1*—Driver with machined slots for *2*; *3* arbor with flange and driving tongue *4* machined from solid; *5*—Clamping nut.

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]

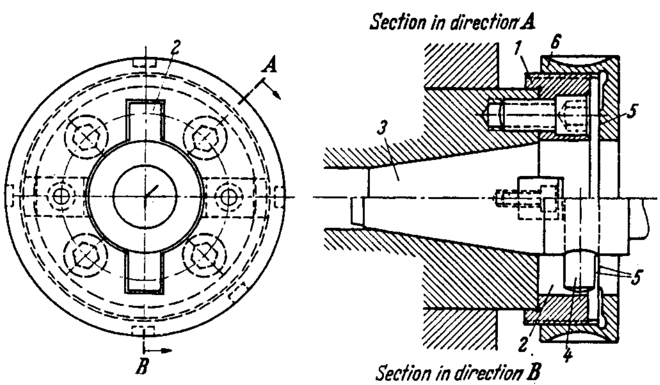
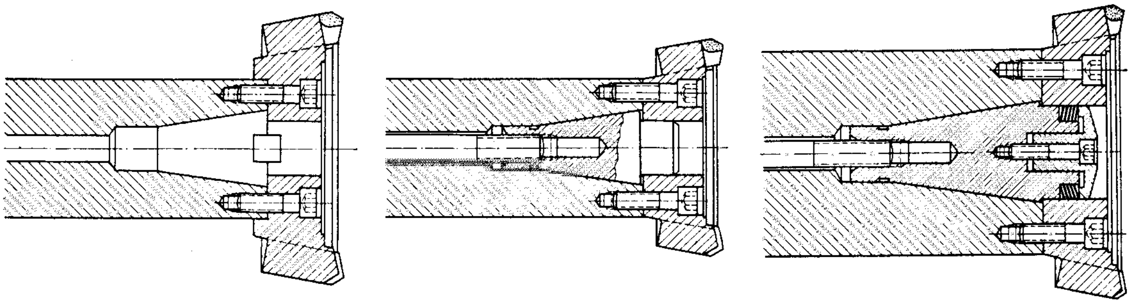


FIG. 412. Milling spindle nose. *1*—Driver with machined slots *2*; *3*—Arbor with driving peg *4*; *5*—Clamping faces; *6*—Clamping nut.

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 when the centring mandrel is equipped with an expanding element (Fig. 415). In order to prevent inadmissible shifting of a cutter head due, e.g. to unequal tightening of the clamping screws, spherical washers (Fig. 416) are used in cases where the highest accuracy is essential.



FIGS. 413, 414 and 415. ISO milling spindle heads.

The operational movement of the spindle may be either purely rotational (lathe, grinding machine), or rotational and axially sliding (drilling machines, boring machines, some milling machines). Excessive stresses, deformations and vibrations are prevented by strength and stiffness, balancing, and suitable layout and design of the driving elements and bearings.

(i) *Strength and stiffness*: The deformation of the spindle under load has been previously discussed (see page 51). A short, stout design results in high stiffness against bending and torsion and, if necessary, against buckling under the action of axial forces. The spindle diameter is, of course, limited by the dimensions of the driving elements and bearings. Provisions for relieving the spindle of the action of belt tension or tooth pressure by independent bearings for the driving elements (see Figs. 142 and 425) reduce the magnitude of transverse forces acting on the spindle.

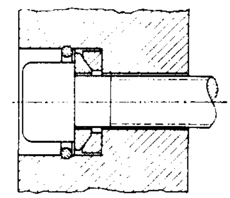


FIG. 416. Clamping screw with spherical washer for fastening cutter heads.

The displacement of the spindle caused by bending deformations may result in skewing in the bearings. This means that lack of stiffness may indirectly become the cause of additional pulsating transverse forces and thus lead to vibration trouble.

The main spindle is one of the most expensive parts of the machine tool and for this reason, full use should be made of materials which combine high strength, good running properties and long life. As the spindles usually have to be heat-treated, difficulties may arise, e.g. when long hollow cylinders are hardened. The boring operation is, therefore, often carried out after the heat-treatment. For this purpose the bearing faces are usually pre-ground so that they can later serve as datum faces for the boring operations. The case hardening steels normally employed enable a hard surface with a relatively soft and easily machinable core to be produced.

(ii) *Balancing*: In unbalanced high speed spindles centrifugal forces may lead to deformations and rough running. It is, therefore, necessary to avoid or eliminate any unmachined and irregularly shaped cast components, unilateral keys, oil holes, etc. Play between the spindle and the torque transmitting elements (gear wheels, chain sprockets, pulleys) may contribute to irregular torque transmission, high stressing of the keys and rough running. To prevent this, the driving elements are often shrunk on to high performance spindles or pressed on to tapered location faces if assembly or dismantling have to be facilitated. Assembly and dismantling of the spindle can also be made easier if the diameters of the various portions are suitably stepped. In this connexion, it is essential to watch the possibility of pressing all keys into their keyways before assembly of the spindle unit.

(iii) *Layout and design*: One of the most important aspects from the point of view of satisfactory performance is the general layout and design of the driving elements and the spindle bearings.

(a) *Layout*: Unless taper roller or angular contact ball bearings are used the axial and transverse forces can be supported separately. In the layout of bearings designed for this purpose, a certain compromise cannot be avoided because both types of bearings, the radial journal from the point of view of bending deflexions and the axial thrust bearing from the point of view of buckling, should be placed as near as possible to the point of working load application, i.e. the spindle nose. In general, therefore, other design considerations may decide whether the axial thrust bearings should be near the spindle nose, at the spindle end, or on either side of the main journal bearing. There is, however, little doubt that it is most important for the axial thrust bearing to be as near as possible to the front (main) journal bearing. The danger of buckling in general is not as important as the possible longitudinal expansion of the spindle which must not be directed towards the tool, but away from the spindle nose towards the rear.

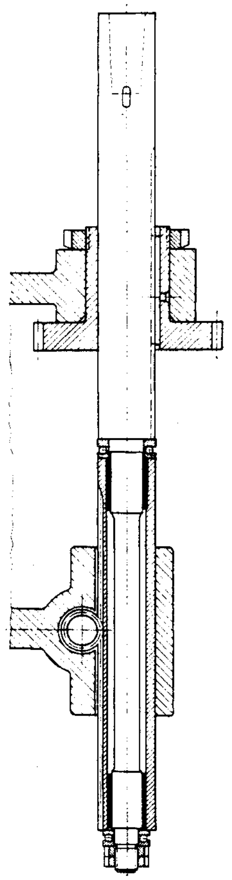


FIG. 417

The part of the spindle length which lies between the drive and the point of application of the cutting force is subjected to bending and torsion. For stiffness reasons, it must be kept as short as possible. The main driving wheel should, therefore, be close behind the main journal bearing. This requirement cannot be satisfied if the spindle is axially displaceable, because the driving element is usually in a fixed axial position, and contains a splined bore through which the spindle has to slide. In such a case, it is essential to have independent bearings which support the driving elements. The axial traverse of a milling spindle is relatively small, but contradictory requirements are often encountered in the case of drilling machine spindles. If, for instance, the torque transmission for the spindle drive is placed near the main bearing in order to keep torsional deformations small (Fig. 417), the axial drive for the feed movement has to be placed in a relatively high position. Apart from the increased buckling stresses in the spindle and the need for a tight sliding fit in the bore of the driving wheel, relatively long levers or shafts are necessary for the mechanisms which connect the feed drive with the position of the operator. For this reason drilling machine spindles are usually designed of sufficient strength to keep torsional deformations within permissible limits, even if the spindle drive is further removed from the main journal bearing, and the torque transmitted at the top. This makes it possible to arrange the feed drive at the lower end of the spindle (Fig. 418) and nearer the operator. The sliding spindle sleeve *a* carries the journals *b* and *c* and the thrust bearings *d* and *e*. The feed drive is provided by pinion *f* and rack *g* which is machined into the spindle sleeve *a*. The rotational movement of the spindle is driven by the independently supported driving sleeve *h* and splines *i* (feed drive small gear *k*, cutting drive worm wheel *l*). A balance weight acting at *m* prevents dropping of the spindle when the drill breaks through.

The question of accessibility for the feed drive is less critical with horizontal boring machines, because the nose and tail of the spindle are at the same height. Frequently, however, the spindle drive must transmit considerable power, because not only drilling but also milling operations are carried out on these machines. The spindle drive (keys *a*, Fig. 419), is, therefore, arranged as close as possible to the spindle nose whilst the feed drive (feed screw *b*, nut *c*, sliding bracket *d*) is applied at the tail. An accurately concentric combination of the bearings for the spindle sleeve *e* and the face plate carrier *f* is essential and thus requires great stiffness and high precision (see page 52), because the superposition of the bearing inaccuracies and possible play may result in excessive running errors. The fitting of such bearings can be facilitated by the provision of separate bearing bushes *g* (see also Figs. 339 and 425).

The drive of the face plate *h* is introduced at the most favourable point, i.e. at the maximum possible diameter, through pinion *i* and internal gear wheel *k*. It is also possible to couple the face plate carrier *f* with the spindle sleeve *e* by means of the gear *l* and the internally geared part *m* of the sliding gear *n*.

In spite of the manufacturing difficulties spindles are often supported at more than two points in order to increase their stiffness. An investigation of the relative stiffness obtained with spindles in

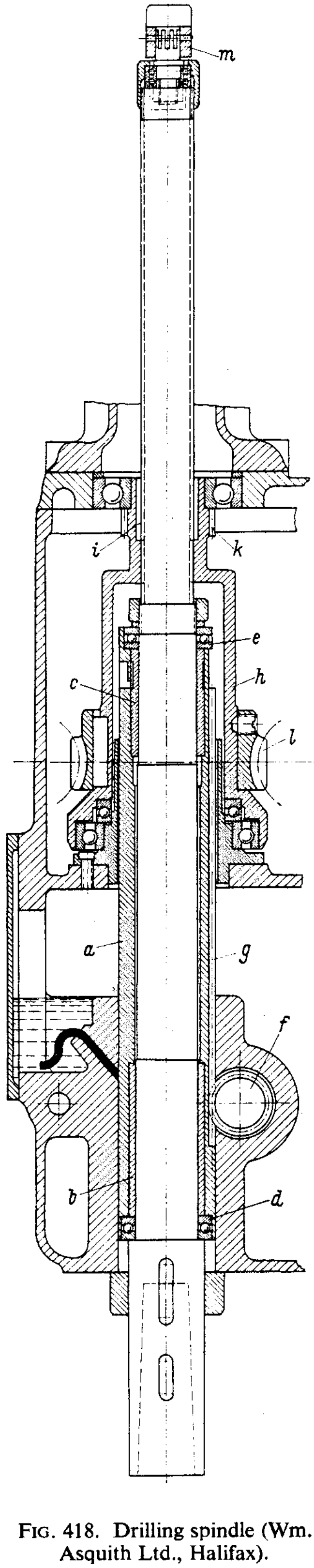


FIG. 418. Drilling spindle (Wm. Asquith Ltd., Halifax).

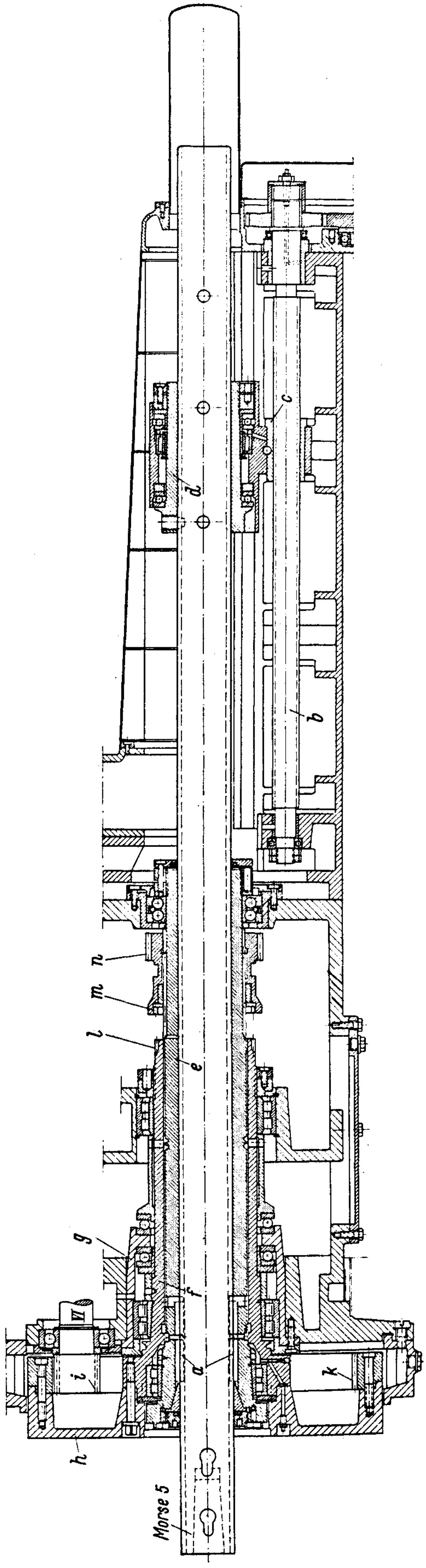


FIG. 419. Spindle of a horizontal boring machine (Collet & Engelhard, Offenbach, Germany).

two (statically determinate) and three (statically indeterminate) bearings[101] confirmed the predictable increase in stiffness obtained with three bearings, because the conditions were then nearer to those of a clamped support as compared with those of a free support encountered with two bearings (see page 52). However, misalignment and other errors are possible in multiple bearing arrangements, especially if the bores cannot be produced in one setting, and this may result in skewing, jamming and vibrating. It would, therefore, appear preferable not to correct the insufficient stiffness of a spindle design by means of "back bending moments" exerted by additional bearings, especially in view of the fact that in general engineering, bearing designers endeavour to prevent such back bending effects by making use of self-aligning bearings.

(b) *Design.*[102] The design of the bearings and their elements, especially for the spindles of multi-purpose machines, is made difficult because these spindles are likely to be subjected to considerably varying loads and their speeds cover a very wide range. With low spindle speeds it is difficult to obtain hydrodynamic lubrication conditions in plain bearings. On the other hand, the centrifugal forces which arise at high spindle speeds may give rise to difficulties if ball or roller bearings are used for large diameters. Ball and roller bearings have not only the advantage of simple lubrication requirements, even at high speeds, but they also provide the possibility of obtaining almost complete elimination of play. However, the damping capacity of ball and roller bearings is less and their sensitivity to impact loads greater than those of plain bearings. In addition, it must be remembered that commercial ball and roller bearings are usually unsuitable for spindle supports, so that expensive special bearings must often be applied.

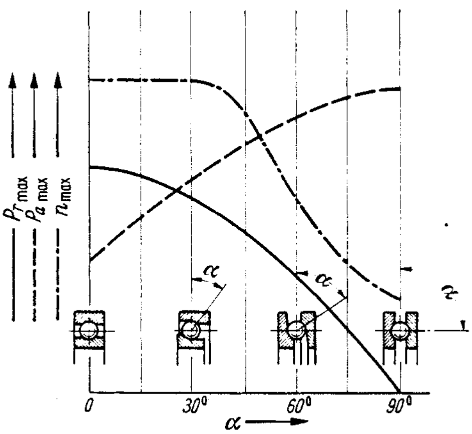


FIG. 420. Load carrying capacity of ball bearings. $P_{r_{max}}$—Radial; $P_{a_{max}}$— Axial; n_{max}— Max. speed (rev/min).

In order to obtain the required working accuracy, spindle bearings must be adjustable, so that the unavoidable manufacturing inaccuracies of the bores and the corresponding cylindrical parts can be compensated.

(1) *Ball and Roller Bearings*[102]

The axial and radial running accuracy of these bearings depends upon their design and manufacture and is influenced by the shape inaccuracies of races, rollers and balls, the radial play, the dimensional differences of the various elements and by the radial and axial supports in the housings. Quite generally, it is preferable to separate the support of radial and axial forces, because in this manner optimum conditions for both cases are obtainable (Fig. 420).[103] On the other hand, the design can often be simplified considerably by the application of taper roller bearings (Fig. 421), which have a relatively high load carrying capacity under both axial and radial forces. Ball bearings are usually the cheaper and are less sensitive to small alignment errors. However, roller bearings have a greater load carrying capacity. The danger of excessive pressure at the points of contact between the roller edges and the races can be reduced by the use of slightly barrel shaped rollers. The radial play in taper roller bearings can be adjusted fairly easily by axial displacement of the two races (Fig. 421). It is, however, difficult with taper roller bearings to obtain the required running accuracy, as this depends upon the manufactured accuracy with which the axes of several tapers (the outer race, the inner race and the rollers), the axis of the spindle and that of the bearing housing coincide or intersect at one point.

In the design (Fig. 421), the stiffening effect of the third bearing (see page 271) is used to increase the stiffness of the design. If, however, the stiffness of the spindle is so low that the "back bending effect" of the centre bearing comes into full effect, there is a danger that the axes of the rollers and the races will be displaced relative to each other over and above their manufacturing inaccuracy.

Moreover, if the two taper roller bearings are too far apart from each other, the bearing play as originally adjusted, increases with any rise in temperature and the resulting expansion of the spindle.

In the design of a lathe spindle (Fig. 422), these dangers are avoided by using a double taper roller bearing at the spindle front. The play of these high precision taper roller bearings is adjusted by a ring nut a at the spindle front, and by springs b acting on the single taper roller bearing at the spindle end.

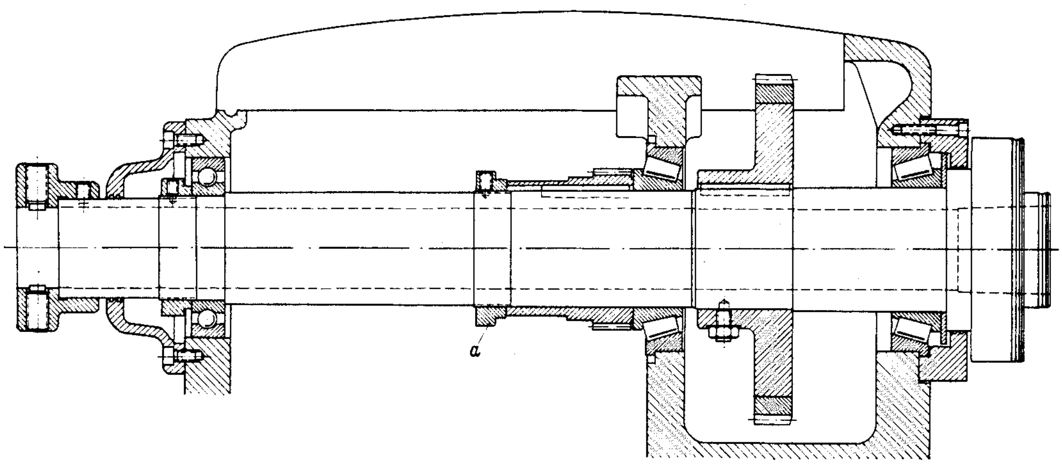


FIG. 421. Lathe spindle bearings (Dean, Smith & Grace Ltd., Keighley). a—Adjusting nut for the taper roller bearings.

When considering the lubrication of taper roller bearings it is important to ensure that the pumping action of the rollers, arising from their axial inclinations, does not oppose the direction of the oil supply. It is best to feed the oil in at the smaller diameters d_1 (via pipes c), so that it is pumped by the taper rollers towards the larger diameters d_2 and from there brought back to the tank (via bores e).

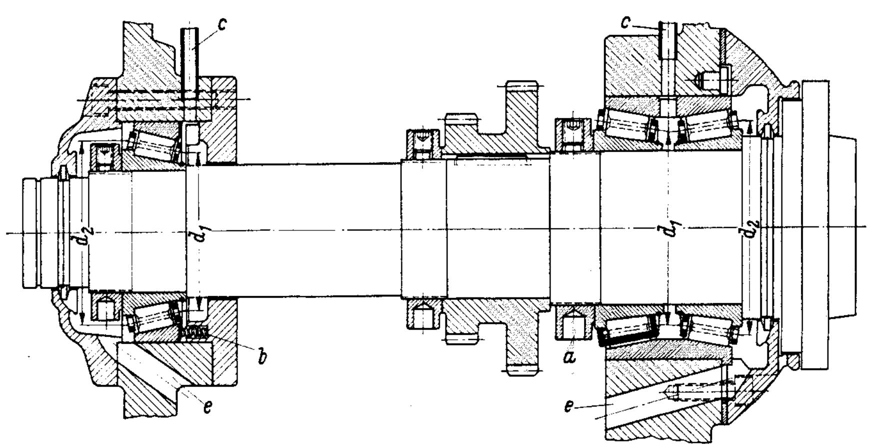


FIG. 422. "GAMET" bearing arrangement (La Précision Industrielle, Rueil-Malmaison, France).

The cylindrical roller bearing carries radial loads independently of the axial loads. It requires, however, special provision for play adjustment and, if necessary, pre-loading. The spindle of a turret lathe (Fig. 423) is supported by two double roller bearings, one at the nose and one at the opposite end. The outer races are located axially in the housing. The bores of the inner races are tapered and can be displaced axially on the spindle by nuts a and b at the front and nuts c and d at the other end. Fine adjustment, which requires a sensitive touch, could be impaired by the static friction between the bore of the inner race and the taper on the spindle. This friction must be overcome before adjustment is started, i.e. before the inner race begins to slide axially on the spindle. For this reason, oil can be inserted through the small bores *e* by means of an injector, and an oil film

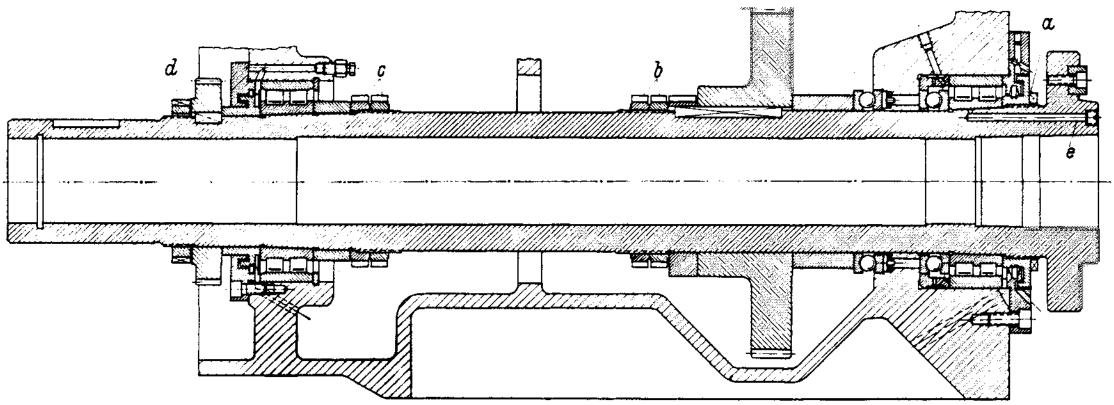


FIG. 423. Main spindle of a turret lathe (Gebr. Heinemann A.G., St. Georgen, Germany).

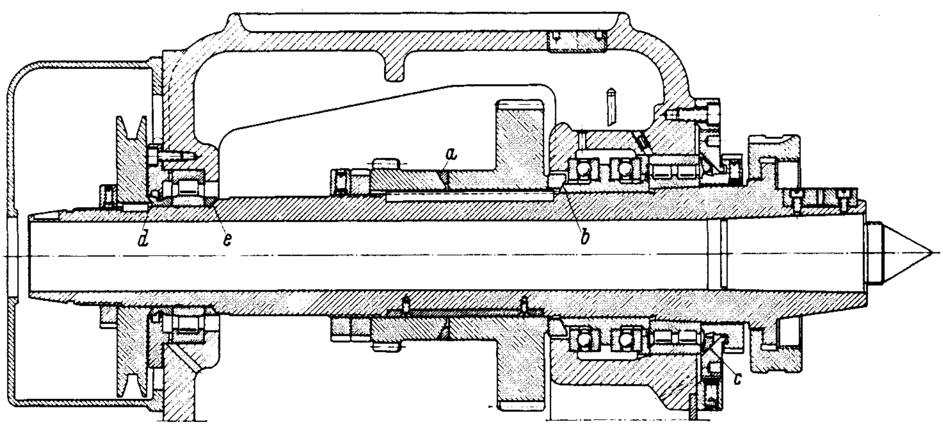


FIG. 424a. Main spindle of a copying lathe (Georg Fischer A.G., Schaffhausen, Switzerland).

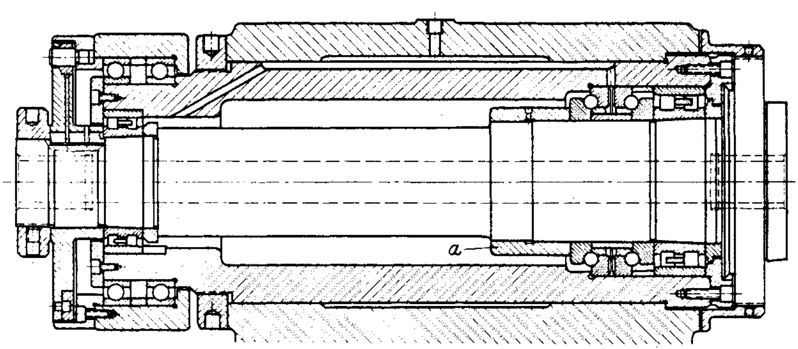


FIG. 424b. Work spindle incorporating SKF double row cylindrical roller bearings type NN 30k and angular contact ball thrust bearings.

formed between the sliding surfaces, so that stick-slip phenomena and the consequent difficulties in fine adjustment are avoided. Once adjustment is completed, the oil film must, of course, be removed so that the inner race is pressed tightly on to the spindle surface over its full length. Provision does not appear to have been made for this in the design of the lathe spindle (Fig. 424a) which is, however, of particular interest because of its compact arrangement and the possibility of mounting both the roller bearings and the two thrust bearings on the spindle before the latter is introduced into the bore of the headstock casting. All thrust faces, which serve for axial adjustment or for the support of axial loads (*a*, *b*, *c*, *d*, *e*), are lapped into a spherical shape and are, therefore, self-aligning. This ensures that the axial forces exerted on to the various parts of the spindle are uniformly distributed over the circumferences and do not produce bending moments on the spindle, even if the faces of rings, gears, etc., are not exactly at right angles to the spindle axis. Another means of accurate aligning is the complete avoidance of screw threads, shrunk bushes *a* replacing the adjusting nuts (Fig. 424b).[104]

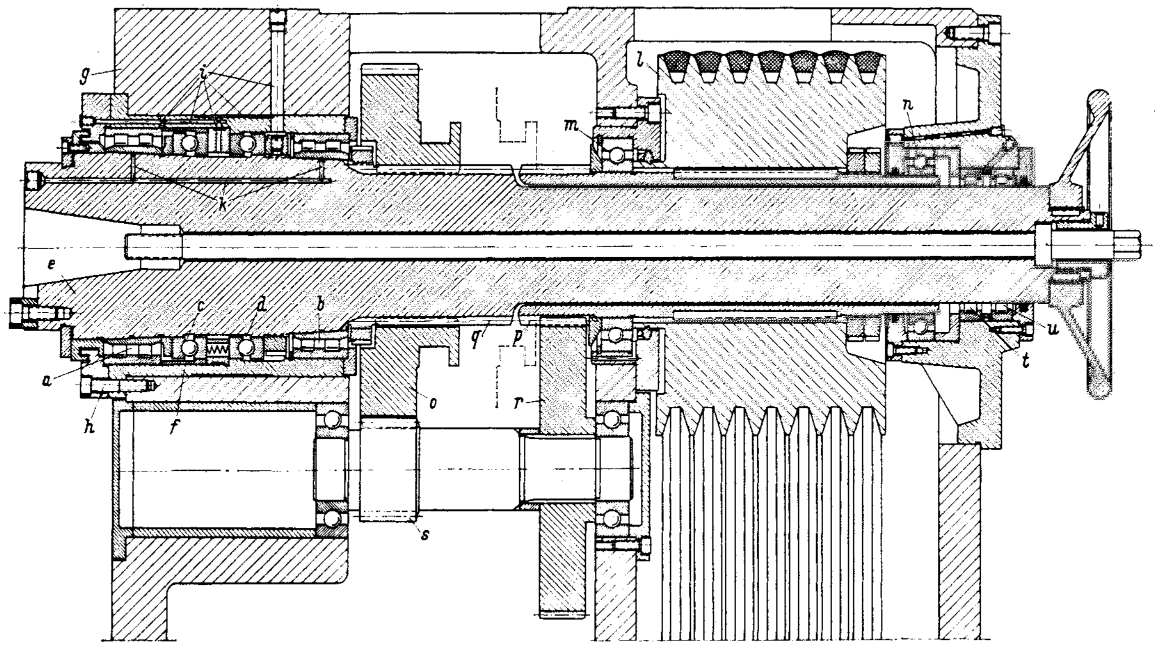


FIG. 425. Milling machine spindle (Gebr. Heller G.m.b.H., Nürtingen, Germany).

The milling machine spindle (Fig. 425) combines high stiffness with ease of assembly. The whole set of bearings at the spindle front, two double roller bearings *a*, *b* and two thrust bearings *c*, *d* can be mounted on the spindle *e* and in the bush *f* before the whole unit is inserted into the spindle head *g* and fastened to it by screws *h*. Oil supply bores *i* are provided for the lubrication of the ball and roller bearings and *k* for adjusting and preloading the inner races (see page 275). The Vee-belt pulley *l* is supported by two independent ball bearings *m*, *n*, so that the spindle is relieved of the belt tension. The driving torque is transmitted over the full circumference of the spindle by means of the internal teeth in the sliding gear *o*, which connects the gear teeth *p* of the pulley with those (*q*) of the spindle, unless the sliding gear is moved to the left and the drive goes via the back gear (*p*–*r*–*s*–*o*). The bearing at the spindle end is loaded only slightly and is of little importance for the quiet running of the spindle. For this reason, the cylindrical rollers *t*, *u* run directly on the spindle surface.

The bearings for the rotating tables of vertical borers present special problems,[105] because not only the radial forces but also the axial ones may act on relatively large diameters and exert considerable tilting moments. In order to keep the bearing loads, which are produced by such tilting moments, as low as possible, either very long centre bearing arrangements or large diameter axial bearings are employed. If taper roller bearings are used, a long centre bearing arrangement is unsuitable because of the change in play which occurs when the spindle temperature rises (see page 274). On the other hand, the use of large diameters for radial bearings requires increased tolerances for the shafts and bores and these have an unfavourable effect upon the running accuracy of the table. As a result the separation of radial and axial bearings (Fig. 426)[106] is again advantageous. A design of particularly low height for large diameter tables is shown in Fig. 427.[107]

By calculating the elasticity of the bearings and the table, it is possible to determine the stiffness of the arrangement as a whole under different conditions of load application.

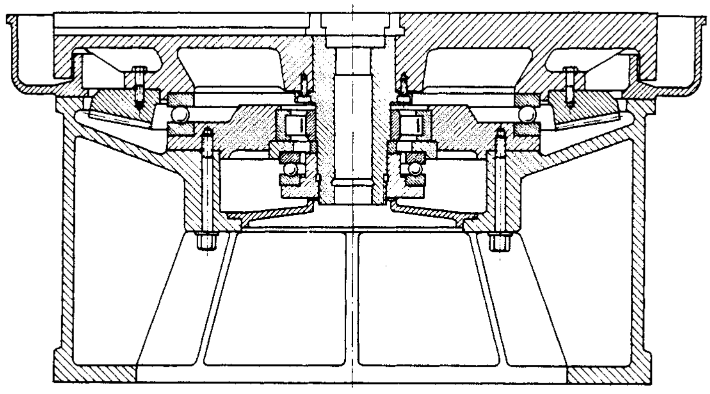


FIG. 426. Patented rigid bearing arrangement for a vertical boring machine table.

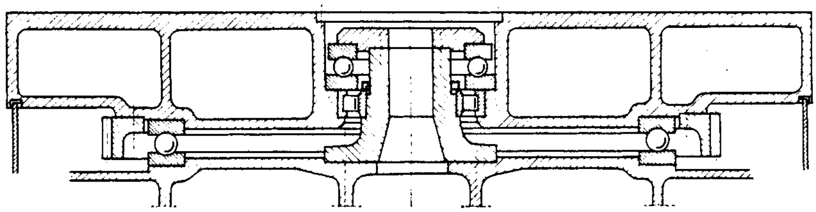


FIG. 427. "Low" design of a bearing arrangement for a vertical boring machine table.

In the application of ball and roller bearings, it is often considered a disadvantage that their external diameters are relatively large and require considerable space. This disadvantage is reduced by using needle roller bearings. As the diameters of the rollers are small, it is possible not only to reduce the external diameter of the outer race, but also to increase the number of rollers. This results in turn in a lower load per roller and permits a reduction in the wall thicknesses of the races.

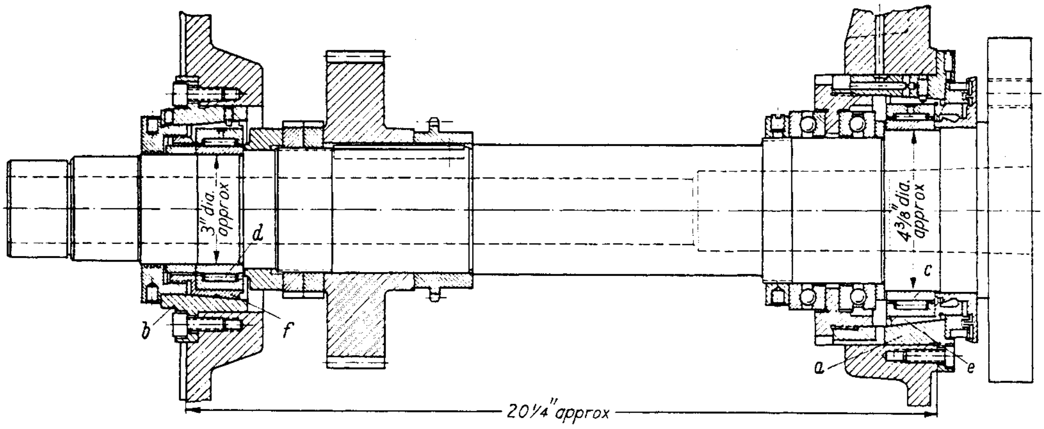


FIG. 428. Main spindle of the "Frontor" lathe (J. G. Weisser Söhne, St. Georgen, Germany).

Figure 428[108] shows a lathe spindle which is supported in needle roller bearings. It is again interesting to note the design of the bearing unit at the spindle front (needle roller bearing and two thrust bearings) and of the end bearings, which are each mounted in a bush and inserted as units 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.

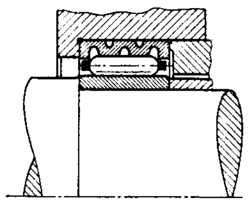


FIG. 429

(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

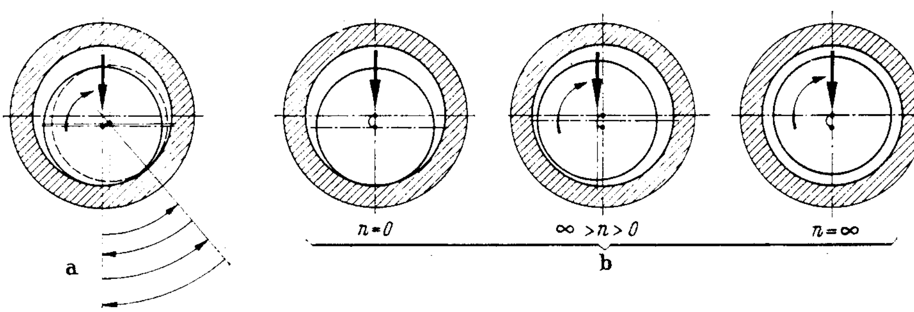


FIG. 430

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.

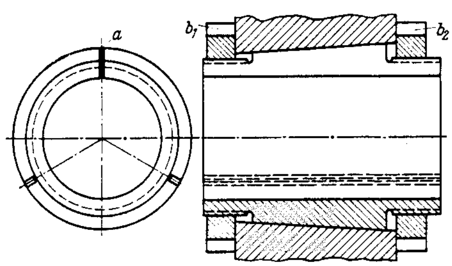


FIG. 431

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

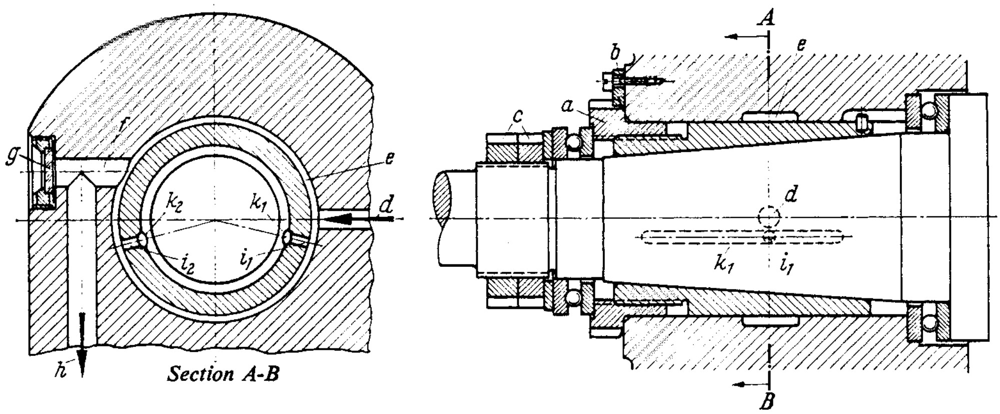


FIG. 432

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

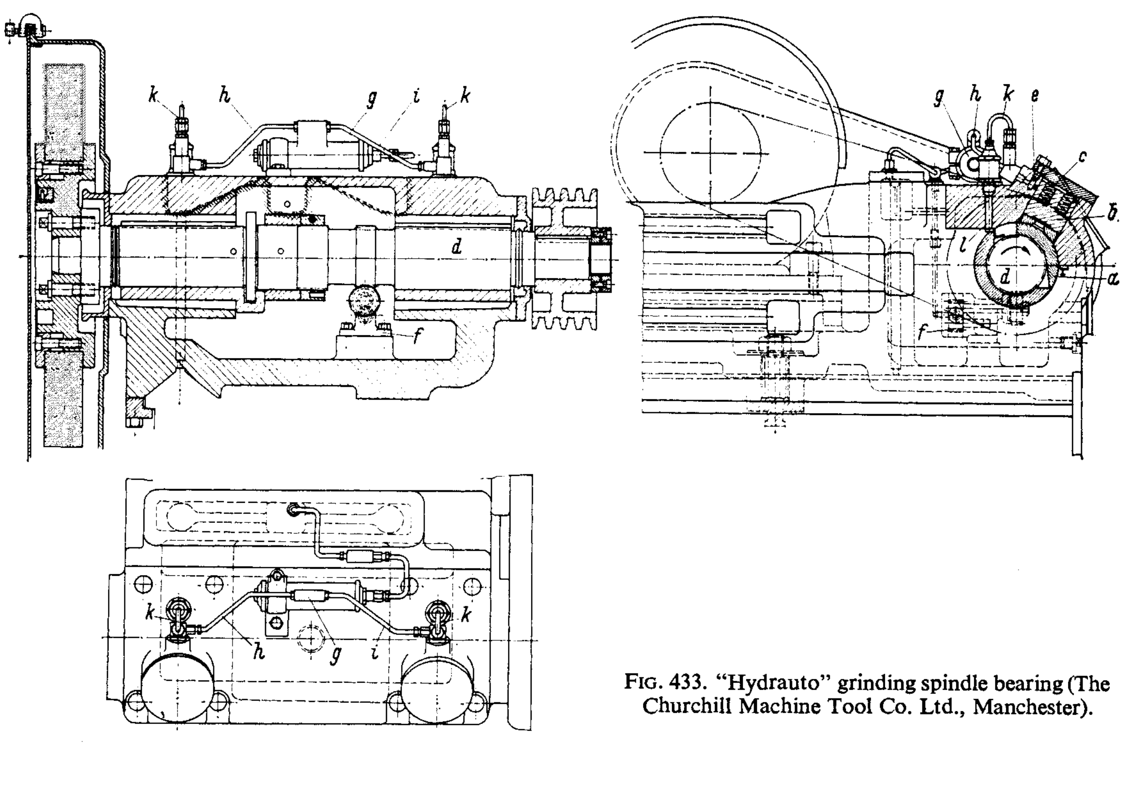


FIG. 433. "Hydrauto" grinding spindle bearing (The Churchill Machine Tool Co. Ltd., Manchester).

bores i_1 and i_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

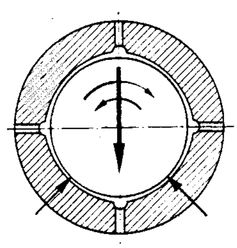


FIG. 434

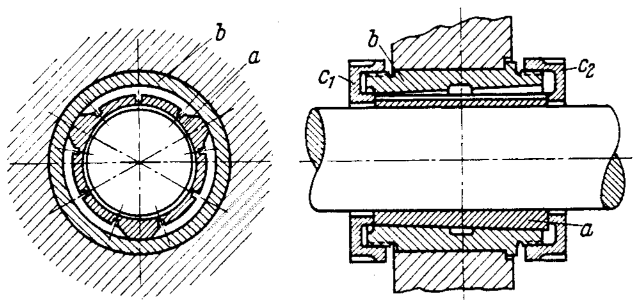


FIG. 435. Mackensen bearing.

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.

0·00005 in., the maximum total play being approximately 0·0002 in. The use of three bearing surfaces results in a statically determinate arrangement (Fig. 436).

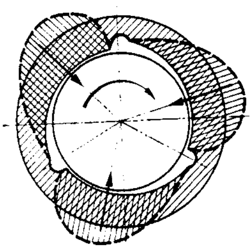


FIG. 436

The bearing designs shown previously are very rigid, and if the stiffness of the spindle is so low that its deformations under load are greater than the radial play in the bearing, excessive edge pressures may arise, even if the lubrication conditions are most favourable. This can be avoided by the use of self-aligning bearings in which the supporting pads or bushes can swivel on pegs or in spherical seatings and thus adjust their position to suit the elastic line of the spindle axis. Such bearings do not, however, exert "back bending moments"! It must be stressed once again that it is better to design a spindle of the required bending stiffness, rather than to compensate for too low a bending stiffness by clamping moments in the bearing. A design for self-aligning axial thrust bearings was suggested by Michell over 50 years ago. A design similar to that of Michell is used in the "Filmatic" Bearing (Fig. 437).[113] Five tilting bearing segments a are arranged around the spindle and the whole bearing space is kept filled by slightly pressurized oil (oil pipe b). The spindle rotation causes the oil to lift the leading edges a_1 of the segments, so that these become slightly tilted and hold the spindle tightly with their trailing edges a_2. This results in a type of hydrodynamic clamping effect.

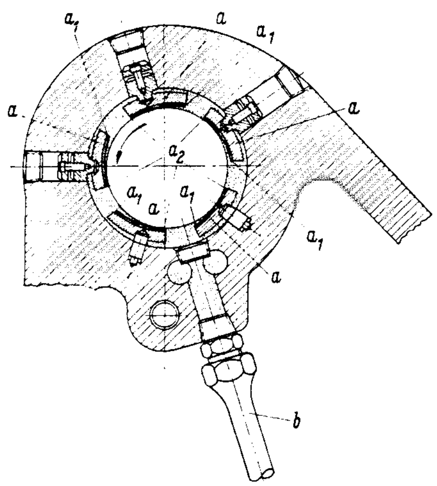


FIG. 437. "Filmatic" grinding spindle bearing (Cincinnati).

The possibility of purchasing ball or roller bearings as ready-made units relieves the designer of the rather specialized task of designing them, a job which requires considerable skill and special experience. As these bearings are designed and produced by specializing manufacturers, this has the advantage of high quality in design and performance being obtainable at a relatively low price. A similar development is now taking place in the field of plain bearings. Amongst the various available types, an expansion bearing (Fig. 438)[114] is interesting, in which a specially shaped bearing bush a is supported by a housing b, which in turn is held in position at its ends b_1 and b_2 by the headstock bore. In this self-aligning design, small bearing diameter changes due to heat expansion are also automatically compensated. The internal shape of the bearing bush is similar to that shown in Fig. 434 and provides wedge-shaped oil pockets which favour hydrodynamic lubrication.

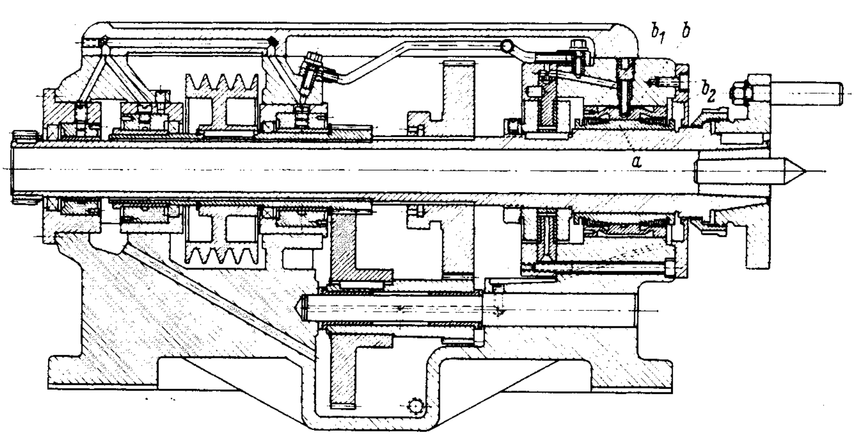


FIG. 438. Lathe spindle with ready-made plain bearings.

4. CUTTING DRIVES

The cutting drive produces the relative movement between the cutting edge and the workpiece material as required for the particular cutting operation and at the desired speed (the cutting speed, see page 1) and transmits the necessary power as far as possible without vibrations. The cutting speed depends upon the speed of the driving motor and the transmission ratio between the motor shaft and the driven element (spindle, ram, etc.). The mechanism providing the required transmission ratios can be designed so as to produce either a stepped range (gear drives, see page 109), or an infinitely variable range (see page 119). In both cases, the drives must be so designed that the resulting cutting speeds satisfy the range of working conditions required. The net power at the cutting

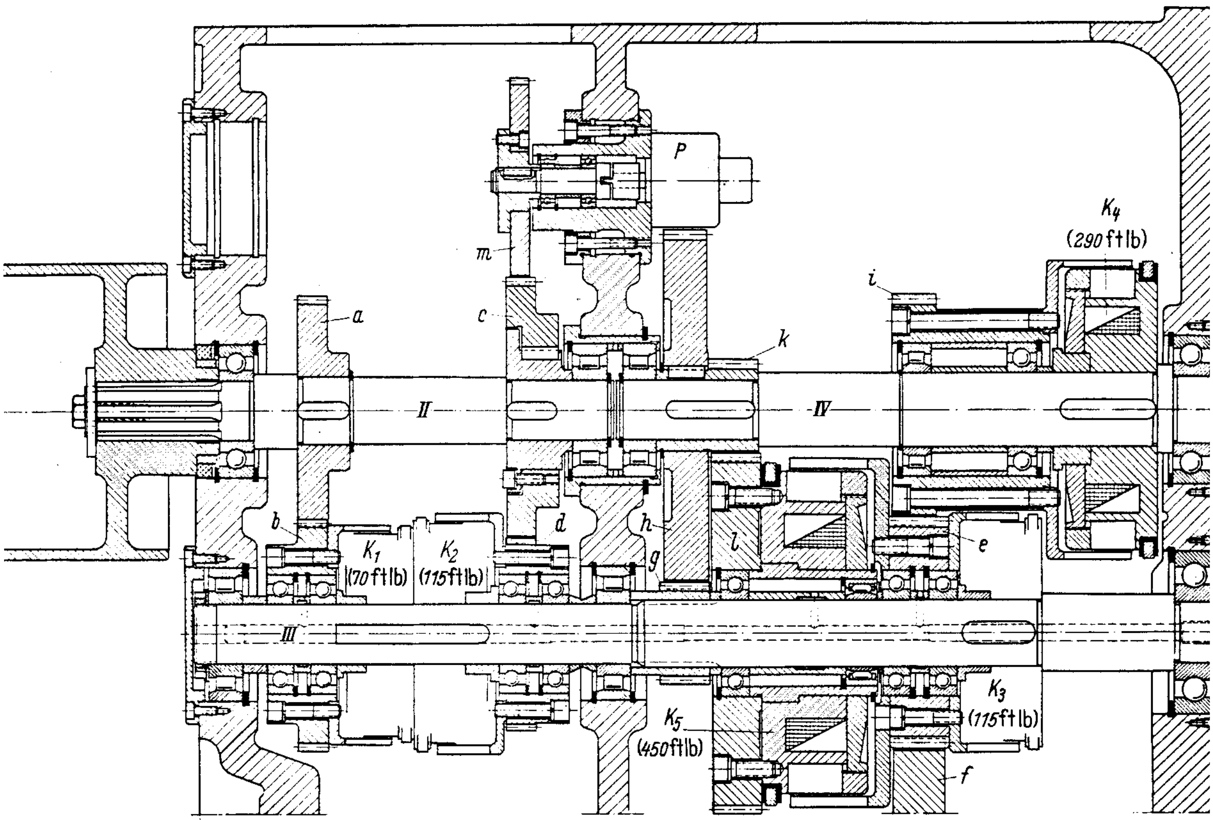


FIG. 439. Spindle head gearbox of a turret lathe (Gebr. Heinemann, St. Georgen, Germany).

edge is equal to the product of the main force component P_1 and the cutting speed v. For a given power input, the forces and torques which have to be transmitted by slow running elements (gears, worm wheels, lead screws, etc.) may become considerable and so also will the corresponding deformations of these elements. Under pulsating loads, these deformations can lead to vibrations which would affect unfavourably the quality of the machined surfaces. For this reason, the elements of cutting drives and their bearings must be strong, stiff and free from vibrations. It is important in this connexion for the necessary high ratios for the generation of low speeds to be arranged as closely as possible to the last members in the train of driving elements, so that only these last members are highly loaded, whilst the more numerous other members running at relatively high speeds are, therefore, working under much lower loads.

The gearbox for the spindle head of a turret lathe (Fig. 439) is driven by a flat belt from a pole change motor having two output speeds (1400 and 2800 rev/min) under load. Speeds are changed by means of electromagnetic clutches K_1 to K_5 and such changes are, therefore, possible under load. The maximum transmissible torques which are determined by the clutches are also indicated in Fig. 439. Figure 440 shows the speed diagram and also the torques which can be transmitted before the fixed reduction ratio e/f. Clutch K_1 on shaft *III* transmits a smaller torque than clutch K_2 on the same shaft. This limits the torque which can be transmitted at the lowest speed and which

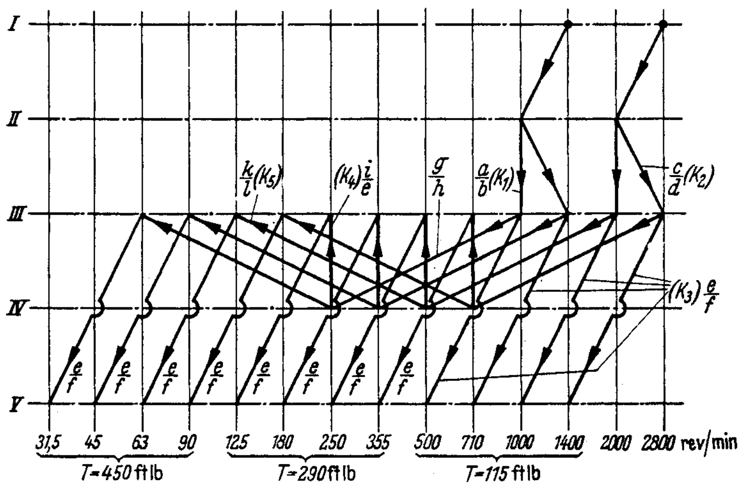


FIG. 440. Speed diagram for the drive (Fig. 439). *I*—Motorshaft (Pole change motor with 2 speeds not shown in Fig. 439): *II*—Belt pulley shaft; *V*—Main spindle. The maximum transmissible torques appear below the speed values.

stresses shaft *III* over its full length, so avoiding overload and excessive deformation of this shaft. It is also interesting to note the arrangement of the gears b, d, e, i and l, which can idle on their respective shafts and are carried on ball, roller and needle roller bearings, as well as the mounting of the clutch bodies on the gear wheels. The lubrication pump P is driven directly from the first shaft in the gearbox (*II*) via gears c and m.

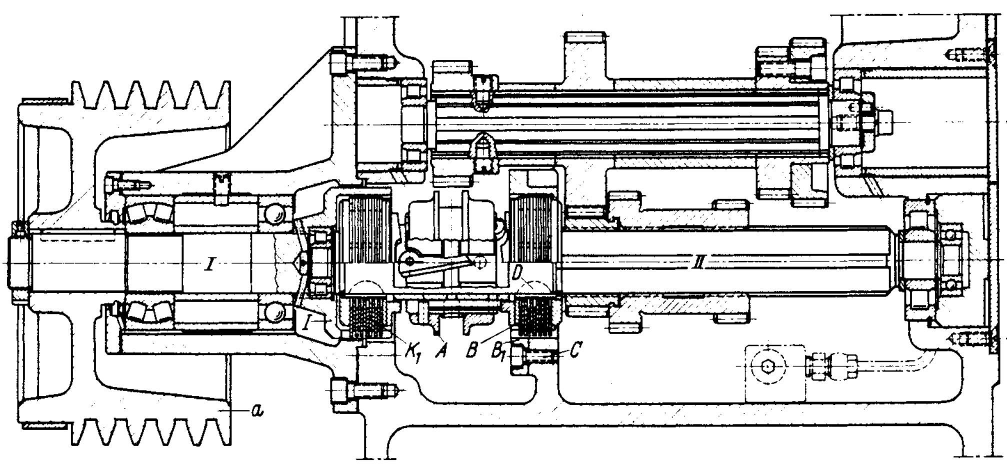


FIG. 441. Spindle drive of a copying lathe (Georg. Fischer A.G., Schaffhausen, Switzerland).

The gearbox (Fig. 441) of a copying lathe is equipped with sliding gears. The main driving shaft *I* is driven from the Vee-belt pulley a. Between it and the first gear shaft *II* a disc clutch K_1 is arranged, and this is operated by the sliding bush A. When the clutch is disengaged by moving the bush A to the right, the brake B is engaged and this serves for stopping the spindle rapidly. The brake is designed in the form of a disc clutch, the outer part B_1 of which is bolted to the gearbox casting (bolts *C*), whilst the inner part is keyed to shaft *II* by means of a Woodruff key *D*.

The gearbox (Fig. 442) of the pillar drill (Fig. 443) is driven via an infinitely variable friction transmission *a*/*b*, which covers a speed range ratio of 3 : 1. This is magnified to a speed range ratio of 22·5 : 1 by means of sliding gears *e*/*f*, *g*/*h* and *i*/*k* (Fig. 444). As the feed drive must be positively related to the spindle speed, it is taken from the pinion *1* which is fitted to the driving bush for the drill sleeve (see Fig. 454). The infinitely variable speed change is obtained by displacing the motor carrier *A* relative to the gearbox *B* along slideways B_1 and is operated by means of lever *C*, shaft

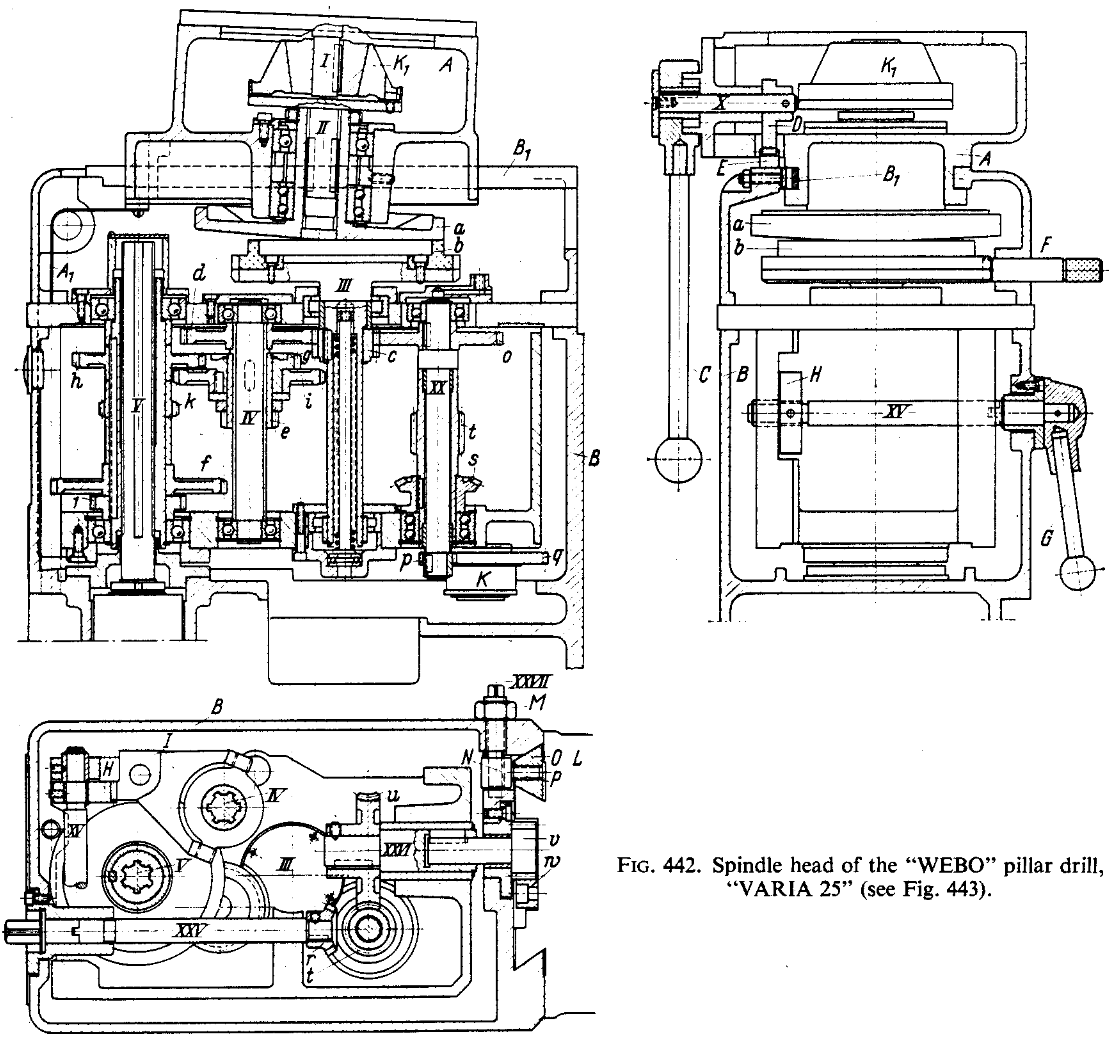


FIG. 442. Spindle head of the "WEBO" pillar drill, "VARIA 25" (see Fig. 443).

X and gear segment *D* which is in mesh with the fixed rack *E*. The actual position of the motor carrier is shown to the operator by means of the tape A_1 at the front of the gearbox. In order to maintain the necessary pressure between the driving and the driven friction wheels, the pinion *c* and the wheel *d* have helical teeth and their axial force component, supported by a compression spring in the bore of shaft *III* presses the friction disc *b* against the driving disc *a*. For maintenance purposes, it may be necessary to separate the friction discs and for this reason, it is possible to lock disc *b* against rotation by means of index pin *F*. By rotating the main spindle by hand, the effect of the helical gears *c* and *d* can then be used to separate the friction discs.

The movement of the sliding gears *e*, *g* and *i* for selecting the infinitely variable speed ranges, is operated by lever *G*, shaft *XV*, gear segment *H* and fork *I*. The pinion *c* also drives the lubrication pump *K* via gear *o*, shaft *XX*, pinion *p* and wheel *q*. The whole spindle head can be raised or lowered by means of a crank handle on the shaft *XXV*. This shaft *XXV*, drives bevel gears *r* and *s* and a worm *t*, which in turn drives a worm wheel *u*, shaft *XXVI* and pinion *v*. This pinion runs along the rack *w*, which is fastened to the pillar *L*, and thus moves the spindle head *B*. The square head screw *XXVII*, with its lock nut *M* and eccentric *N*, serves for fine adjustment of the strip *O*, after this has been set coarsely by means of screw thread *P*.

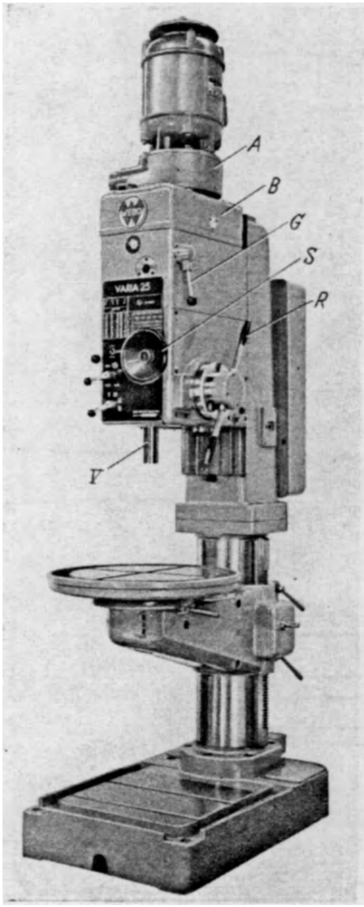


FIG. 443. "WEBO" pillar drill "VARIA 25" (Rheinische Maschinenfabrik und Eisengiesserei, Anton Röper K.G., Dülken, Germany).

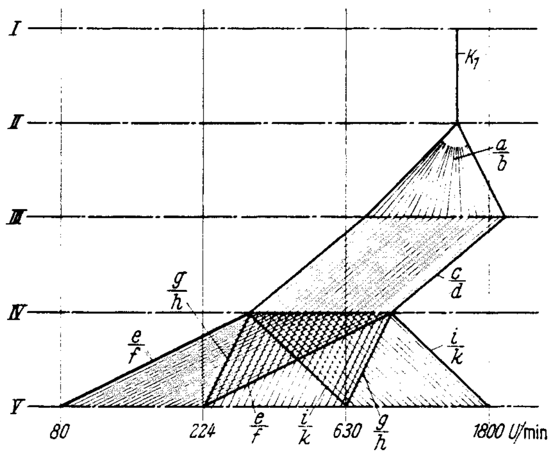


FIG. 444. Speed diagram of the gearbox (Fig. 442). *I*—Motor shaft; *II*—Friction disc shaft; *III*—Driven friction drive shaft; *IV*—Sliding gear shaft; *V*—Drilling spindle; K_1—Coupling.

The spindle drives for milling machines must often cover very wide speed ranges. They must thus be able to transmit without excessive vibrations both heavy torques and torque pulsations of high frequencies. The application of a fly wheel type element (see page 68) may reduce torsional vibration conditions. In addition, the bearings of the main spindle (see page 276) and the layout and assembly of the driving elements (gears) on the spindle (see page 270), must be carefully studied. A high transmission ratio between the final driving shafts and the spindle for the lower spindle speeds enables all other driving shafts (except the spindle), to run at relatively high speeds (see Fig. 124). The idea of producing a number of different gearboxes by exchanging some gears and without costly modifications (see page 80) has been developed for the gearboxes of production milling machines (Fig. 445). Pick-off gears *a*/*b* are used for setting the speeds within various speed ranges (step ratio 1·25). For the lowest speed range (7 to 90 rev/min), a worm and worm wheel drive *c*/*d* is used (Fig. 445a), whilst higher spindle speeds (up to 335 rev/min) are obtained through gears *e*/*f*. For changing from a high to a low speed range (i.e. from the gear drive to the worm drive), or vice versa, a clutch *K* is provided (Fig. 445b). If the lower speed range is not required, the gear drive *e*/*f* alone is used (Fig. 445c). In the case of high spindle speeds, endless belt drives are preferable, because gear drives can give rise to detrimental vibrations, even if the gearing errors are small. However, great care must be taken that the forces to be transmitted by the belts are not excessive and that the belt speeds are not too low, a problem which is difficult in spindle heads because of the limitation of the pulley diameters due to the space available. In the layout (Fig. 445d), the gear drive e/f has been retained for the lower spindle speeds, whilst for the higher speeds the clutch K connects the spindle with the belt drive g/h. At these high speeds, the belt speeds can be high enough for the purpose and the torques resulting from the available motor power can be transmitted by the permissible belt tension. If the high speed range only is required, the belt drive alone is sufficient (Fig. 445e).

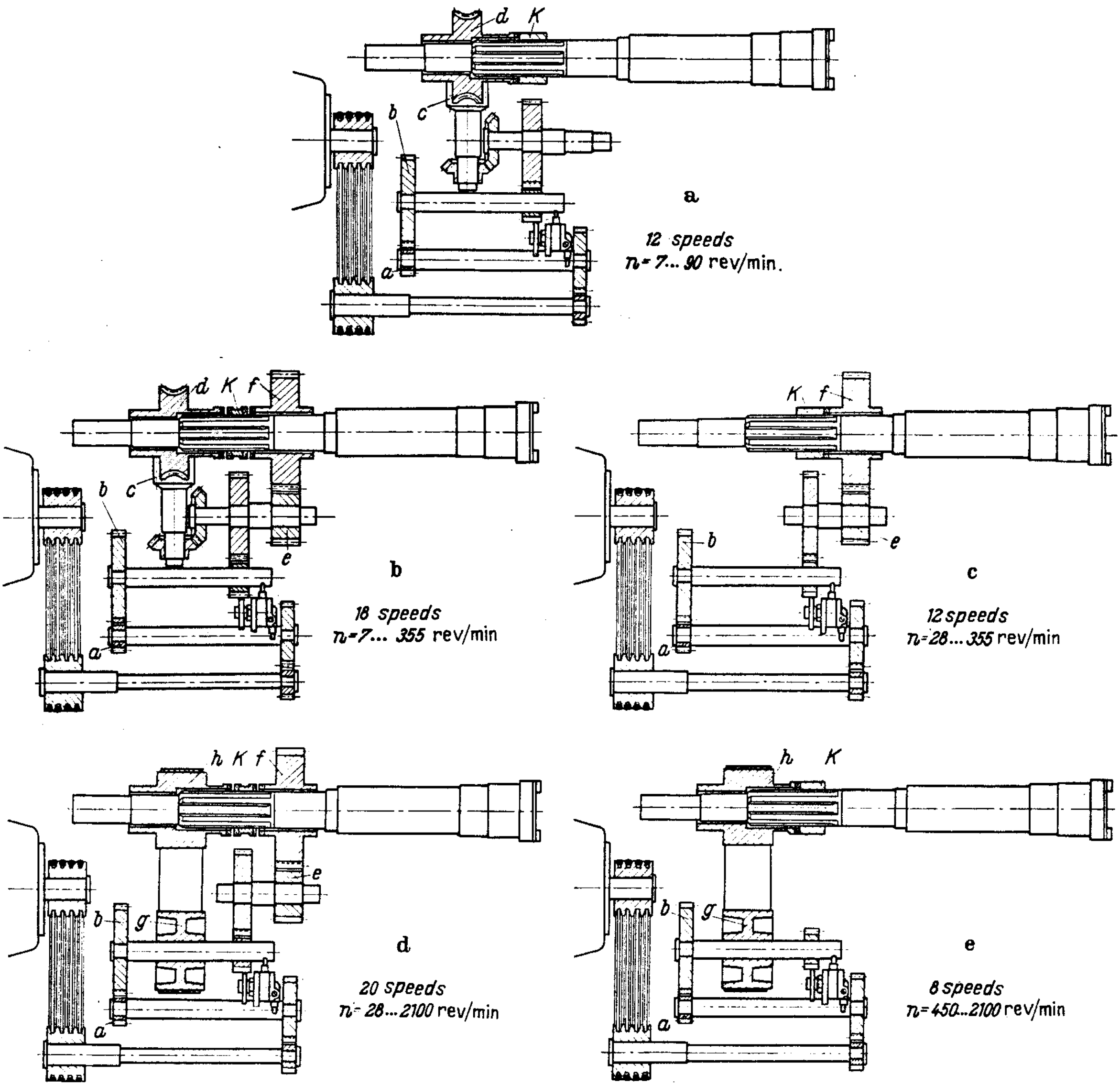


FIG. 445. Spindle drives for a production milling machine (Wanderer, Haar, Germany).

In the design of heavy spindle drives for vertical milling machines particular consideration must be given to the possibility of accurately aligning the driving elements, especially bevel gear drives. In addition, the possibility of axial movement of the spindle and the problem of bearing lubrication must be carefully studied. The bevel gear drive a/b of the vertical milling spindle (Fig. 446) is carried in a housing A, which is bolted to the column of the machine. The horizontal driving shaft I is supported in bearings which are housed in a separate bush B in the housing A. The bush B also serves as a spigot for locating the housing A on the machine column C. The spindle II is carried in taper roller and ball bearings in the headstock D, which can slide vertically on the housing A.

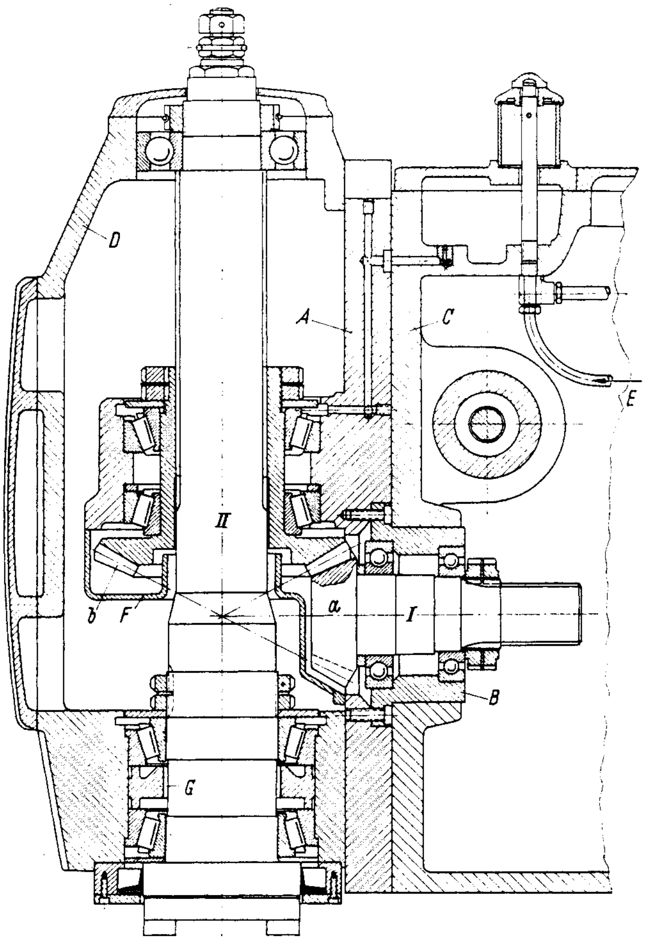


FIG. 446. Spindle head of a vertical milling machine (Wm. Asquith Ltd., Halifax).

The spindle is shown in its top position and when the headstock is moved vertically downwards the spindle slides in the splined bore of the bevel gear b.

The bevel gears and the bearings are lubricated through a pipe E, which is supplied from a central lubrication pump. The trough F prevents oil from running into the headstock. The spacing ring G between the two taper roller bearings for the main spindle support prevents excessive loss of oil from the upper bearing.

For lighter work, a universal milling head, which can be applied to ordinary horizontal milling machines, often proves of great value. In the orthodox designs, the milling head can be swivelled around two axes I and II lying at right angles to each other (Fig. 447). A novel and extremely stiff design is shown in Fig. 448. Here, the tilting movements are carried out around two axes I and II which lie at 45° to each other.

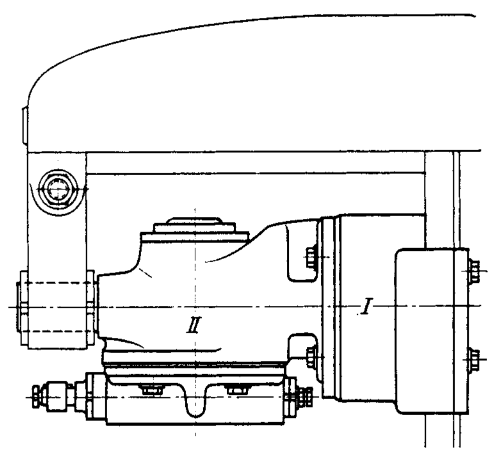


FIG. 447. Universal milling head (Wanderer, Haar, Germany).

The design of rectilinear cutting drives for the

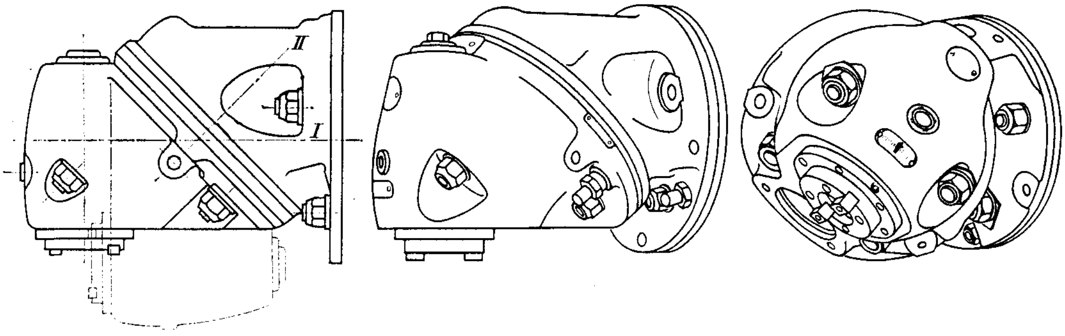


FIG. 448. Universal milling head (Société Léon Huré & Cie, Paris, France).

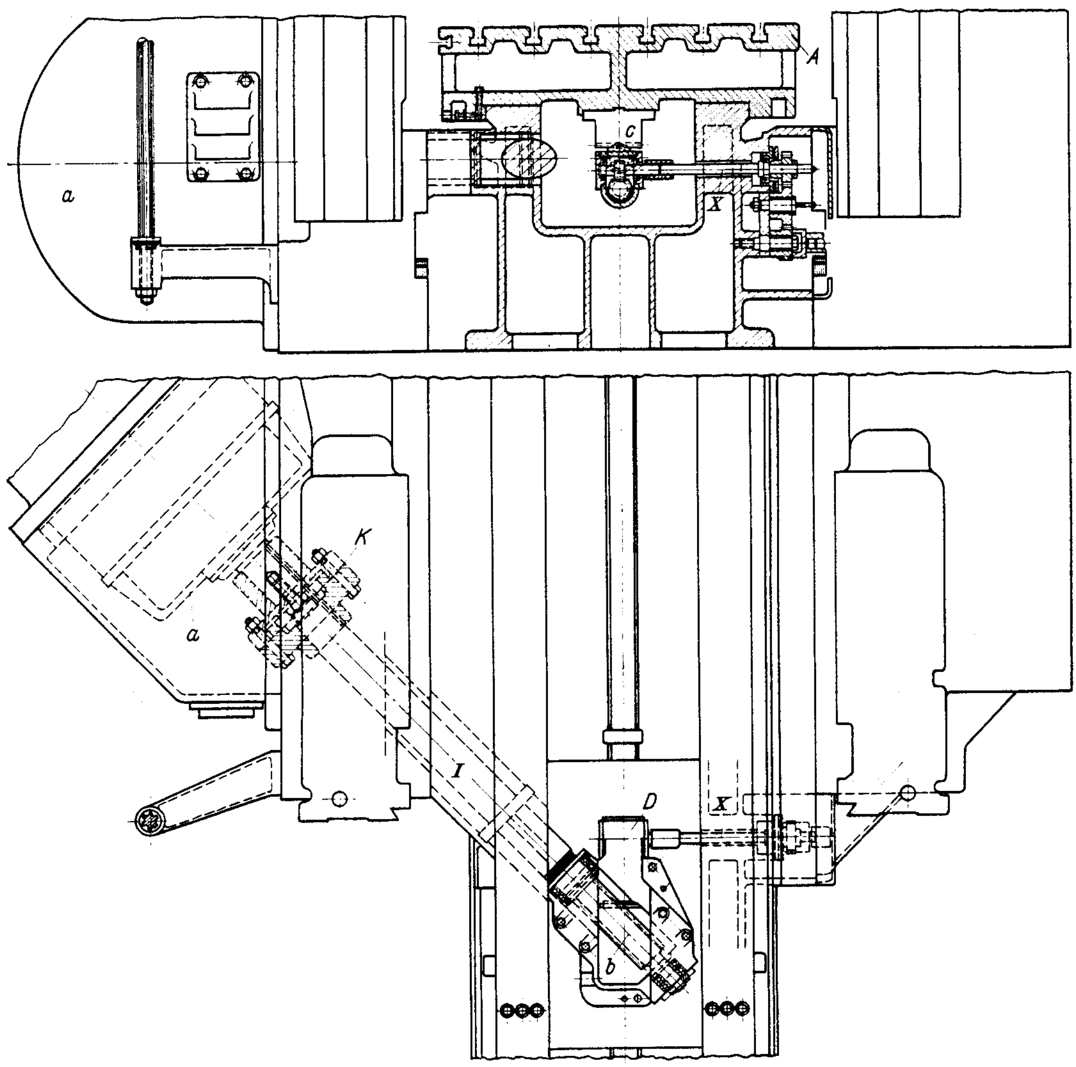


FIG. 449. Planing machine table drive with worm and rack with inclined teeth (John Stirk & Sons Ltd., Halifax).

tables of planing machines must take into account not only the possible high cutting forces, but also the inertia forces which occur during the reversing action. As long ago as 1910, Schlesinger carried out a detailed calculation for an old planing machine and determined not only the very high forces acting on the driving gears and the rack during cutting operations, but also found that the magnitude

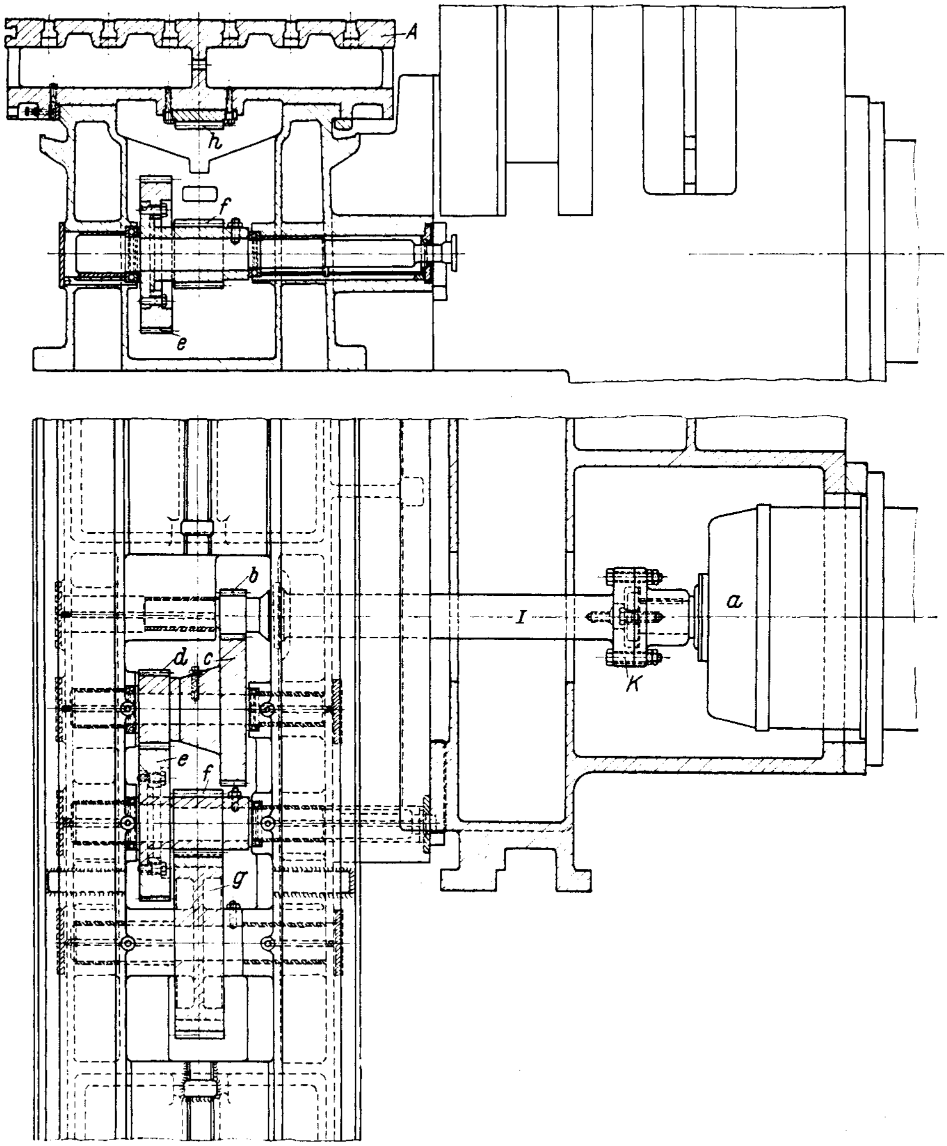


FIG. 450. Table drive for an open sided planer (John Stirk & Sons, Halifax).

of the energy required to reverse the masses which rotated at high speeds (pulleys, gear and shafts), was about 30 times that required for reversing the masses which moved along straight lines (table and workpiece).

In order to obtain quiet running conditions together with smooth force transmission between the rotating driving element and the rack for the table drive it is often advisable to use either a small diameter worm driving a rack with inclined teeth (Fig. 449), or a gear wheel of large diameter, often with helical teeth, which ensures simultaneous action of a number of teeth (Fig. 450). In the case of the worm drive (Fig. 449), the driving motor *a* drives the shaft *I* via coupling *K*. Shaft *I* carries the worm *b* and this in turn drives the rack *c* which is fixed to the table *A*. The simplicity of this drive is obvious for it does not require intermediate gears between the motor and the relatively short worm shafts.

If a helical gear wheel drive is used, with the necessary intermediate speed reductions, the required torsional stiffness can be obtained by eliminating long shafts and by arranging the gear wheels in blocks of two. In the design shown in Fig. 450 the driving motor *a* drives shaft *I* via coupling *K*. Shaft *I* carries the pinion *b* and the drive is then transmitted by gears *b/c*, *d/e*, *f/g*, the large size gear wheel (the "bull wheel") *g* driving the rack *h* on the table *A*. The gear blocks *c* and *d*, *e* and *f* and the very short bearing distances for all shafts ensure the required transmission stiffness.

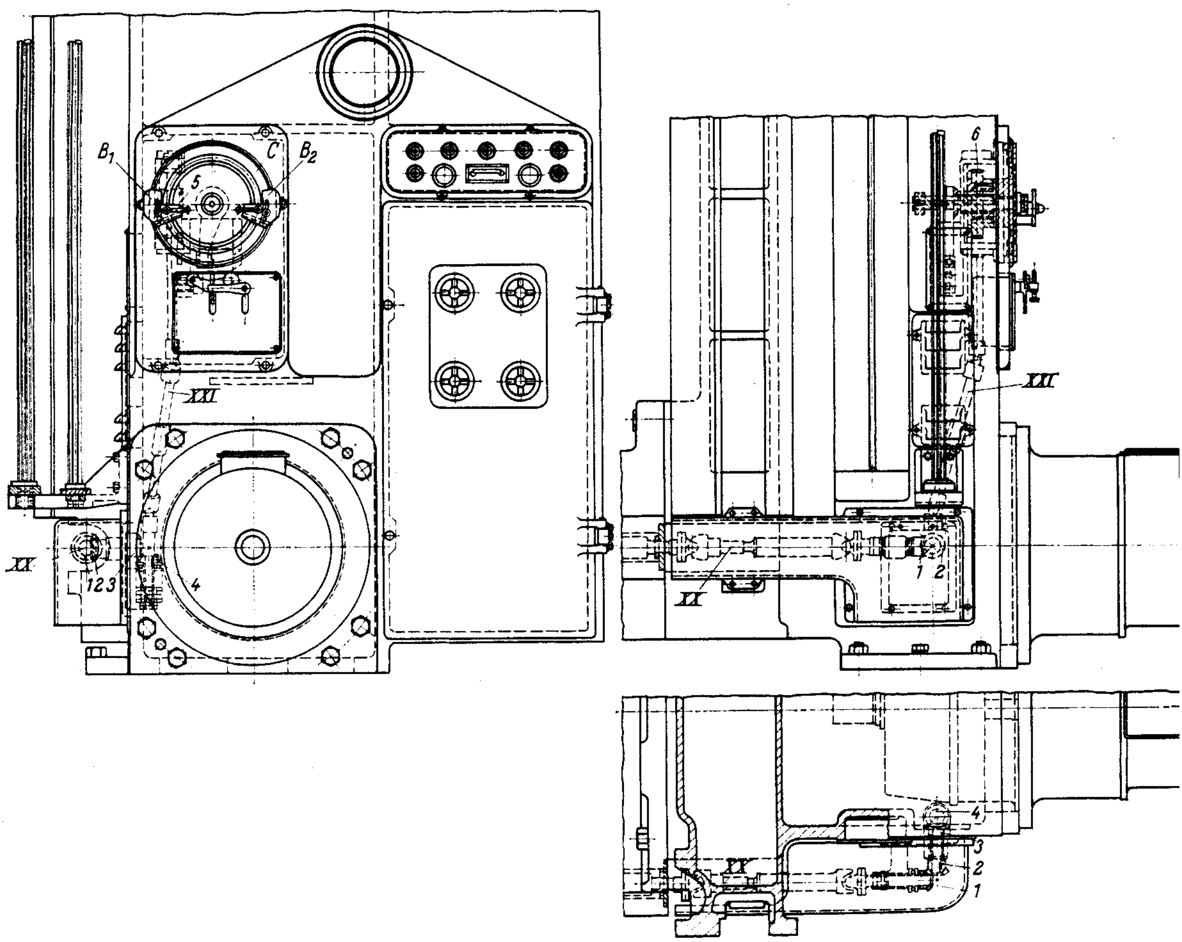


FIG. 451. Reversing drive of planing machine (John Stirk & Sons, Halifax).

The bull wheel *g* is only an intermediate gear of large diameter which ensures the satisfactory meshing conditions with rack *h*. The drive is reversed electrically (see page 148) and the reversing positions of the table movement are determined by adjustable trip dogs B_1 and B_2, located on a rotating disc *C*, which is geared to the table drive (Fig. 451). The drive for the disc *C* and the table drive are connected through gearbox *D* and shafts *X* and *XX* (see Figs. 449 and 451). From shaft *XX* (Fig. 451), the drive for the reversing disc *C* is then transmitted by bevel gears *1*, *2*, 3 and *4*, shaft *XXI* and helical gears *5* and *6*.

5. FEED AND SETTING DRIVES

With the exception of a few processes, e.g. cylindrical milling and grinding, the feed movement is usually rectilinear. It must be either dependent on (turning, drilling) or independent of (milling) the cutting movement, and it may be either continuous (turning, drilling, milling) or intermittent (planing). In the latter case, it is perhaps not correct to refer to a feed movement, but rather a setting movement, similar to that which occurs during surface grinding after each feed traverse, in addition to the feed movement proper. As far as the feed drive is concerned, the power requirements at the cutting edge are usually relatively low (see page 36) and although the more complicated feed drive gearboxes often have a relatively low efficiency (see page 36), usually it is not found difficult to control the powers which have to be transmitted. However, stiffness and rigidity against vibration of the driving elements are important, especially if the stroke of a feed, or setting, movement has to be within fine limits (see page 69).

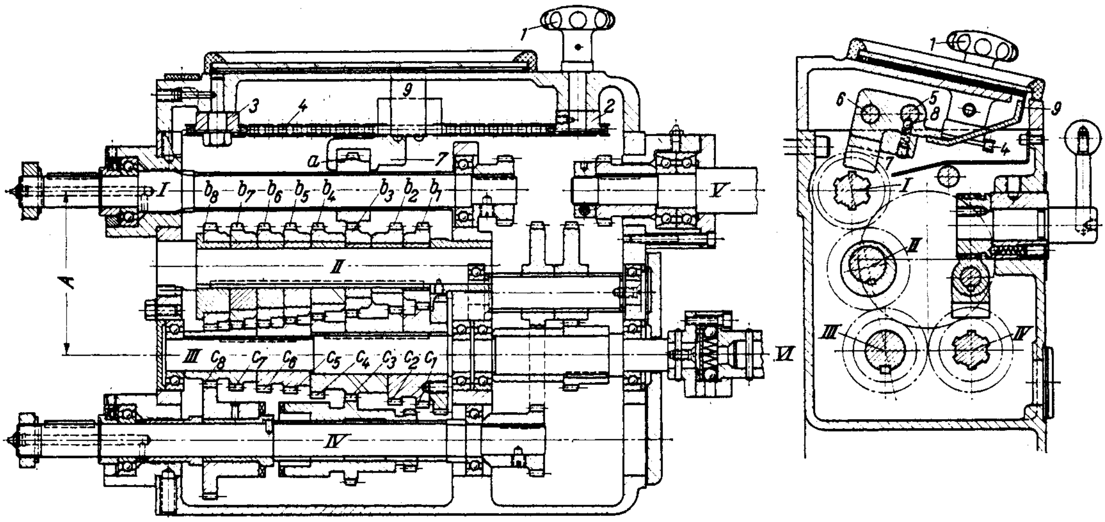


FIG. 452. Feed gearbox of a centre lathe (Dean, Smith & Grace Ltd., Keighley).

The most important distinction between the various feed movements is perhaps their dependence on or independence from the cutting movement. In both cases the power transmission must be continuous and smooth so that the resulting surface finish satisfies the requirements laid down for the workpiece which is being machined. In the case of dependent feed movements, their speed must be kept within given limits of accuracy in relation to the speed of the cutting movement, so that not only a uniform surface pattern is maintained (e.g. in turning, drilling), but also a certain pitch accuracy when this is required (e.g. in screwcutting).

Finely stepped drives are essential in order to make available the many different feed rate values which may be required during turning and screwcutting operations. The Norton gearbox (see Fig. 144) and the draw-key type (see Fig. 147) provide compact designs for a finely stepped and wide range. Both have, however, weaknesses due to the swinging pinion carrier or the draw key (see pages 111–113). Figure 452 shows a sliding gear layout for the feed drive of a centre lathe in which the high load carrying capacity of the sliding gear drive is combined with the compact layout of the Norton gearbox. The drive is derived from the main spindle and transmitted to the gearbox by means of pick-off gears, via either the shaft *I* or the shaft *IV*. The output shaft drives either the lead screw *V* or the feed shaft *VI* of the lathe. The speed change gears between shafts *I* and *III* consist of a pinion a, which slides on the splined shaft *I* and drives 8 intermediate gears b_1 to b_8, which idle on eccentric bushes keyed to the fixed shaft *II*. The intermediate gears drives the 8 gears c_1 to c_8 on the output shaft *III*. The eccentricity of the bushes which carry the gears b_1 to b_8 is chosen in accordance with the differences of their pitch circle diameters in such a manner that the sum of the centre distances between gear a and gears b_1 to b_8 on the one hand and the gears b_1 to b_8 and c_1 to c_8 on the other, remains constant. In other words, if the diametral pitch of the gears is denoted by P and the numbers of teeth in them n_{ta}, n_{tb}, etc., then:

$$\begin{aligned}\frac{n_{t_a}}{P}+\frac{2n_{tb_1}}{P}+\frac{n_{tc_1}}{P} &= \frac{n_{t_a}}{P}+\frac{2n_{tb_2}}{P}+\frac{n_{tc_2}}{P}\\ &= \frac{n_{t_a}}{P}+\frac{2n_{tb_3}}{P}+\frac{n_{tc_3}}{P}\\ &= \dots\dots\dots\dots\\ &= \frac{n_{t_a}}{P}+\frac{2n_{tb_n}}{P}+\frac{n_{tc_n}}{P}\\ &= \text{const.} = 2A\end{aligned}$$

where A is the centre distance between shafts I and III. With gear b_1 running on a concentric bush, the centre distance between shafts I and II is:

$$A_1 = \frac{1}{2P}(n_{t_a} + n_{tb_1})$$

and the centre distance between gears a and b_n:

$$A_n = \frac{1}{2P}(n_{t_a} + n_{tb_n})$$

Hence the eccentricity of the bush for any gear b_n can be determined by:

$$e_n = A_n - A_1 = \frac{1}{2P}(n_{tb_n} - n_{tb_1})$$

For moving the sliding gear a, a handwheel *1* through sprocket wheel *2* moves a chain *4*, which is held in tension by another sprocket *3*. The moving fork *7* slides on shafts *5* and *6* and is fastened to the chain *4*. A spring loaded steel ball *8* fitting into notches on shaft *5*, determines the various positions of the fork *7* and with it those of the sliding gear a. The corresponding feed rate is shown by a pointer *9* fixed to the fork *7*.

For accurate screwcutting, the lathe saddle is usually driven by a lead screw (*V*, see Fig. 452), which rotates at the required speed in the feed nut of the saddle. The requirements of feed rate accuracy are not so high for normal turning operations. In order not to wear the lead screw unnecessarily, the feed shaft (*VI*, see Fig. 452) is used for driving the saddle via a gear and worm arrangement in the apron and a pinion which in turn is in mesh with a rack on the lathe bed.

In the apron (Fig. 453), the half nut *1* is moved into engagement with the lead screw *V* by means of screw *3*. The feed shaft *VI* drives gears *11/12*, overload clutch *13* (see Fig. 260), shaft *VII*, worm and worm wheel *14/15* and gears *16/17*. At this point, it is possible to engage either clutch *21* or clutch *31*. Clutch *21* is for longitudinal turning by means of gears *22/23*, coupling *24*, shaft *VIII*, pinion *25* and rack *26*, which is fixed to the bed. Clutch *31* is for surfacing through gears *32/33* and cross traverse screw *34*. It may be pointed out that all long shafts are arranged before the worm drive *14/15*, and that in the case of all gears which run at low speeds and are therefore subject to high torques, the torques are not transmitted by the shafts, but through the gear blocks *15–16*, *17–21–22*, *17–31–32*, *23–24–25*. The change from longitudinal to cross traverse is effected by the hand lever *2* through gears *41*, *42*, *43* which operates the clutches *21* or *31* respectively. The peg *50* serves for interlocking the lead screw against the feed shaft drive for the longitudinal traverse as, obviously, both must not be engaged simultaneously.

The manual longitudinal traverse is operated by hand-wheel *61*, shaft *X*, gears *62/63*, *64*, *65* and *23* and from there again via coupling *24*, pinion *25* and rack *26*. The shaft *VIII* can be withdrawn by means of a small handle and the pinion *25* can therefore be disengaged from the rack *26*. The pinion and its driving gears thus need not rotate all the time in the case of the lead screw drive.

Figure 454 shows the feed drive for the drilling machine (Fig. 443). The drive is taken from the pinion *1*. This forms part of the gear block *f*, *h*, *k*, which drives the drilling spindle (see Fig. 442). Pinion *1* drives gear *2*, shaft *XXX*, worm drive *3/4*, electromagnetic disc clutch (*5*) and shaft

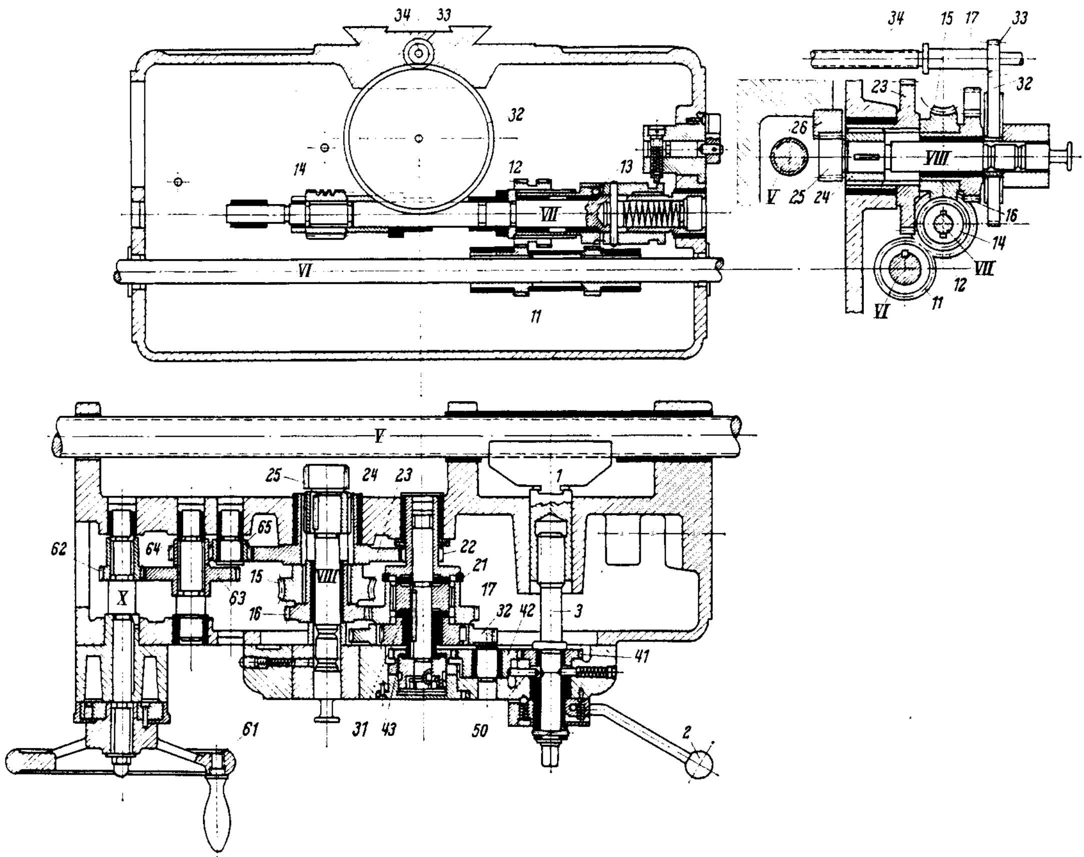


FIG. 453. Apron of a Schaerer centre lathe (Industrie-Werke, Karlsruhe, Germany).

XXXI. This is a splined shaft with a sliding gear block *6*, *8*, *10* and an idling gear *13* with internal teeth *12*. By moving the sliding gear block along shaft *XXXI*, it is possible to obtain 4 speeds (via *6/7*, *8/9*, *10/11* and *10/12/13/14*) of shaft *XXXII*. These speeds are transmitted through worm gear *15/16* and clutch *17* (operated by lever *R*), to shaft *XXXIII* and pinion *18*, which in turn drives the rack *19* on the spindle sleeve.

The rapid manual feed for setting the spindle into its correct starting position is also operated by hand lever *R*, which is keyed to the shaft *XXXIII*. During this operation, the clutch *17* must be disengaged, because of the irreversibility of the worm drive *15/16*. It is also possible to produce a slow feed by hand, for heavier drilling operations. This is done with hand-wheel *S* on the worm shaft *XXXII*, the high reduction of the worm drive *15/16* making it possible to overcome a high axial cutting resistance. Again, it will be necessary to disconnect the irreversible worm drive

3/4 from hand-wheel *S* and this is done by putting the sliding gear block *6*, *8*, *10* or the clutch *5* into a neutral position.

The feed rate is selected by hand-wheel *T* which is located at a height easily accessible to the operator and drives the shaft *XXXVI* (via shaft *XXXV*), sprocket *U*, chain *V* and sprocket *W*. This carries a cam and a gear shifting fork *X*. In the top and bottom positions of the spindle sleeve, the feed drive can be stopped by means of limit switches *Y*, which disengage the electromagnetic disc clutch *5* on shaft *XXXI*.

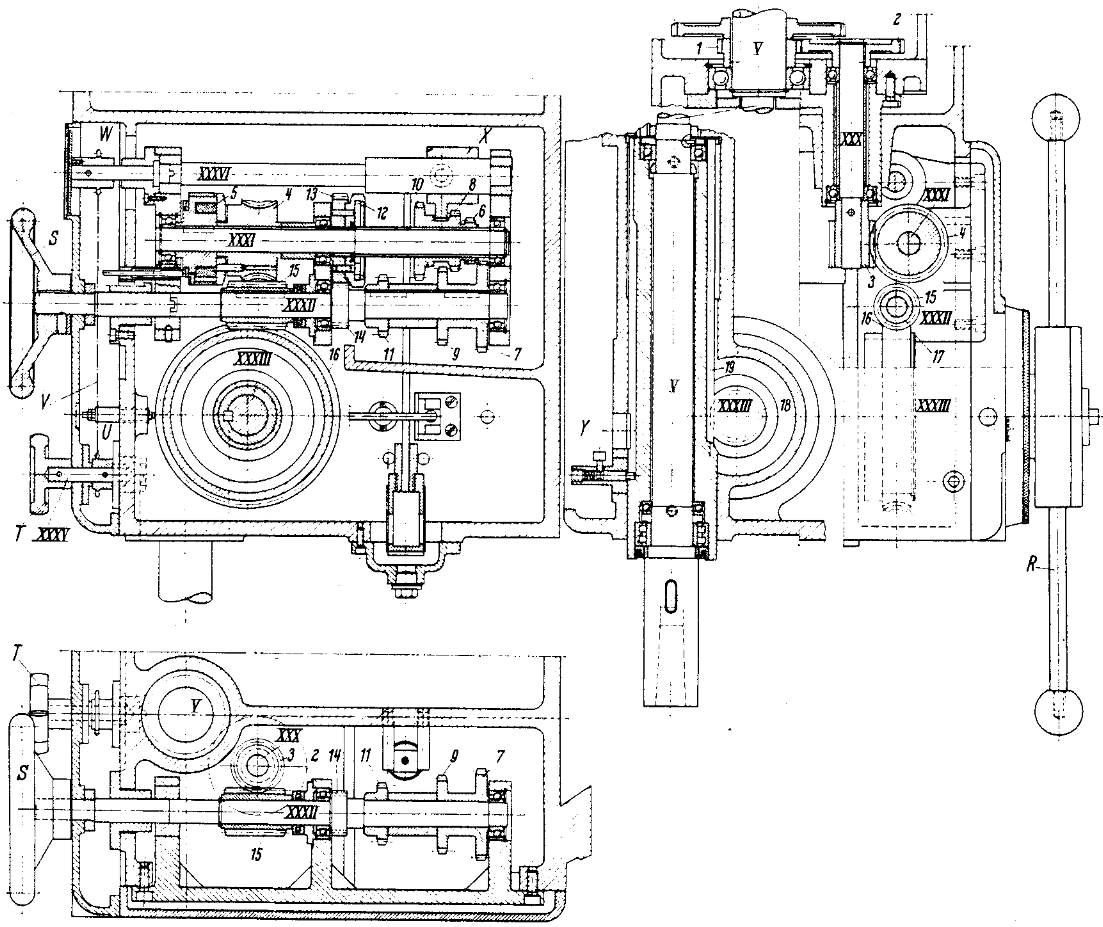


FIG. 454. Feed drive of the pillar drill (Fig. 443).

For the work of horizontal boring and milling machines, the feed rate must be related either to the spindle revolutions (in/rev, boring and screw-cutting operations), or to time (in/min, milling operations). In addition, it is often necessary to provide for the possibility of facing operations, i.e. for a tool-holder on the face plate to be moved radially as a function of the face plate revolution.

In the headstock (Fig. 455) of a horizontal boring machine the spindle and the face plate (see Fig. 419) are driven by shaft *I* of the flange motor via an infinitely variable P.I.V. gear (see page 123), intermediate shafts *II*, *IIIa* and *IIIb*, a splined shaft *IV*, a main transmission shaft *V* and either through gear n_1 and gear n on the spindle, or clutch gear i_1 and pinion i on shaft *VI*, which drives the face plate (see Fig. 419).

The headstock can be moved up or down the column during setting or feed movements, and for this reason the feed drive for the spindle, the headstock and the table are derived from a central vertical shaft *XXX* along which the headstock slides. In any position of the headstock, this

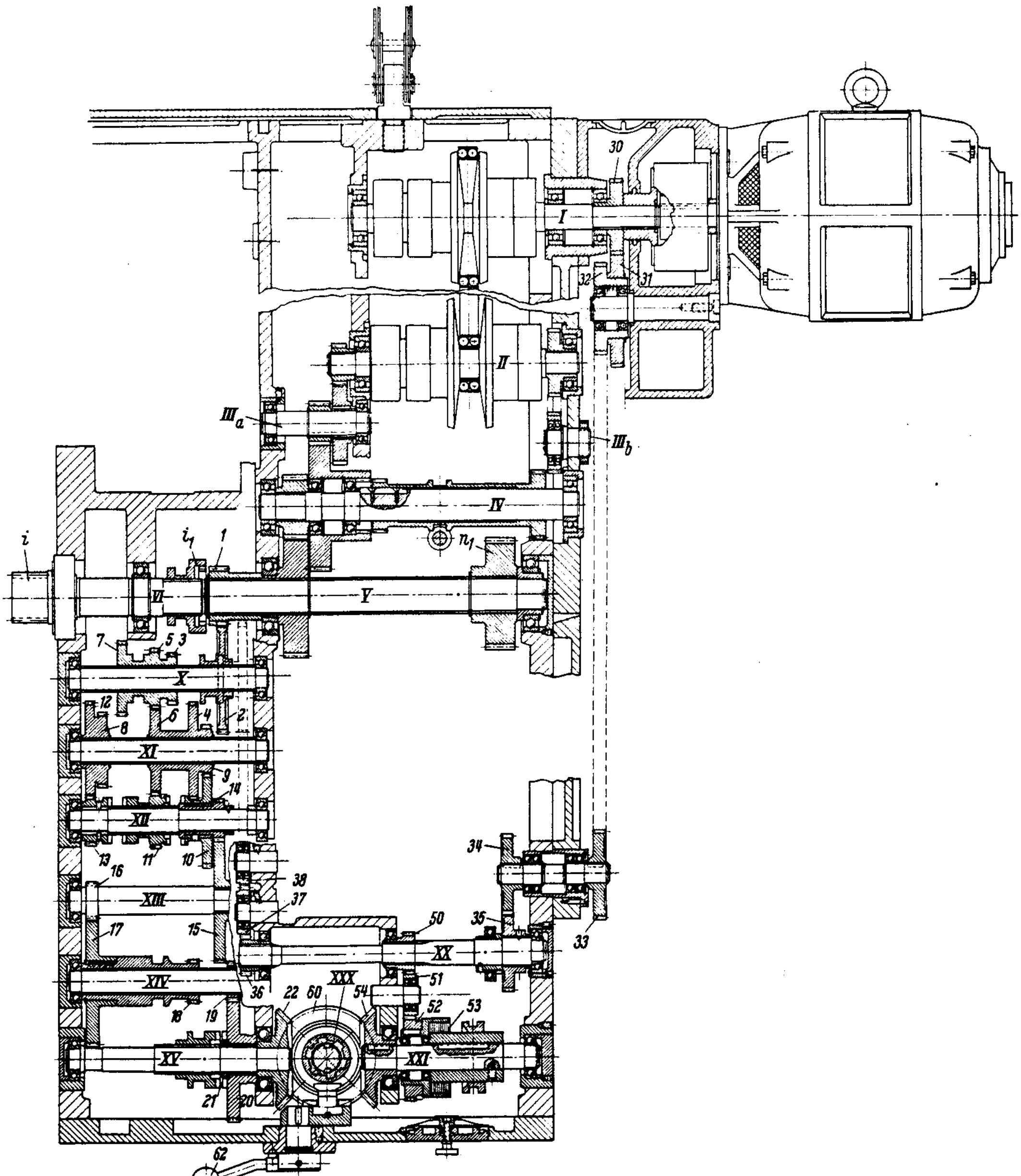


FIG. 455. Spindle and feed drive of a horizontal boring machine (Collet & Engelhard, Offenbach, Germany).

shaft is coupled with the feed drive by bevel gears. Bevel gears *60a* and *60b* on shaft *XXX* are arranged above and below the driving bevel gears (Fig. 456), so that the direction of rotation of shaft *XXX* can be determined by engaging the clutch *61* through hand-lever *62* with either the top or the bottom bevel gear.

If the feed rate is to be dependent on the spindle revolutions, it is derived from shaft *V*, which has a constant transmission ratio with the spindle (pinion n_1, wheel n, see Fig. 419). The pinion *1* drives the splined shaft *X* via wheel *2*. The sliding gears *3*, *5* and *7* on this splined shaft transmit three speeds to the shaft *XI* through gears *4*, *6* or *8*. Three further speed changes (i.e. a total of nine speeds) can be obtained for shaft *XII* by means of the sliding gear and clutch arrangement (*9/10*, *6/11* or *12/13*). Shaft *XII* drives shaft *XIII* (gears *14/15*) and shaft *XIV* (sliding gear set *16/17* or *15/18*) (18 speeds). From here, the vertical shaft *XXX* is driven by a fixed transmission *19/20*, clutch *21* and bevel gear *22*.

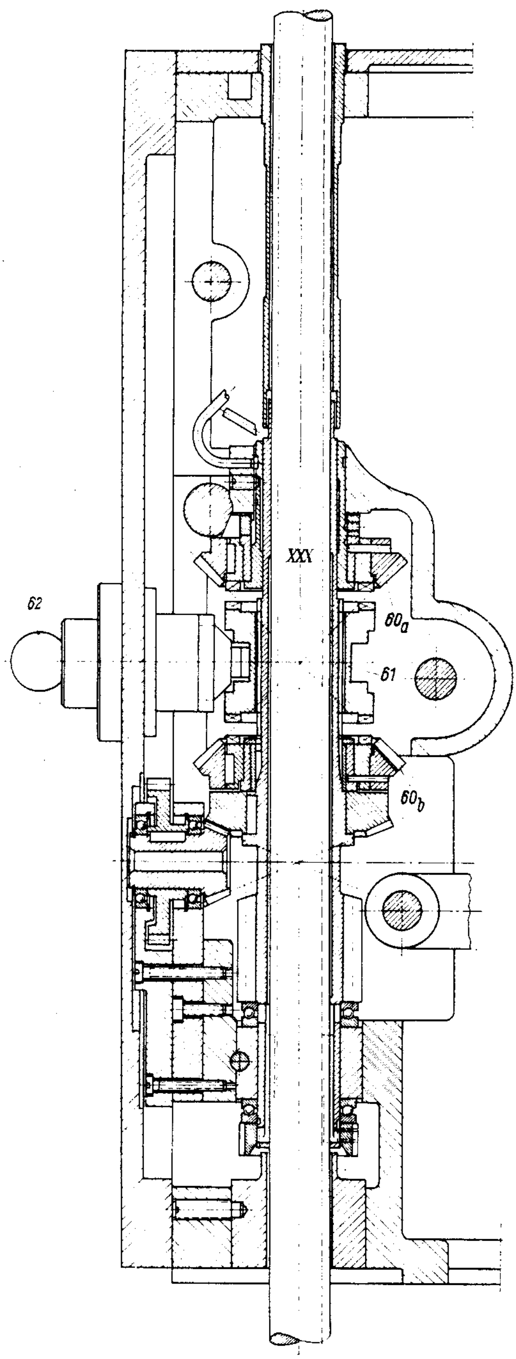


FIG. 456. Feed reversing drive of the machine (Fig. 455).

If the feed rate has to be independent of the spindle speed, the drive is derived before the P.I.V. gear, i.e. from the shaft *I* which is coupled to the motor and runs at constant speed. Pinion *30* drives shaft *XX* through a constant transmission ratio (gears *30/31*, *32/33*, *34/35*). From here, shaft *X* is driven by a further constant transmission (*36/37*, *37/38* and *38/2*, sliding gear *2* moved into mesh with *38* and, therefore, out of mesh with gear *1*), and shaft *X* provides the input to the 18-step feed gearbox and through bevel gear *22* to the vertical feed shaft *XXX*.

The rapid traverse drive has to be at constant speed. It is taken straight from shaft *XX* through gears *50/51*, *51/52* and disc clutch *53* to shaft *XXI*, which drives bevel gear *54* and through it again the vertical shaft *XXX* (bevel gears *60a* and *60b*). As bevel gear *54* drives shaft *XXX* in a direction opposite to that of the feed drive, the rapid traverse will always produce a return movement opposed to the feed movement, unless the direction of movement is reversed through clutch *61*.

Figure 457 shows a patented surfacing drive for a horizontal boring and milling machine. The face-plate *A*, which is driven by gear wheel *a* carries a sliding tool-holder *B*. This is guided in its radial movement by strips C_1 and C_2 and adjusting strips D_1 and D_2 (see page 241). The feed drive is introduced through gear wheel *1* and transmitted via helical teeth *2* on to the helical gears *3a* and *3b*. These gears in turn drive two racks *4a* and *4b*, which are fixed to the tool-holder (slide *B*). The slot *E* allows a radial movement (stroke l_E) of the tool slide, even if the boring spindle *F* is not withdrawn behind the face plate. This makes it possible to carry out boring and facing operations simultaneously. A feature which is particularly important lies in the fact that in this design the face plate has very little overhang outside the main bearing and thus combines great simplicity with a smooth drive.

In the case of milling operations, the feed movement must be smooth and vibration-free, even under the effect of the sometimes heavily pulsating forces acting at the cutter edge. The required static and dynamic stiffness of the drive can be obtained either by mechanical (large diameter feed screws, short pinion shafts, etc.), or hydraulic means (hydraulically locked piston, see page 157). The purely hydraulic drive which can be easily and accurately controlled, loses stiffness with increasing stroke due to the elasticity of the oil column (see page 126), the cylinder and the piston rod.

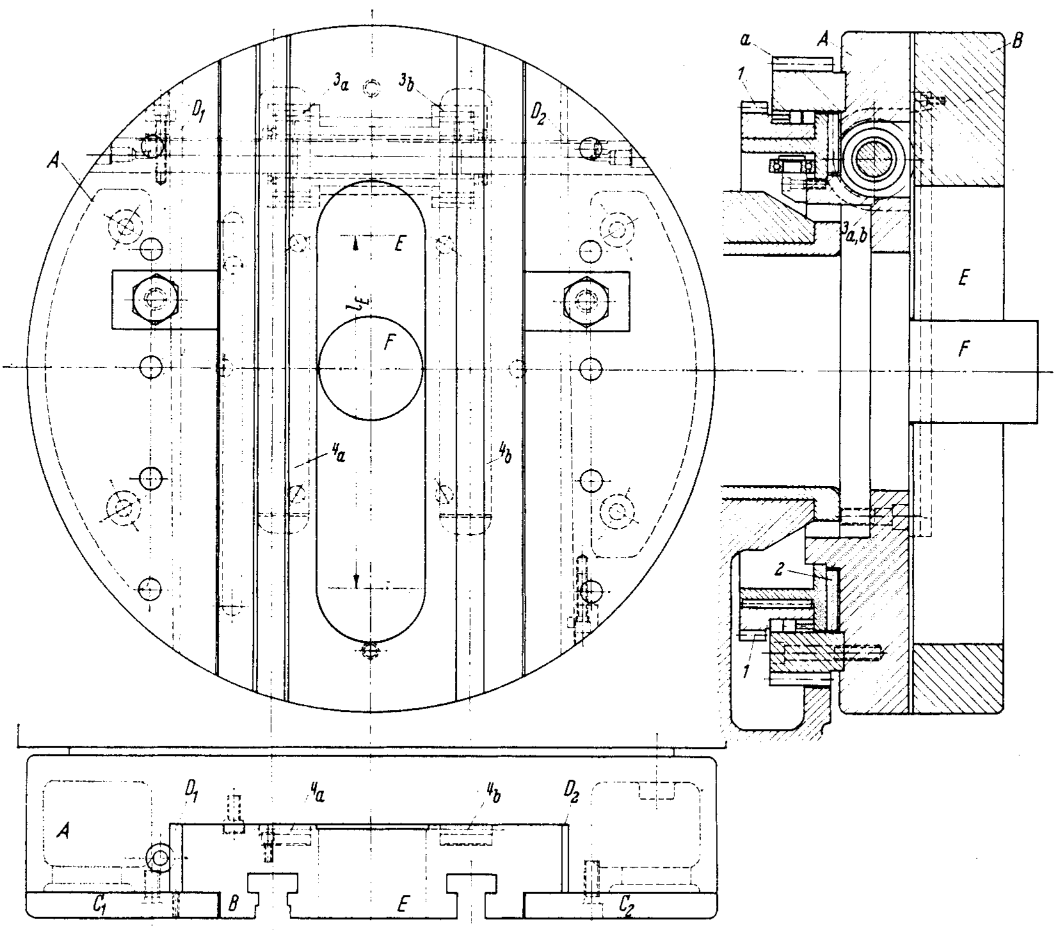


FIG. 457. Surfacing drive for a horizontal boring machine (H. W. Kearns & Co. Ltd., Broadheath).

Figure 458 shows a drive which combines, even in the case of long strokes, the stiffness of the short stroke hydraulic drive with the advantage of hydraulics and their sensitive control, especially in the case of electronic control equipment.

The milling machine table *A* carries the short stroke cylinder *B* in which a piston *D*, controlled by a valve *C*, is normally kept in its middle position. The piston rod *E* is not fixed to the machine bed but to a sliding rack *F*, which in turn is driven by an electric or hydraulic motor via gears *1*, *2* and worm *3*. This latter drive serves merely for bringing the piston continuously into its middle position in the cylinder. The rack thus resets, so to speak, any displacement of the piston in the cylinder which has taken place under the effect of the precision control by the valve. As long as the electro-mechanical drive is backlash-free, it need not work with special precision. The freedom from backlash is obtained by the worm being carried in a rocking cradle *4*, which is held against the rack by hydraulic pressure (cylinder *5*). The thrust bearings of the worm and the cradle are also hydraulically pre-loaded (cylinder *6*). Figure 459 shows in diagrammatic form the layout of this drive arranged for electronic control by means of a magnetic tape with feedback from an optical grating (see page 198).

The cutting of rolled sections and profiles with a circular saw is an operation which is similar to

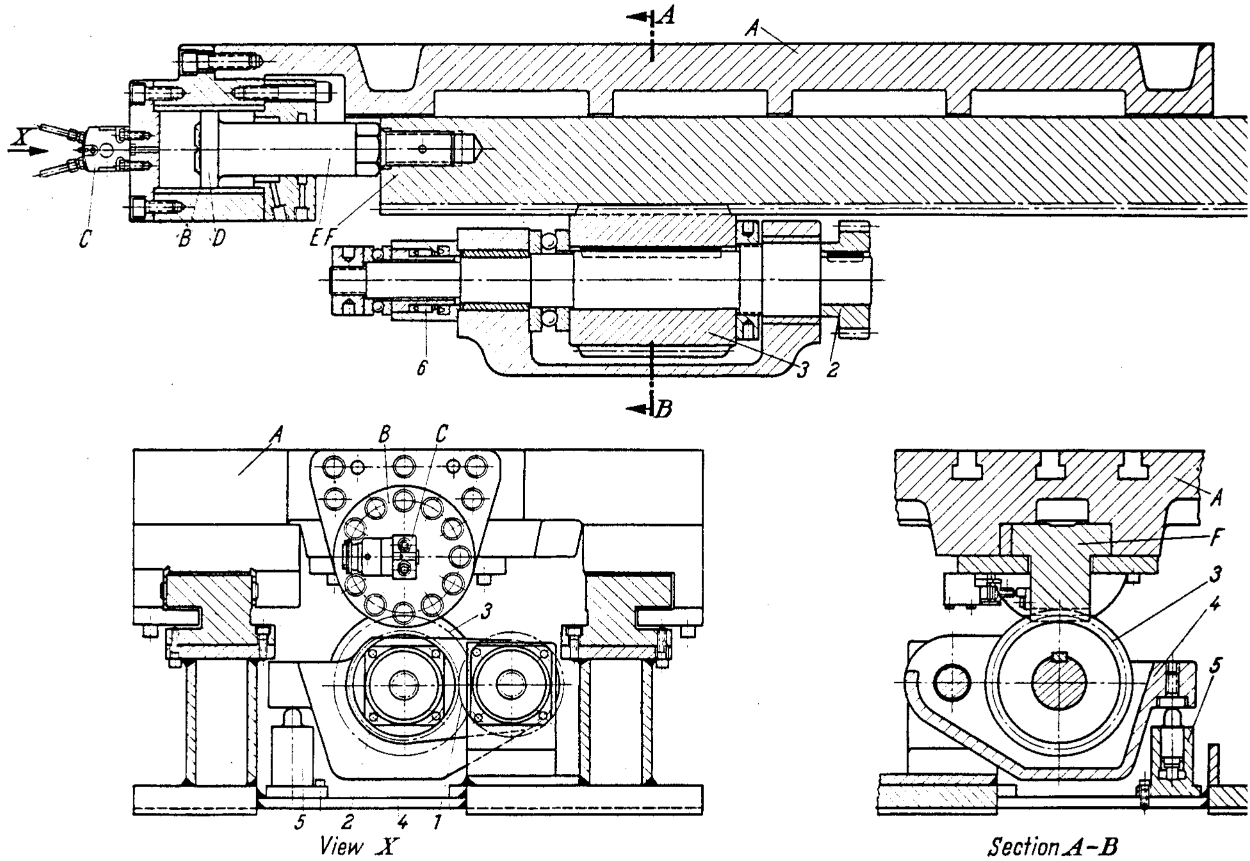


FIG. 458. Milling machine table drive.

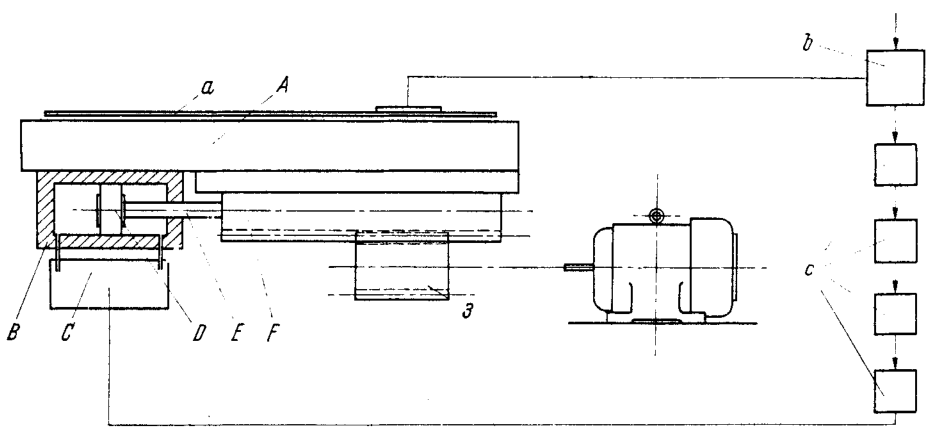


FIG. 459. Schematic sketch of the drive (Fig. 458). *a*—Optical grating fixed to the moving table; *b*—Magnetic tape; *c*—Electronic equipment.

that of milling. However, the finish of the machined surface is less important than the operational speed and for this reason it is desirable for the feed rate to be changed with the gradually varying cutting forces (not force pulsations) and for the machine to work continuously against a maximum feed force component. This means that during the cutting of sections as shown in Fig. 460, the feed rate must vary inversely with the depth of the section which is in front of the saw blade. Figure 461 (Ref. 10, part I) shows a hydraulic feed drive which satisfies this requirement. The circuit corresponds to those shown in Figs. 217 and 219 and the symbols shown for the various valves are also the same. [Note that there are two throttle valves "*e*", one in the incoming (see Fig. 217) and one in the outgoing (see Fig. 219) pipeline].

The maximum hydraulic pressure for the feed drive is adjusted by means of overload valve *f* which is operated by crank handle f_1 and the pressure is read from the pressure gauge *A*. If there is a change in the cross-section which is being cut, the feed rate will correspondingly change, i.e. increase for a decrease in section or vice versa, until the cutting resistance (the force component in the direction of the feed movement), reaches a value equal to the product of the present hydraulic pressure and the piston area. The possible maximum feed rate can be adjusted by means of the lower throttle valve *e* with a hand-wheel e_1. The valve *f* also serves for controlling automatically the pressure and the flow as a function of the oil temperature so that, even with increasing temperature and thus decreasing oil viscosity, the feed rate remains constant. For the rapid withdrawal of the saw blade carrier, the oil is returned to the tank after by-passing the lower throttle valve *e* via the reversing valve *B*.

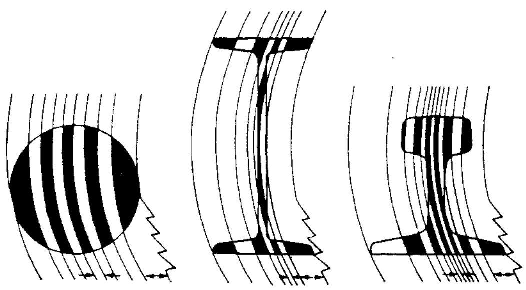


FIG. 460. Feed adjustment during a saw cut to suit the material section in front of the saw blade.

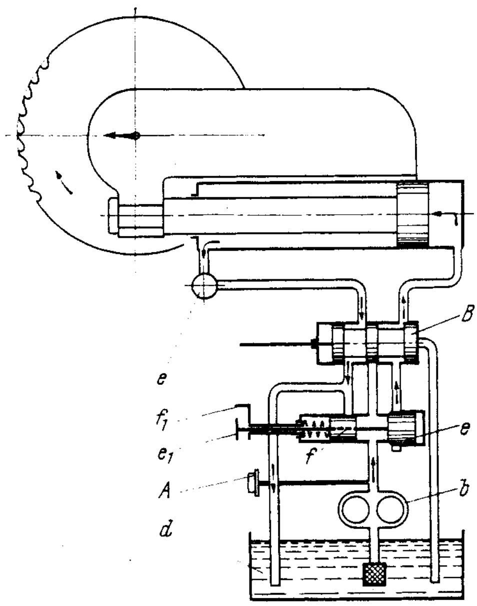


FIG. 461. Schematic lay-out of the hydraulic feed drive for a sawing machine (Gustav Wagner, Reutlingen, Germany).

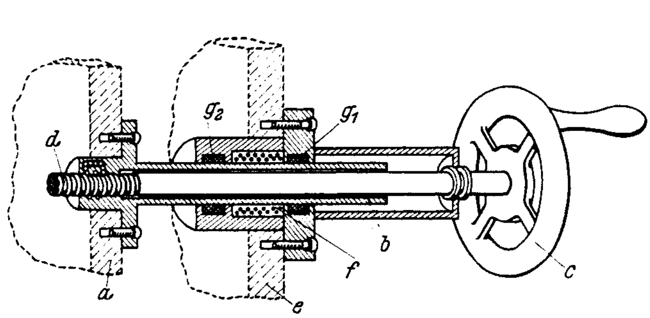


FIG. 462. "Inchworm" device.

Instead of using mechanical or hydraulic feed drives, electrical drives in the form of stepping motors may also be used. The design of such motors is, however, outside the scope of this book.[115]

For intermittent feed movements, for instance those required in grinding, planing and shaping machines, the return movement of the driving shaft is often used for operating a ratchet which in turn rotates a feed screw by a specific amount during each return stroke of the arm or table.

For exceptionally small and accurate setting movements, especially in cases where the danger of "stick-slip" exists, the so-called "inchworm" unit (Fig. 462)[116] has been developed, which is based on the magnetostriction of ferromagnetic materials. The coarse movement of the part *a* which has to be positioned, is carried out by means of hand wheel *c* and feed screw *d*. This screw operates in a recirculating ball nut (see Fig. 203), which is built into a "magnetostrictor" *b*. A solenoid coil *f*, fastened to the machine bed *e*, serves for magnetizing the hollow tube *b*, which can be clamped to the bed by hydraulic bearings g_1 and g_2. A setting operation is carried out as follows. The clamp g_1 on the right is released and that on the left g_2 tightened. The solenoid coil *f* is now energized and as a result the nickel tube *b* contracts lengthwise. Clamp g_1 is now tightened, g_2 released and the solenoid de-energized. The tube *b* immediately expands to its original length and carries the recirculating ball nut, together with the part *a* to the left by the same amount, displacements between 0·00005 and 0·0001 in. being obtainable.

6. CONTROL AND OPERATING DEVICES

The arrangement and layout of operating levers, hand-wheels, etc., must be such as to enable the operator to set and control the machine easily, quickly and reliably. This has already been discussed (see page 37). There are, however, many cases in which a simple direct transmission of control movements by means of levers and shafts does not satisfy one or all of these requirements of easy, rapid and reliable operation. In such cases, either automatic control devices have to be provided or auxiliary equipment (optical, electrical or mechanical) has to be added in the case of manual operation.

The purpose and tasks of such auxiliary equipment are so varied and the design solutions so numerous that it is impossible to discuss more than a small selection of typical examples, which will be mentioned here. These concern the following operations:

(a) Measuring
(b) Speed change
(c) Switching and locking
(d) Positioning and locking
(e) Lubrication.

(a) *Measuring*

Optical instruments which serve for accurately measuring positioning movements have been used for many years and have been described in detail.[117] The main problem in their application lies in the need to fit them in such a manner that they form an organic whole with the machine to which they belong, hence making them integral parts of that machine. This means that their design solutions must be the result of close co-operation between the machine tool designer and the designer of the optical equipment. One of the main difficulties is the reduction to the minimum possible of the influence of temperature changes caused by electric lamps and other heat sources upon the accuracy of the assembly and the obtainable readings. In spite of all precautions taken by the designer the reading and measuring accuracy of the optical instruments is limited by the imperfection of the human eye and the reaction of the human operator. This means that the application of electronics will have a great advantage.

In the measuring system (Fig. 463)[91] of the Lindner jig boring machine, the scale is an axially fixed, but rotating polished cylinder *1*, on to which a helix is marked with high pitch accuracy. The image of the scale is projected via a system of lenses and prisms, on to a screen *2* and passes through the measuring fork *3* of an electro-optical instrument. The fork consists of photo-electric cells which are balanced and thus make the reading on the electrical measuring instrument zero when a line of the scale is exactly in the middle of the measuring fork. As the reading of an electrical instrument is easier than the observation of a scale in an eyepiece, a reading accuracy of about 0·0001 in. can be achieved.

(b) *Speed Change*

Several examples (see for instance Figs. 442, 452, 454, 455 and 456) have been shown where sliding gears or clutch sleeves are shifted by means of forks which are operated by levers, cams, pinions or racks. If, however, the parts which have to be shifted are difficult to get at or are inaccessible, if the operation itself is not easy or it takes a time greater than can be allowed, then hydraulic, mechanical or electrical devices may be used to advantage.

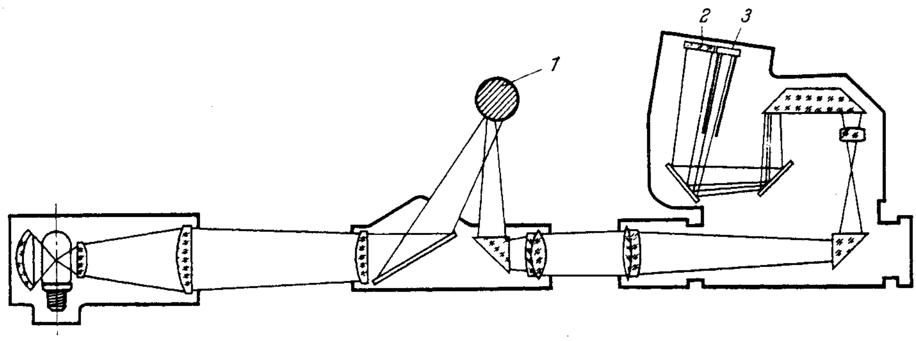


FIG. 463

Figure 464 shows a device for selecting the output speeds of the milling spindle drive gearbox previously shown in Fig. 124. It will be remembered that in this gearbox, 18 spindle speeds can be obtained by means of 3 sliding gear blocks. With the help of the device (Fig. 464), all speeds can be selected by means of a single hand wheel *1*, which replaces three levers and thus avoids the possibility of faulty setting combinations. The hand-wheel *1* drives a cylindrical drum with a cam *5* via a pinion and an internal gear *2*. This is designed in such a manner that during one

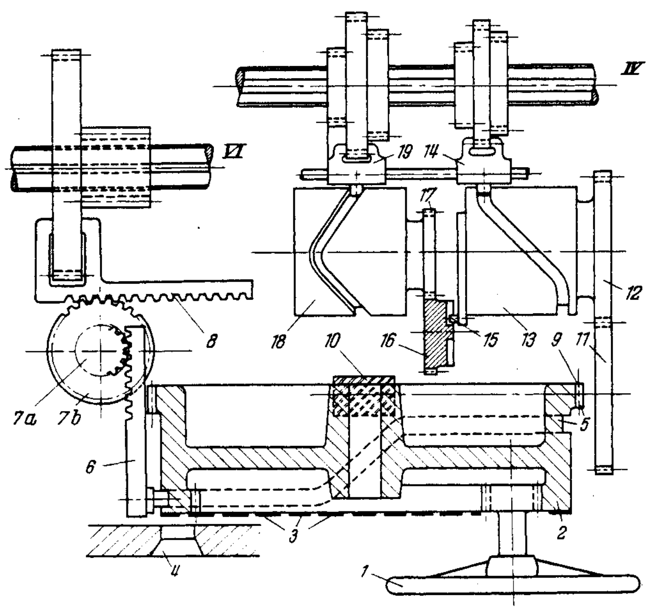


FIG. 464. Speed change device for the drive (Fig. 124); *IV*, *VI*—Shafts; (see Fig. 124).

revolution of the cam cylinder, all 18 output speeds are covered. For each position of the cylinder a figure *3*, corresponding to the set output speed appears in the window *4* of the machine body. In other words, the spindle speed which has been selected by the setting device can be read directly in this window. The cam *5* shifts the rack *8* through a rack *6* and two gears *7a* and *7b*, thus doubling the stroke (the diameter of pinion *7b* is twice that of pinion *7a*). Rack *8* in turn carries the fork which shifts the sliding gear block on shaft *VI* (see Fig. 124). Over one half of the speed range, this gear block must be in one of its extreme positions whilst for the other half it has to be in the extreme opposite position (Fig. 124). The cam cylinder also drives a pinion *11* via helical gears *9* and *10* (transmission ratio 6 : 1), and thus a cam cylinder *13* via gear wheel *12*. The cam *13* shifts the fork *14* for the first sliding gear block on shaft *IV*. It also operates, via a Geneva mechanism *15* and two gears *16/17*, a second cam cylinder *18*, which makes *1/3* turn for each full revolution of the cylinder *13*. The second cam *18* shifts the second sliding gear block on shaft *IV* and moves it by one third of its full stroke (fork *19*) each time the fork *14* has covered its full range.

Speed setting operations can also be motorized as, e.g. in the case of the spindle speed drive (see Fig. 455), whose infinitely variable speed range is produced by a P.I.V. gear (Fig. 465). An

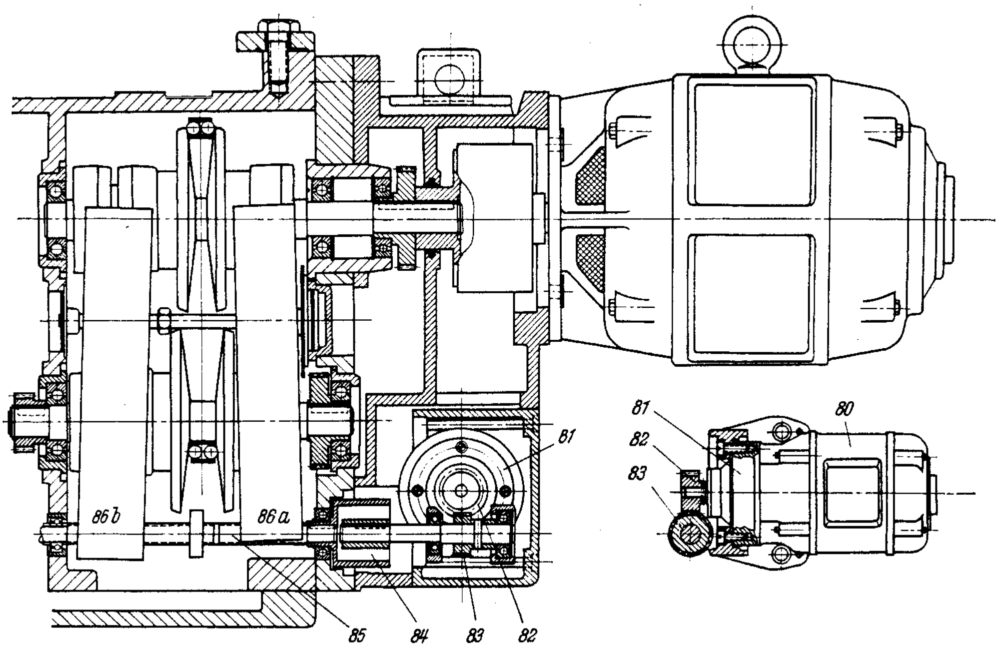


FIG. 465. Motorized speed change for the horizontal boring machine (Fig. 455).

electric motor *80* drives a screw *85* via a reduction gear *81*, helical gears *82/83* and coupling *84*. Screw *85* moves the two levers *86a* and *86b* and thus varies the output speed of the P.I.V. gear (see page 123).

In the Raboma radial drilling machine, hydraulic pre-selection is provided for both the spindle and the feed drive (Fig. 466). The pump *A* is driven from the main motor shaft *I* via gears *B/C* and supplies the pressurized oil to an accumulator *D*. From there, it goes to the pre-selection and setting valves.

The spindle drive (motor shaft *I* to drilling spindle *VII*) consists of a disc clutch operated reversing gear (gears *a/b* or *c/d* with intermediate gear c_1), from which a fixed transmission ratio (*e/f*, *g/h*) drives the sliding gear sets *i/k*, *l/m* or *n/o* and *m/p*, *q/r*, or *k/s* and finally the alternative transmission *t/u*-(*v*–*w*) or *x/y* to the drilling spindle *VII*. The feed drive (shafts V*III* to *XIII*) is derived from the drilling spindle being transmitted via gears *1/2*, sliding gears *3/4*, *5/6* or *7/8* and *9/10*, *11/12* or *8/13*, fixed transmission *14/15*, sliding gears *16/17* or *18/19* and worm drive *20/21*, to drive the pinion shaft *XIII*. Two pre-selector valves (*E* for the spindle speeds and *F* for the feeds) control the gear shifting cylinders *G* and *H* respectively. After the required positions of the pistons in these cylinders have been pre-selected by suitable setting of the pre-selector valves, the lever *I* is rotated In a vertical plane and disengages the clutch on shaft *II* via segment *K* and rack and levers *L*. The eiver *I* can also be rotated in a horizontal plane and then operates the control valve *N* via shaft *M*.

The control valve passes the pressurized oil to the pre-selector valves, whereupon the pistons in the cylinders G and H move the sliding gear blocks into their required positions. If the clutch on shaft II is then re-engaged, the oil supply to the pre-selector valves is cut off by the valve O, so that the valves E and F can now be re-set for the next operation.

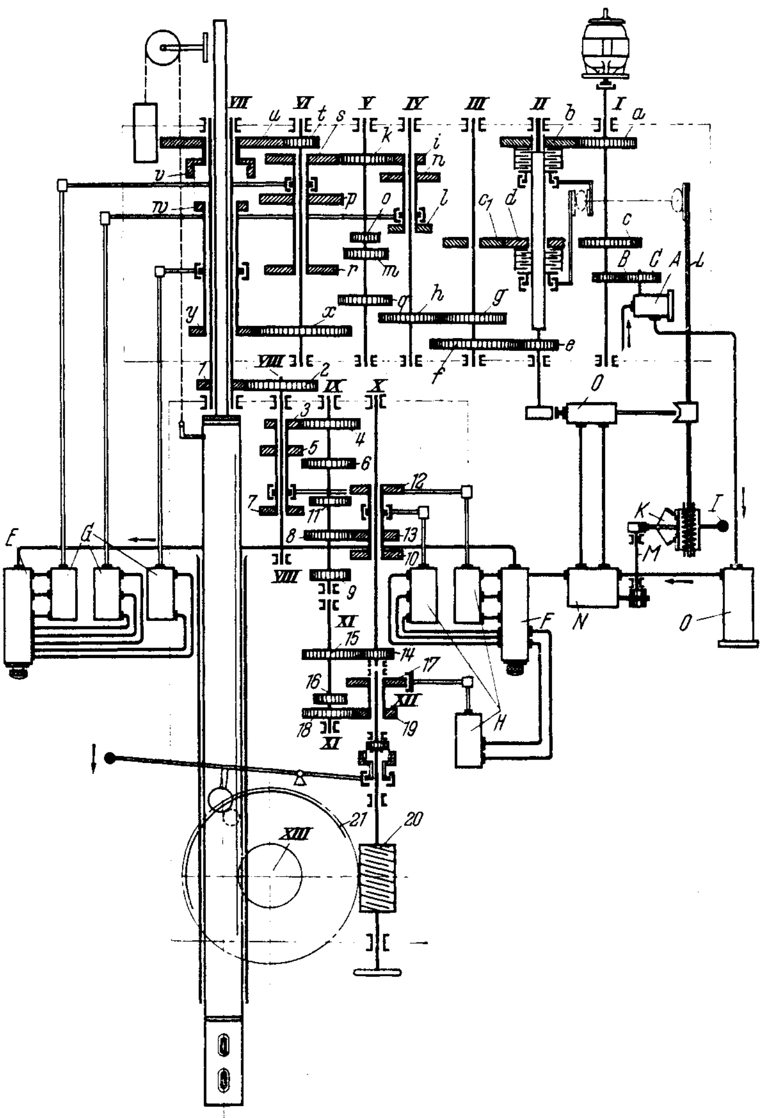


FIG. 466. Schematic diagram of a hydraulic pre-selection device for the gearbox of a radial drilling machine (Raboma, Berlin, Germany).

(c) *Switching and Locking*

The switching and locking of relatively light turret heads by means of Geneva mechanisms or stops has been discussed on pages 170 and 183. The problem becomes more difficult when heavier masses, such as the spindle carrier of a 6-spindle automatic lathe, have to be moved with high speed and at great accuracy. In the design (Fig. 467), a Geneva mechanism is used for rapid and smooth switching, the operating lever which acts directly (i.e. without intermediate gears) on the spindle carrier being given an additional radial movement, thus reducing the impact effect of the operation. The mechanism is driven by gears *1*, *2* and *3* on shafts *I*, *II* and *III*. The gear wheel *3* is keyed to the shaft *III* and carries a radial guide *3a* and a cam *4*. The operating lever *5* of the Geneva mechanism can slide in the radial guide in the direction of arrows *5a* or *5b*. The cam *4* depresses (through roller *6a*) the lever *6*, which moves about its axis *IV*. This results in the index pin *14* being lifted (through pin *7*, rod *8*, pin *9*, bell crank lever *10* on axis *V* against spring *11*, rod *12* and lever *13* on axis *VI*). Index pin *14* slides in a hardened bush *15* and locates the spindle carrier by means of slots *16*.

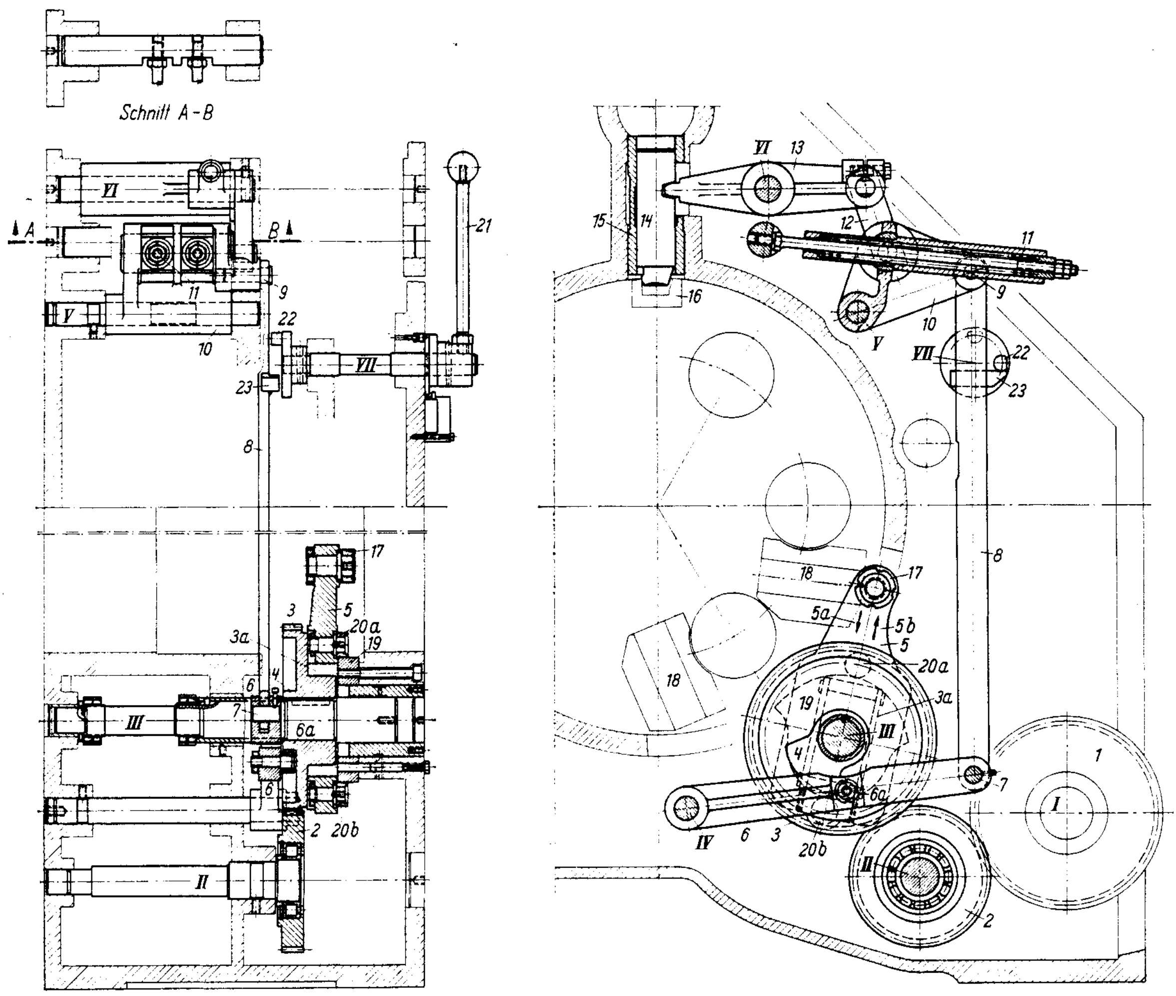


FIG. 467. Switching and locking device for the spindle of a six spindle automatic lathe (Alfred H. Schütte, Köln, Germany).

As soon as the lever is lifted out of the slot, the spindle carrier can rotate freely. At that moment, the lever *5* with roller *17* enters one of the slots *18* in the Geneva wheel on the spindle carrying drum and through the simultaneous radial movement of lever *5* controlled by the fixed cam *19* and rollers *20a* and *20b*, a smooth switching operation is achieved. The whole operation is usually part of the automatic cycle of the machine, but for setting purposes it is possible to withdraw the index pin by means of lever *21*, which presses the operating rod *8* downwards (shaft *VII*, crank pin *22* and stop *23*), so that the spindle carrier can be rotated by hand.

(d) *Positioning and Locking*

It is usually advantageous to arrange all operating levers and hand-wheels in such a manner that the operator can reach and move them comfortably from one position. In addition, if several operations, e.g. the clamping of various parts, can be carried out by moving only one lever, valuable time can be saved. However, there are many cases in which the possibility of executing such operations *either* simultaneously *or* independently, especially when setting the machine, may be an advantage.

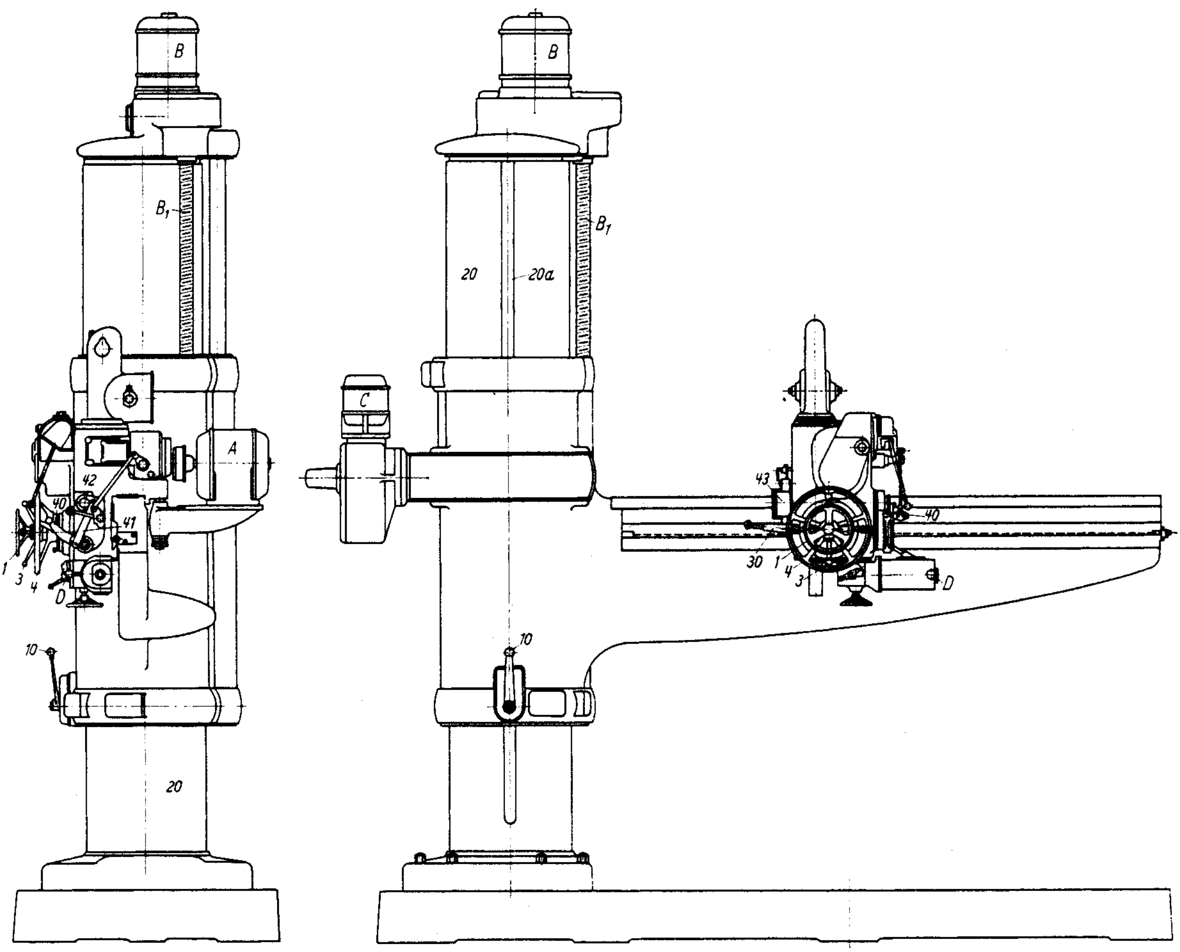


FIG. 468. Radial drilling machine (Wm. Asquith Ltd., Halifax).

The design of the radial drilling machine (Fig. 468) is an example of the application of such an idea. The machine is equipped with three motors:

(A) for the spindle drive,

(B) for raising and lowering the radial arm (screw B_1), and

(C) for clamping the radial arm on the column.

Lever D serves for starting and stopping the motors A and B. Three hand-wheels or levers respectively are arranged on concentric shafts (axis I) on the radial arm (Fig. 469). They are:

(i) the hand-wheel *1*, shaft I and chain wheel *2* for moving the drilling head along the radial arm;

(ii) the lever *3* for the feed drive clutch;

(iii) the hand-wheel *4* for rapid feed movements, together with the depth setting (trip dog 5, Fig. 469).

The radial arm and the drilling head can be clamped in the following alternative manners:

(i) Radial arm lightly clamped (lever *10*):

(a) against rotation around the column (lever *10* to the left),

(b) against rotation around the column as in (a) and against vertical movement along the column (lever *10* to the right).

(ii) Spindle head clamped against movement along the radial arm (lever *30*).

(iii) Radial arm clamped to the column against rotation and vertical movement and spindle head clamped to the radial arm, all by one single operation (lever *40*).

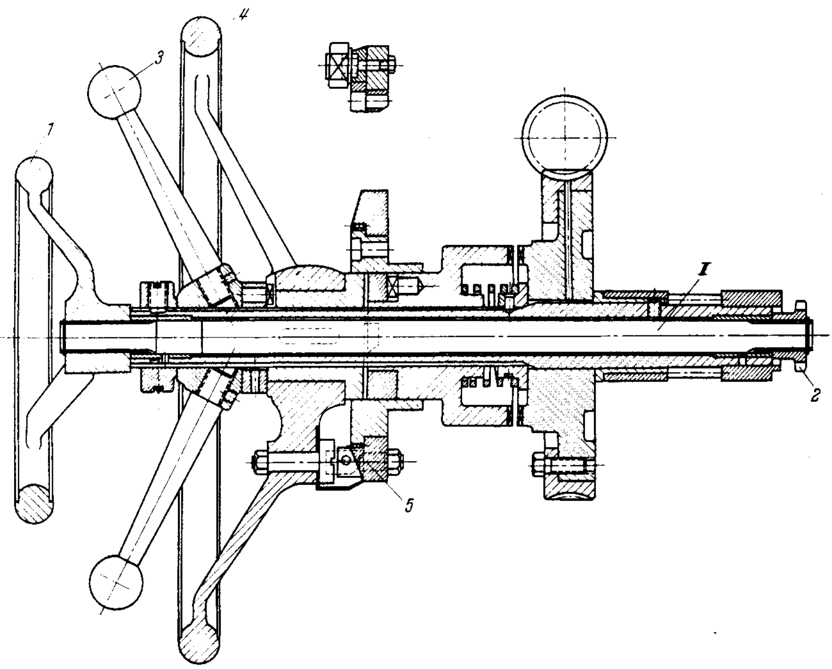


FIG. 469. Control levers for the machine (Fig. 468).

Clamping operations (i) and (ii) are useful when setting and positioning the machine, whilst clamping operation (iii) is carried out immediately prior to drilling and can be completed without the operator having to leave his position.

Operation (i) (Fig. 470a) is carried out as follows.

(a) The neutral position of lever *10*, when everything is unclamped, is determined by index *11*. If the lever is moved, it turns the hollow shaft *12* with the eccentric *13* and thus lifts the clamping strip *15* via connecting rod *14*. Through this movement a tube *16* is pressed against a shoulder *17* of the radial arm sleeve *18*. The tube *16* on which the unclamped radial arm sleeve can rotate freely, is itself secured against rotation on the column by the key *19*, which fits into the keyway *20a* of the column *20* (see Fig. 468). When, therefore, the clamping strip *15* is tightened, the radial arm sleeve and the tube *16* can be moved axially along the column but cannot rotate around it.

(b) When the lever *10* is moved to the right, the hollow shaft *12* together with setting screw *21* and pin *22* is moved axially to the right through screw *23* in nut *24*. As a result, the locking piece *25* is pressed against the column *20* and the radial arm is locked by friction against both rotation and axial movement on the column. At the same time, the cam *26* pushes rod *27* upwards operating an electric switch *28* which interrupts the current supply to motor *B* (see Fig. 468) and thus prevents the drive of the vertical screw being switched on.

(ii) (Fig. 470b). If the index pin *30a* is withdrawn and lever *30* pushed down the wedge piece *33* is drawn to the right via shaft *31* and screw thread *32*. This clamps the spindle head between the faces *36* of the arm and the angle piece *34*. The spindle head is thus pushed upwards (i.e. in the direction of the axial load on the drill) against the slideway on the radial arm *38*.

(iii) (Fig. 470b) and (Fig. 470c). This operation can be carried out by means of lever *40* (see

Fig. 468) or lever *30* (with the index pin *30a* engaged). It is, however, preferable to use lever *40* as it works via a toggle linkage *41* (Fig. 468), and consequently exerts a higher clamping pressure. The rod *41* rotates shaft *42* in the switch box *43*, thus carrying out the clamping operation as follows:

(a) The spindle head is clamped to the radial arm (as before) by moving lever *30* via gears *44*/*45*, rack *46*, linkage *47* and index *30a*.

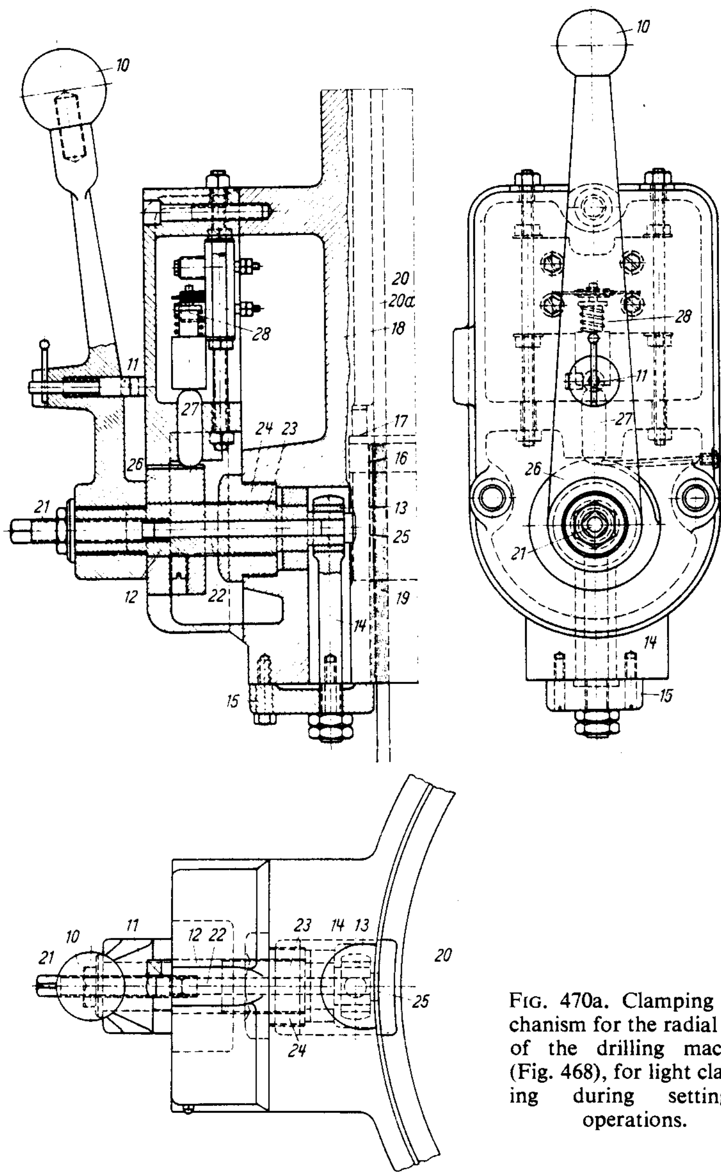


FIG. 470a. Clamping mechanism for the radial arm of the drilling machine (Fig. 468), for light clamping during setting-up operations.

(*b*) The power operated clamping of the radial arm sleeve on the column is initiated via cam *48*, push rod *49* and starter *50* for motor *C* (see Fig. 468). The motor *C* (Fig. 470c) drives worm gear *51*/*52* and through it the disc *53* with the crank pin *54*, the connecting rod *55*, crank lever *56*, hollow shaft *57*, sliding clutch *58* and shaft *59*. This shaft, with its two screw threads, operates the clamping jaws *60a* and *60b* symmetrically against the column *20*. As soon as the required clamping torque is reached at the motor shaft, an overload circuit breaker switches the motor off. The disc 53 also carries a cam *61*, which interrupts the current supply for motor *B* via switch *62*, so that the vertical screw B_1 cannot be operated once the radial arm is clamped to the column.

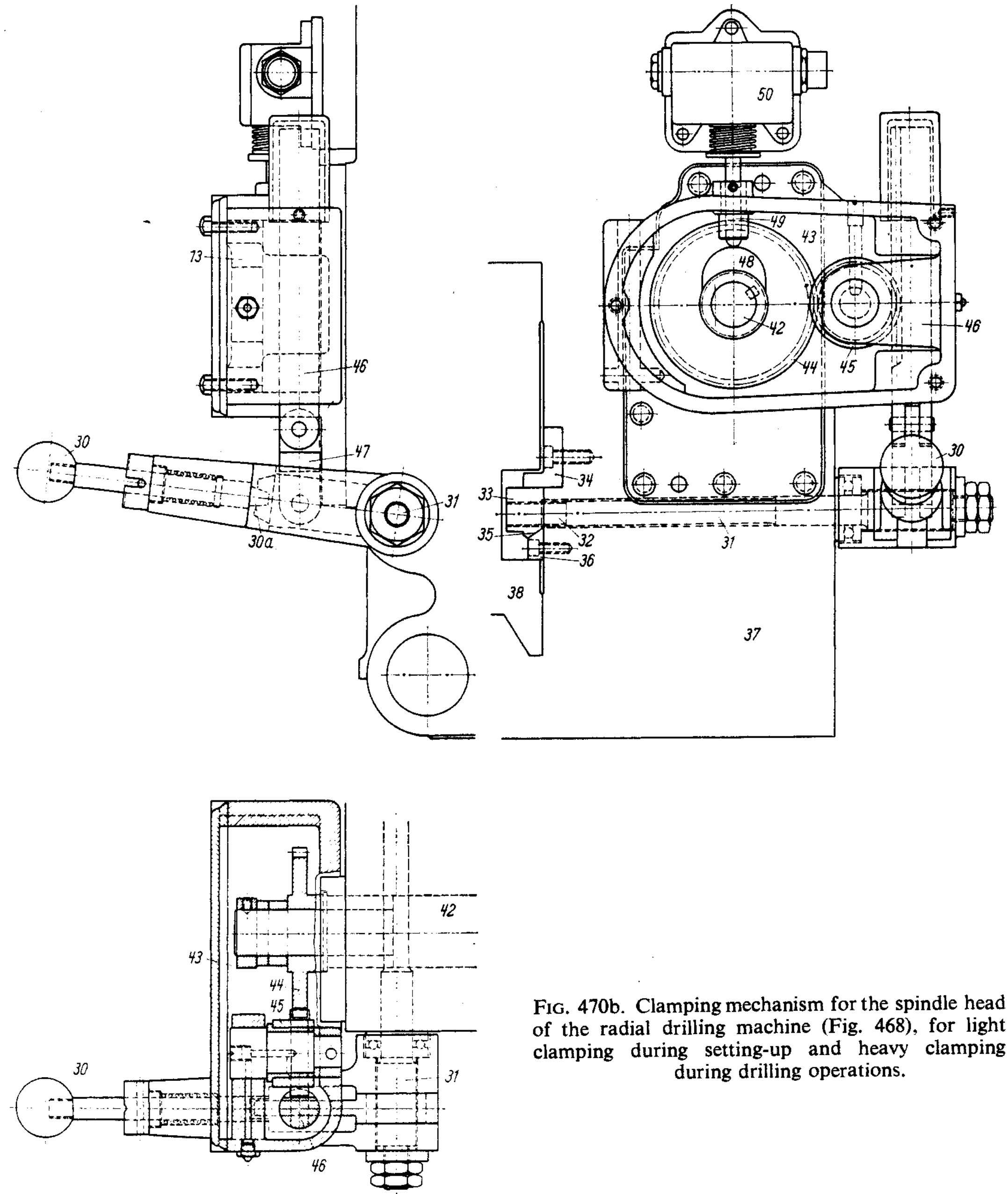


FIG. 470b. Clamping mechanism for the spindle head of the radial drilling machine (Fig. 468), for light clamping during setting-up and heavy clamping during drilling operations.

For unclamping motor *C* is reversed and as soon as the radial arm is unclamped, motor *C* is switched off by the overload which is produced when the brake *65* is operated by cam *63* and lever *64*.

(e) *Lubrication*

In spite of the great attention which is at present given to the introduction of more or less automatic lubrication for bearings and slideways, especially under hydrostatic pressure, the designer must consider carefully the less sophisticated methods of lubricating moving surfaces. Under no circumstances must this be left to chance or to the discretion of the machine operator. It is not sufficient to mention all lubrication points in an "operator's handbook", because it happens frequently that one or more of these points are overlooked or neglected and the handbook is often

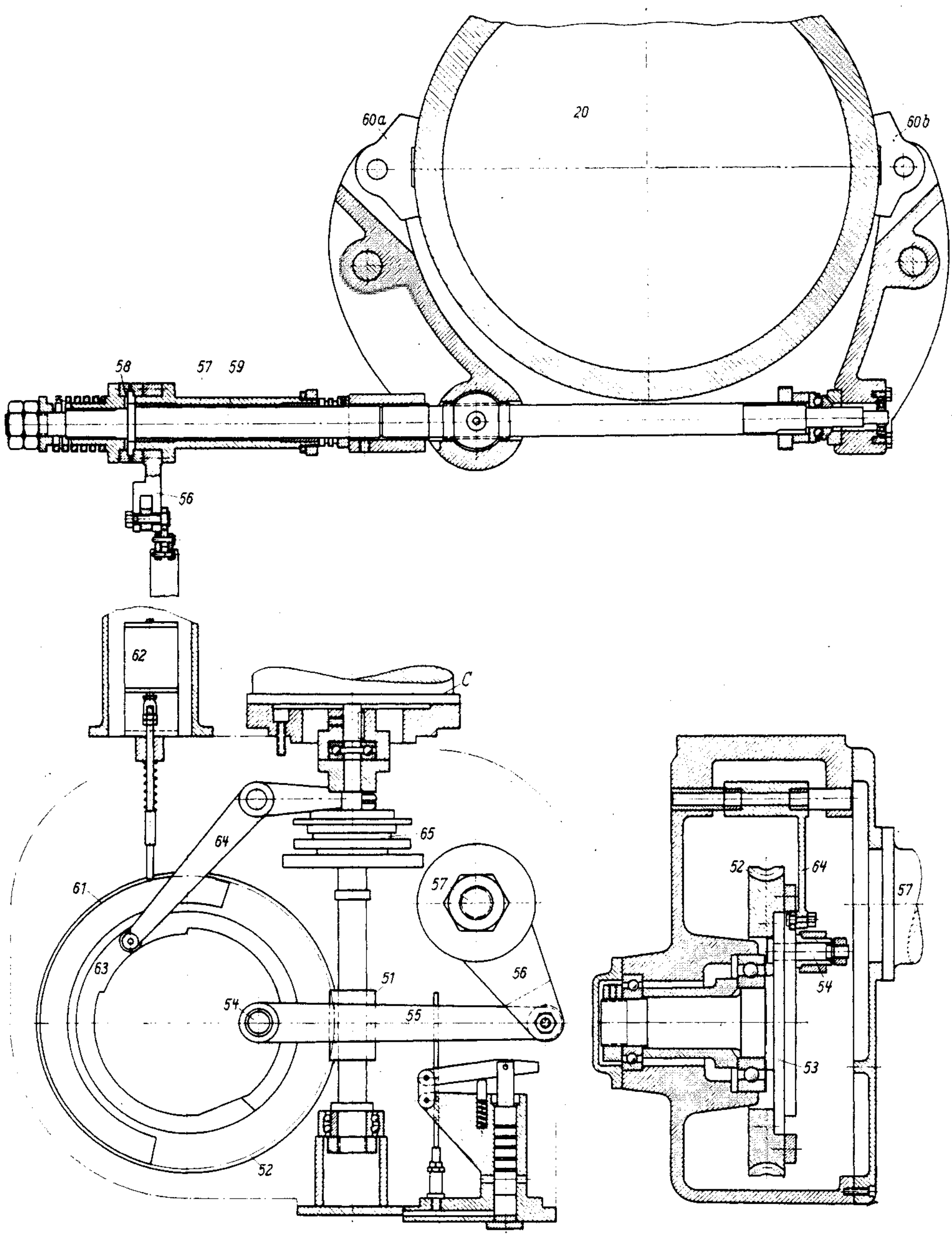


FIG. 470c. Motorized clamping of the radial arm on the column of the radial drilling machine (Fig. 468).

mislaid. It is far better to provide a central point from which all surfaces in question can be lubricated by a single action. Figure 471 shows the saddle of a horizontal boring machine, which is equipped with a hand pump *A* from which all essential parts are lubricated. The oil level in the container (filler *1*) can be checked through the window *2*. The pump *A* is operated by handle *3* and pumps the oil through pipeline *4* to distributor *5*, from where oil is supplied to all those surfaces which

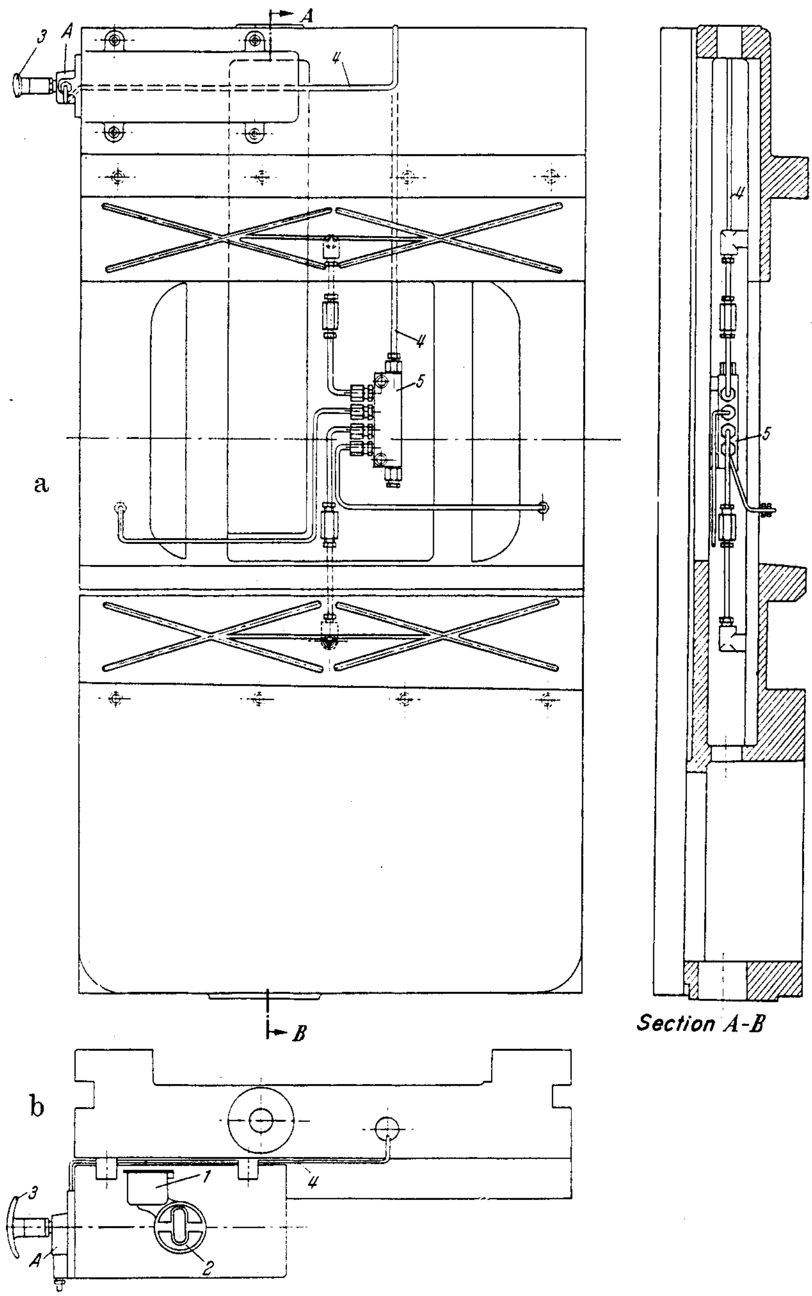


FIG. 471a and b. Centralized lubrication of the slideways for a horizontal boring machine (H. W. Kearns and Co. Ltd., Broadheath).

require lubrication. The appropriate distribution of the oil over these surfaces is ensured by the design of suitable oil grooves (see Fig. 384).

In the case described above, the lubrication still depends upon the reliability of the operator who has to make use of the lubricating pump. In order to become independent of the human element the designer of the milling machine knee (Fig. 472) has provided a pump *A* which supplies the oil

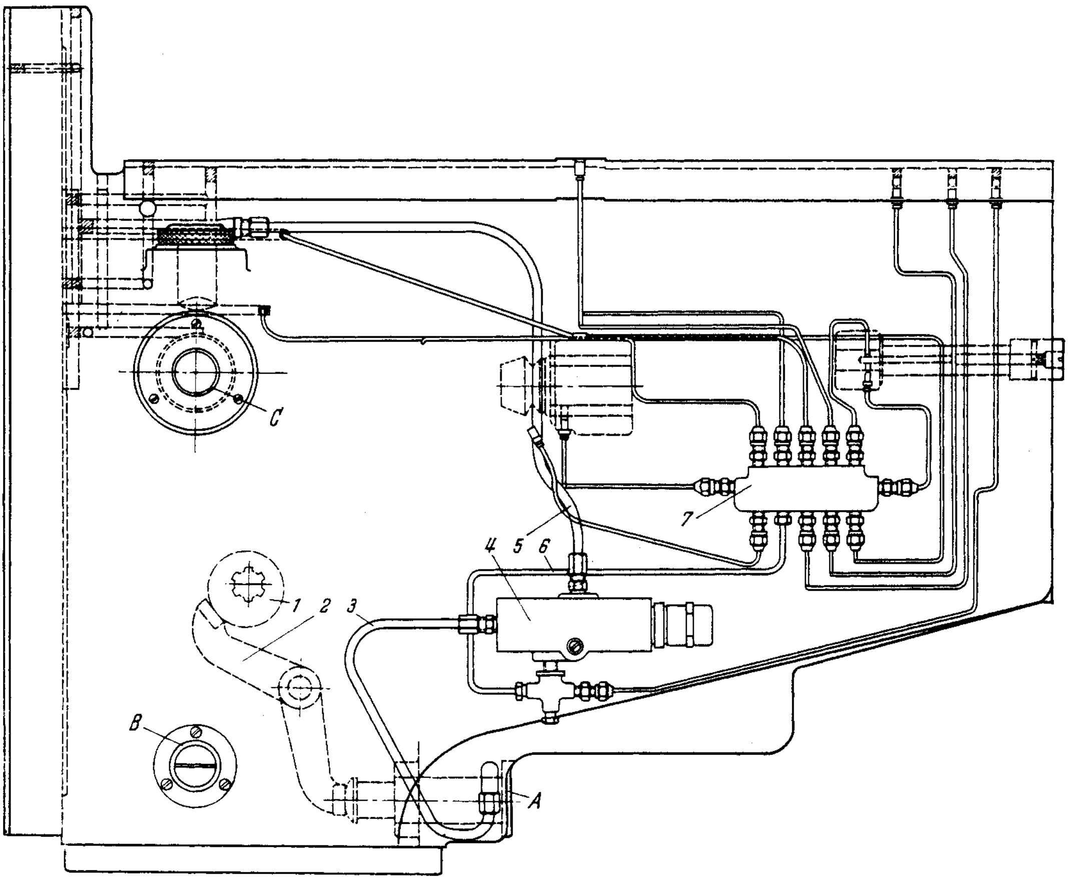


FIG. 472. Pump lubrication in the knee of a milling machine (Brown & Sharpe Mfg. Co., Providence, U.S.A.).

as soon as the feed motor in the knee is switched on. The knee itself serves as the oil reservoir, the oil level being checked through the window *B*. The pump *A* is driven by an eccentric *1* via lever *2* and the oil passes through pipeline *3* to valve *4*. This consists of a spring loaded piston which is displaced by the oil pressure in such a manner that it allows the oil to pass via pipeline *5* to the top of the knee from where it floods the gearbox, becoming visible in window *C*, for checking purposes, on its return. As soon as the feed drive is disengaged and the pump *A* thus stopped, the spring in valve *4* slowly returns the piston to its original position, thus closing the exit port to pipe *5* and opening another one to pipeline *6*. This pipeline supplies a pre-determined amount of oil to the distributor *7* and from there, to eleven different shafts, bearings, sliding surfaces and control devices in the knee. During each engagement and disengagement of the feed drive, this operation is repeated and by the resulting alternate supply of oil during and after feed movements, it is possible to distribute the required quantities of oil in a sequence which is most suitable for the prevailing working conditions.

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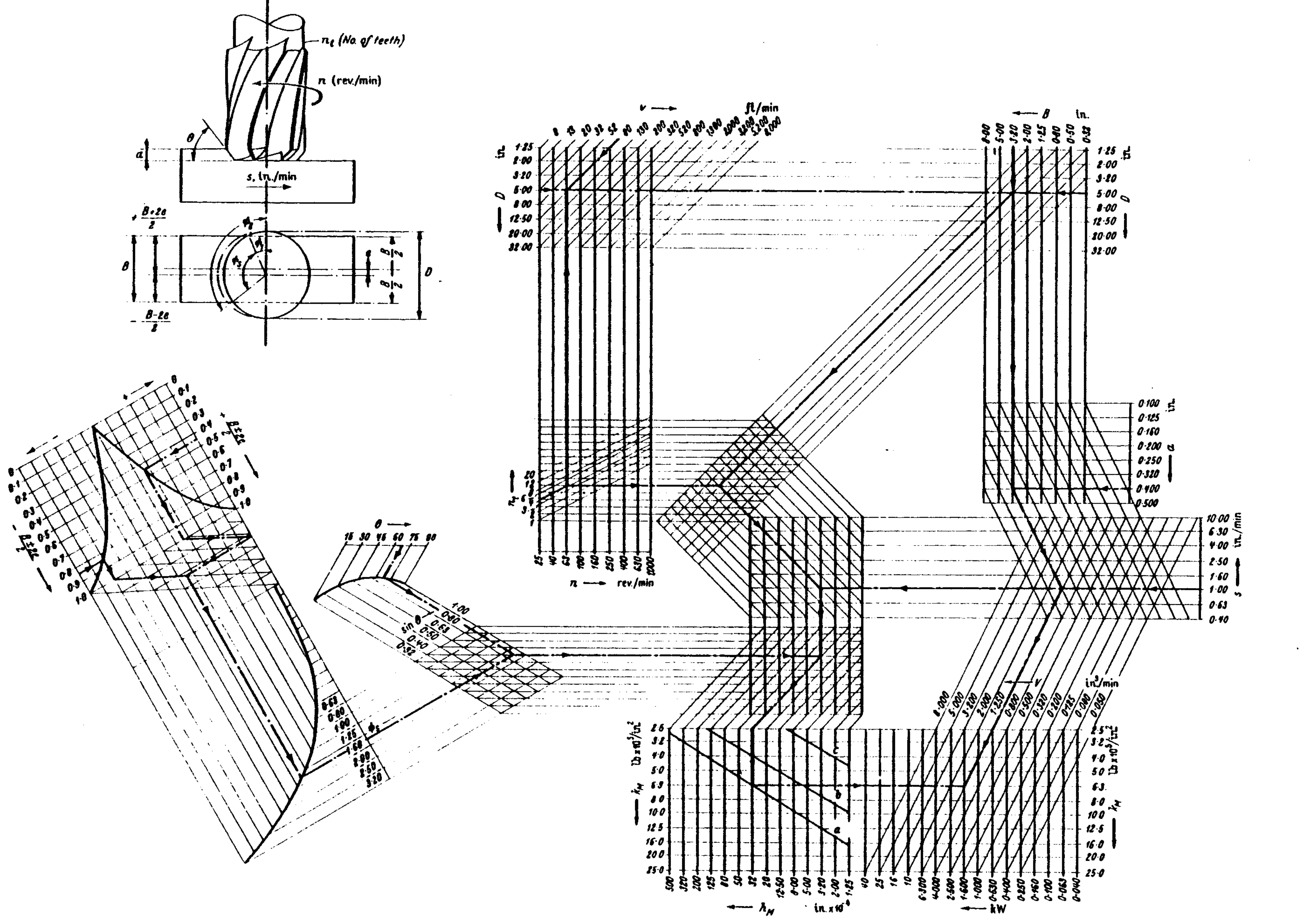
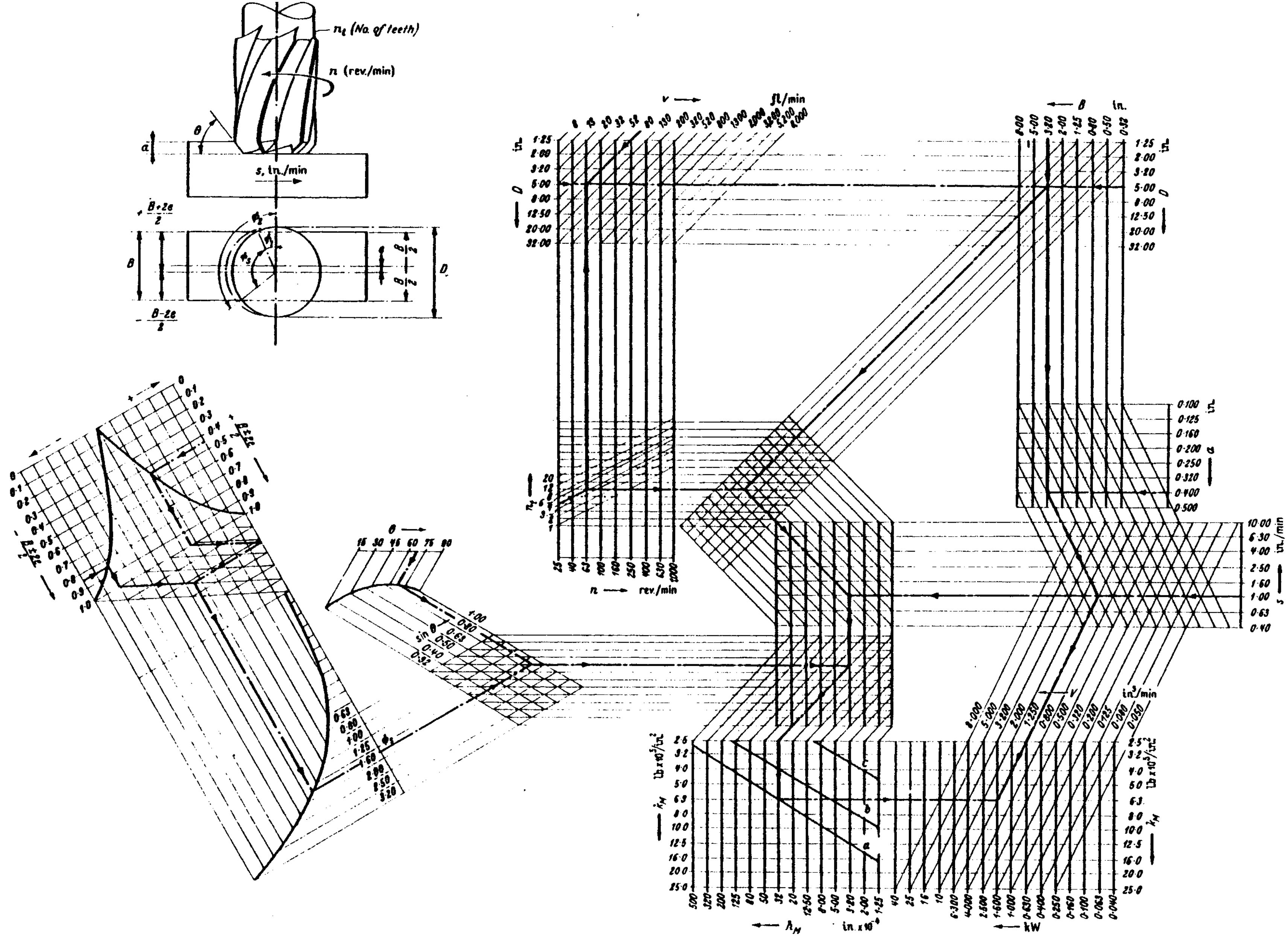


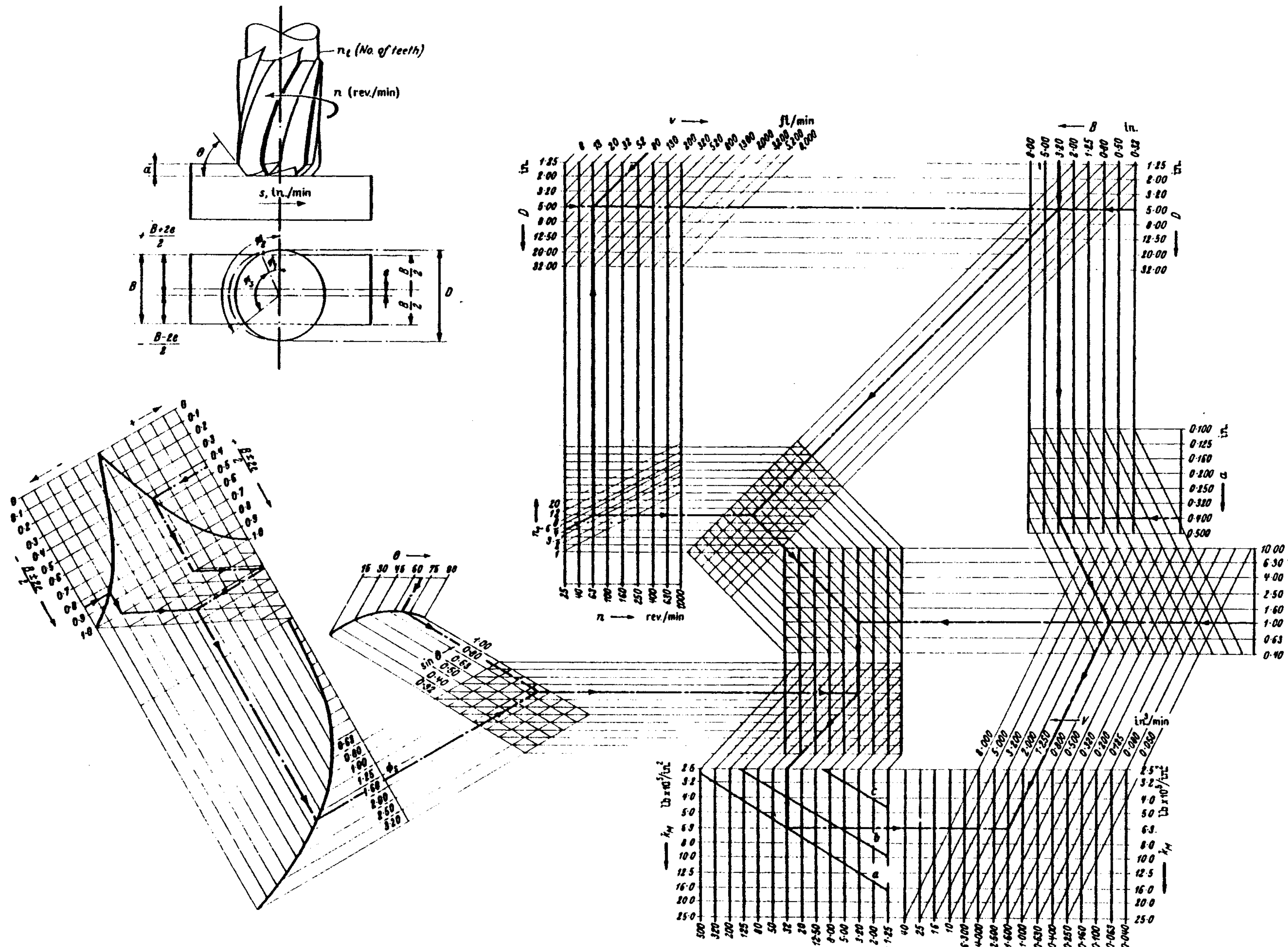
n_t (No. of teeth)
n (rev./min)
s, in./min
v ft/min
B in.
D in.
n rev./min
a in.
s in./min
in³/min
kW
lb x 10³/in²
in. x 10⁻⁴
sin θ
θ
ϕ_s

FIG. 28a

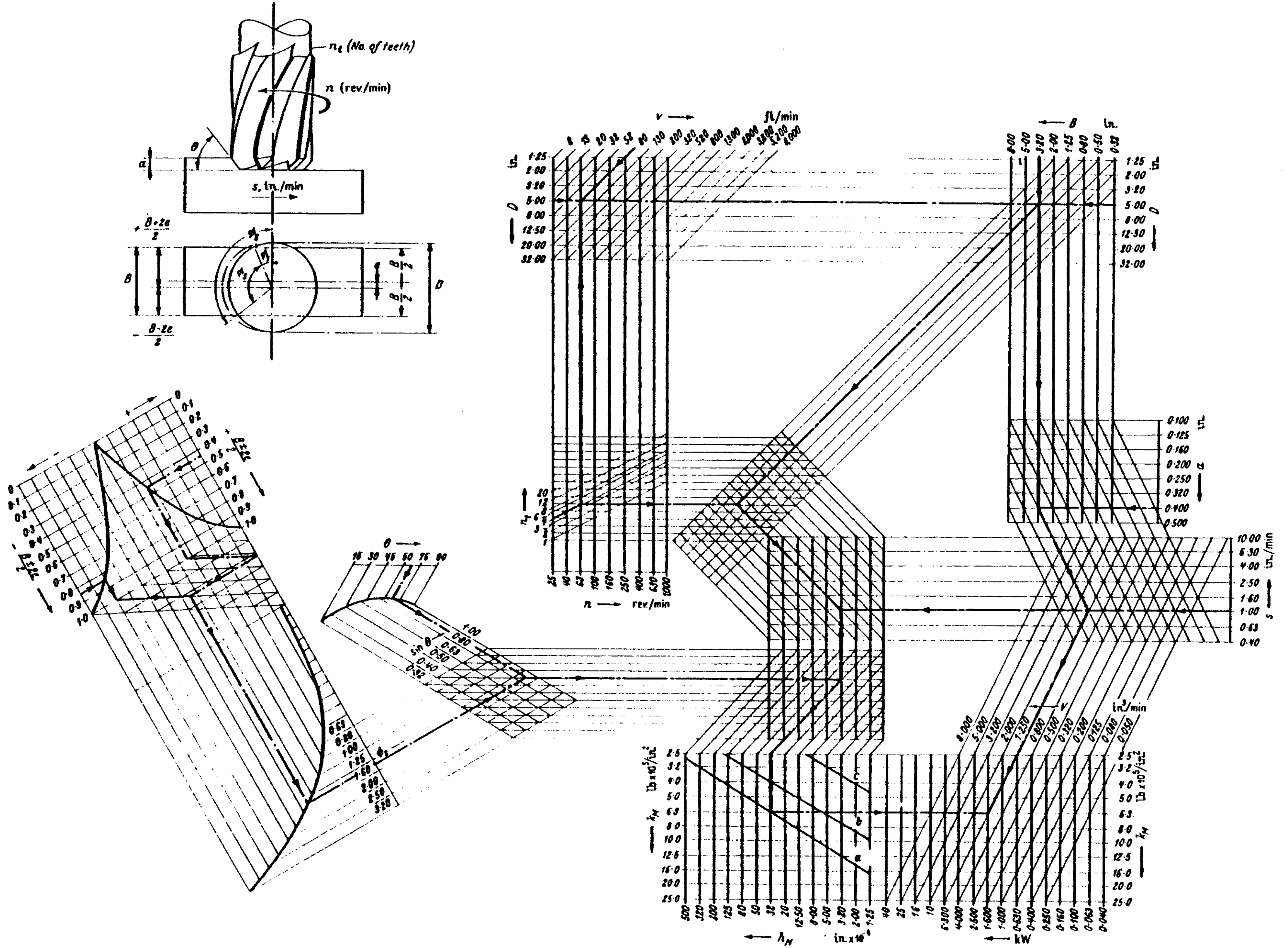


n_t (No. of teeth)
n (rev./min)
s, in./min
n rev./min
v ft/min
D in.
B in.
a in.
s in./min
v in.³/min
θ
sin θ
ϕ_s
k_M lb x 10³/in.²
h_M in. x 10⁻⁴
kW

FIG. 28a



n_t (No. of teeth)
n (rev./min)
s, in./min
v ft/min
n rev./min
D in.
B in.
a in.
s in./min
θ
sin θ
ϕ_s
h_M in. x 10^{-4}
k_M lb x 10^3/in.^{2}
kW
V in.^{3}/min



n_t (No. of teeth)
n (rev./min)
s, in./min
θ
a
B
D
$\frac{B+2e}{2}$
$\frac{B-2e}{2}$
$\frac{B}{2}$
v ft/min
D in.
B in.
a in.
s in./min
n_t
n rev/min
θ
sin θ
ϕ_s
h_M in. x 10^{-4}
k_M lb x 10^5/in.^{2}
kW
V in.^{3}/min

FIG. 28a

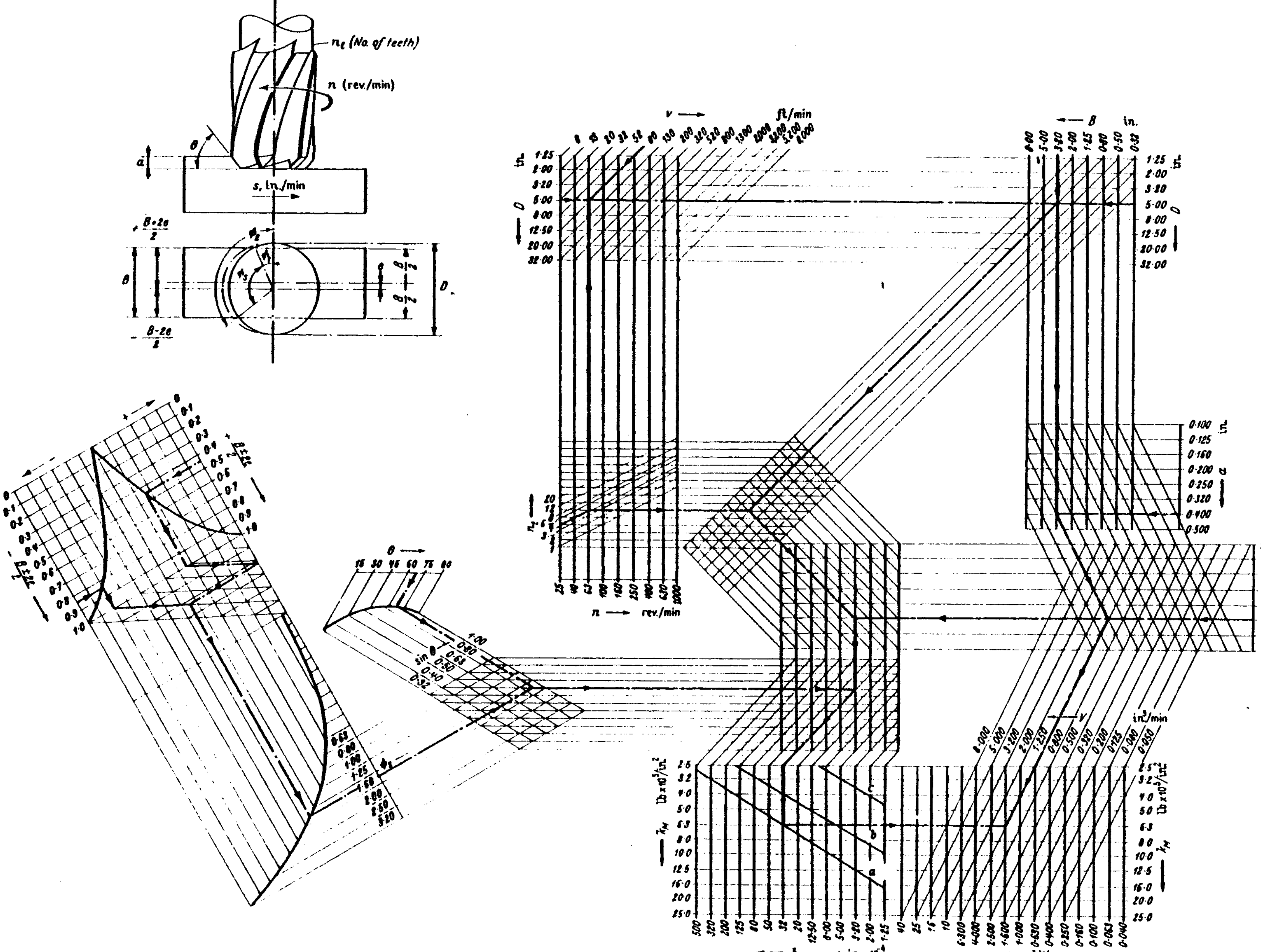

n_t (No. of teeth)
n (rev./min)
s, in./min
v ft/min
B in.
D in.
n rev./min
a in.
s in./min
in³/min
kW
λ_M lb x 10⁵/in²
h_M in. x 10⁻⁴
sin θ

FIG. 28a

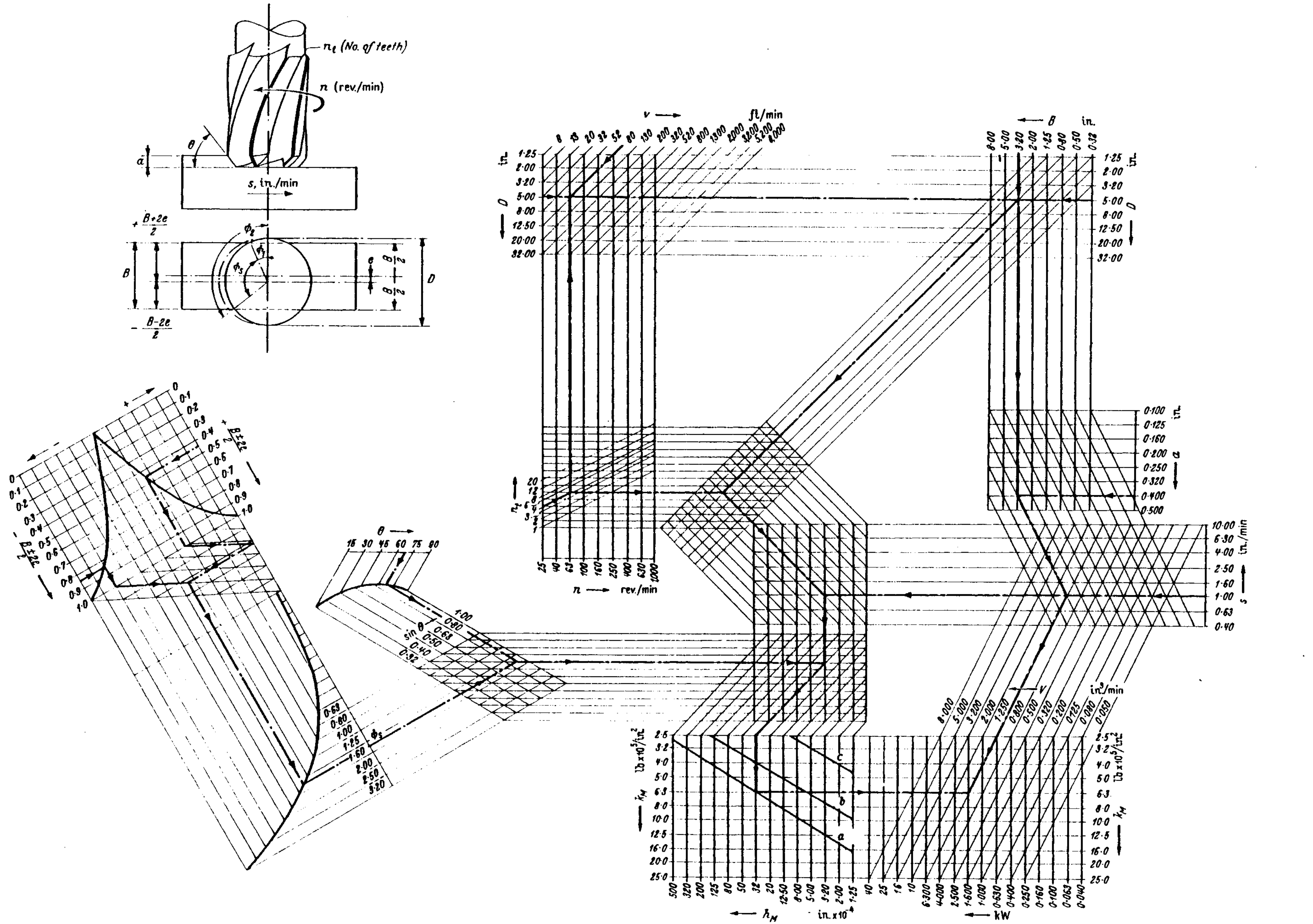
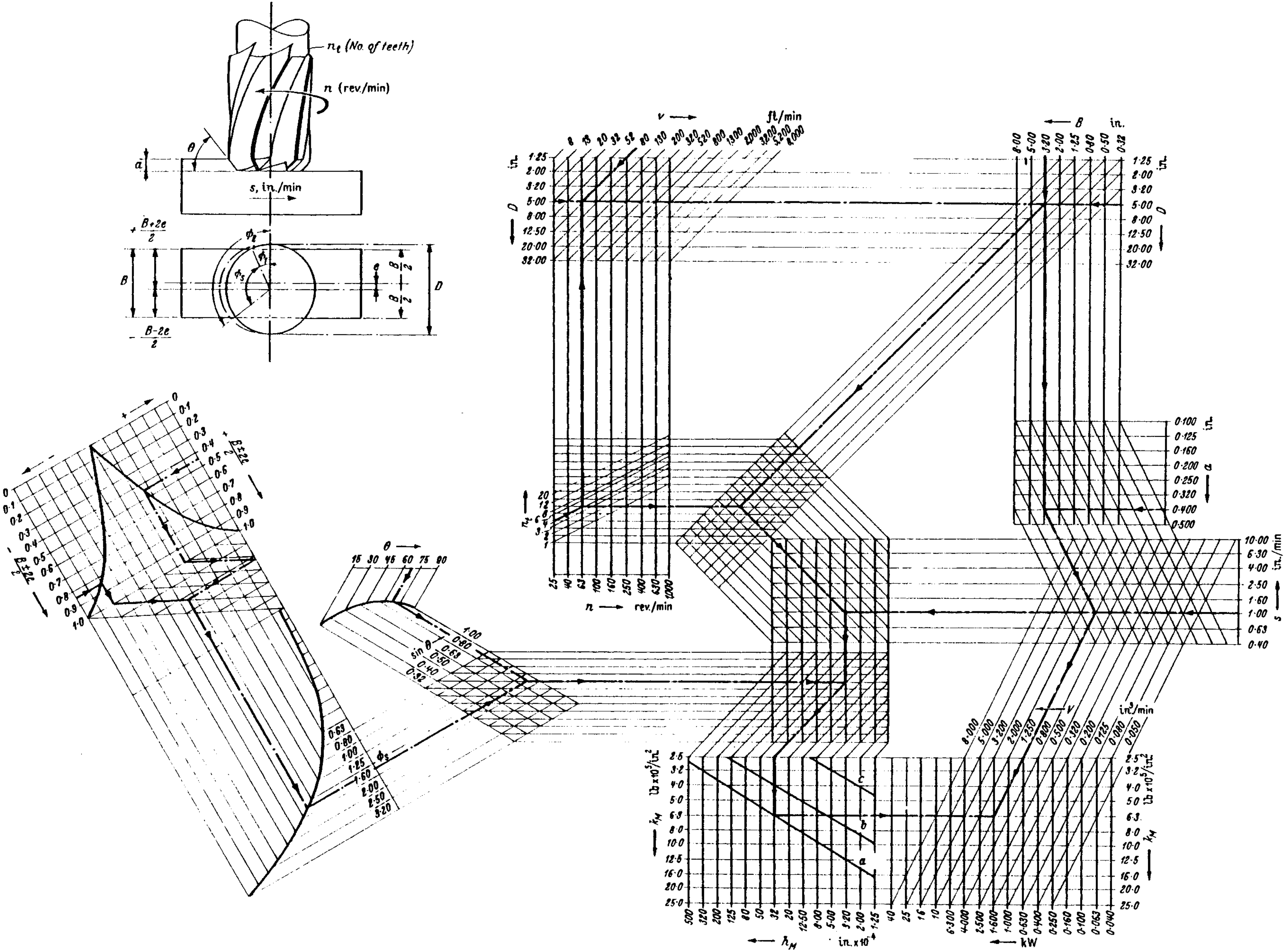
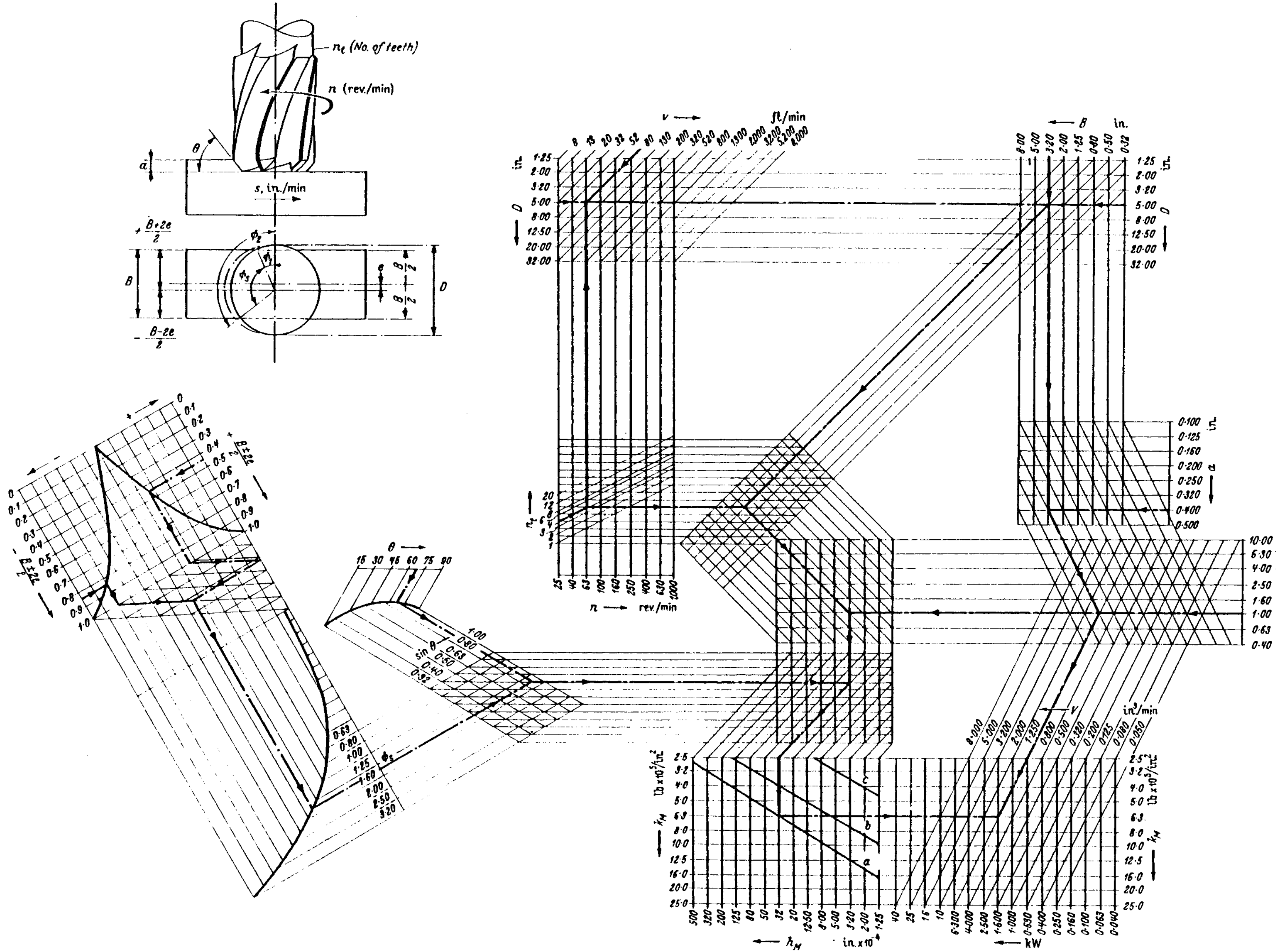

n_t (No. of teeth)
n (rev./min)
s, in./min
v ft/min
D in.
B in.
a in.
s in./min
n rev./min
n_t
θ
$\sin\theta$
Φ_s
V in.³/min
k_M lb x 10⁵/in.²
h_M in. x 10⁻⁴
kW

FIG. 28a



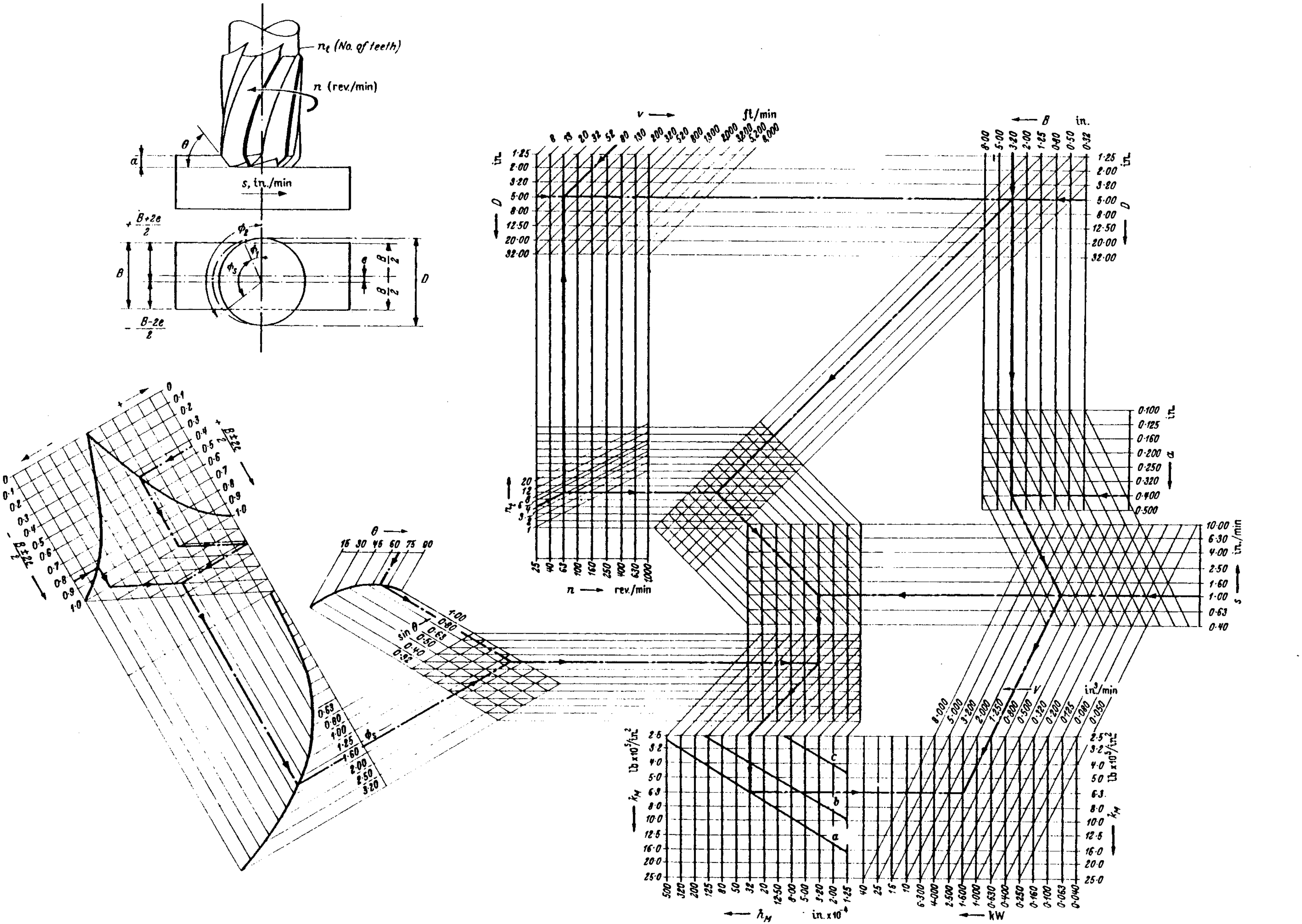
n_t (No. of teeth)
n (rev./min)
s, in./min
v ft/min
n rev./min
B in.
D in.
a in.
s in./min
in.^{3}/min
k_M lb x 10^5/in.^{2}
h_M in. x 10^{-4}
kW
sin θ
ϕ_s

FIG. 28a



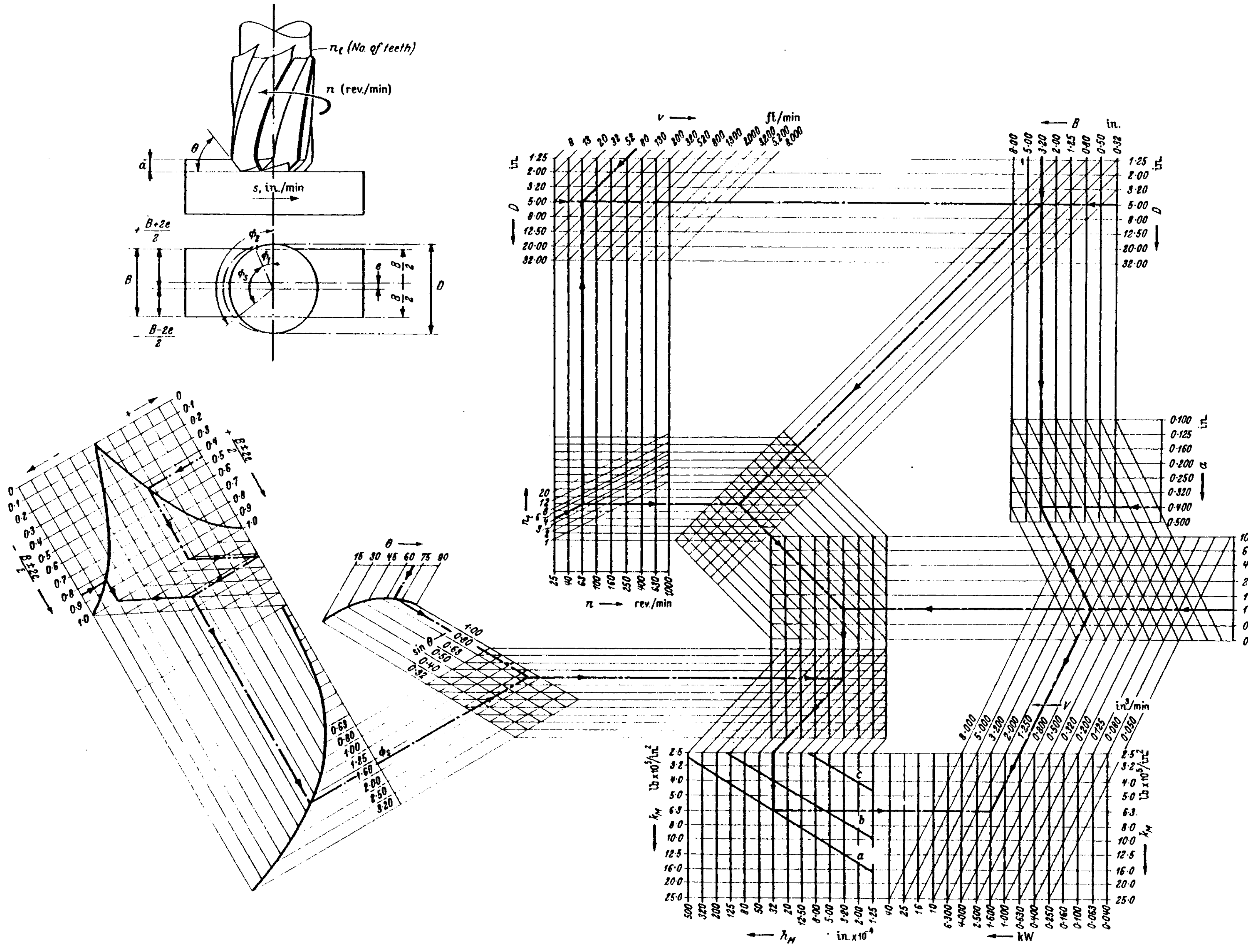
n_t (No. of teeth)
n (rev./min)
s, in./min
θ
a
B
D
$\frac{B+2e}{2}$
$\frac{B-2e}{2}$
$\frac{B}{2}$
e
ϕ_s
ϕ_2
v ft/min
8 13 20 32 52 80 130 200 320 520 800 1300 2000 3200 5200 8000
D in.
1·25 2·00 3·20 5·00 8·00 12·50 20·00 32·00
B in.
8·00 5·00 3·20 2·00 1·25 0·80 0·50 0·32
a in.
0·100 0·125 0·160 0·200 0·250 0·320 0·400 0·500
s in./min
10·00 6·30 4·00 2·50 1·60 1·00 0·63 0·40
n rev./min
25 40 63 100 160 250 400 630 1000
n_t
1 2 3 4 6 8 12 20
θ
15 30 45 60 75 90
sin θ
1·00 0·80 0·63 0·50 0·40 0·32
0·63 0·80 1·00 1·25 1·60 2·00 2·50 3·20
V in.³/min
8·000 5·000 3·200 2·000 1·250 0·800 0·500 0·320 0·200 0·125 0·080 0·050
k_M lb x 10⁵/in.²
2·5 3·2 4·0 5·0 6·3 8·0 10·0 12·5 16·0 20·0 25·0
h_M in. x 10⁻⁴
500 320 200 125 80 50 32 20 12·50 8·00 5·00 3·20 2·00 1·25
kW
40 25 16 10 6·300 4·000 2·500 1·600 1·000 0·630 0·400 0·250 0·160 0·100 0·063 0·040
a b c

Fig. 28a



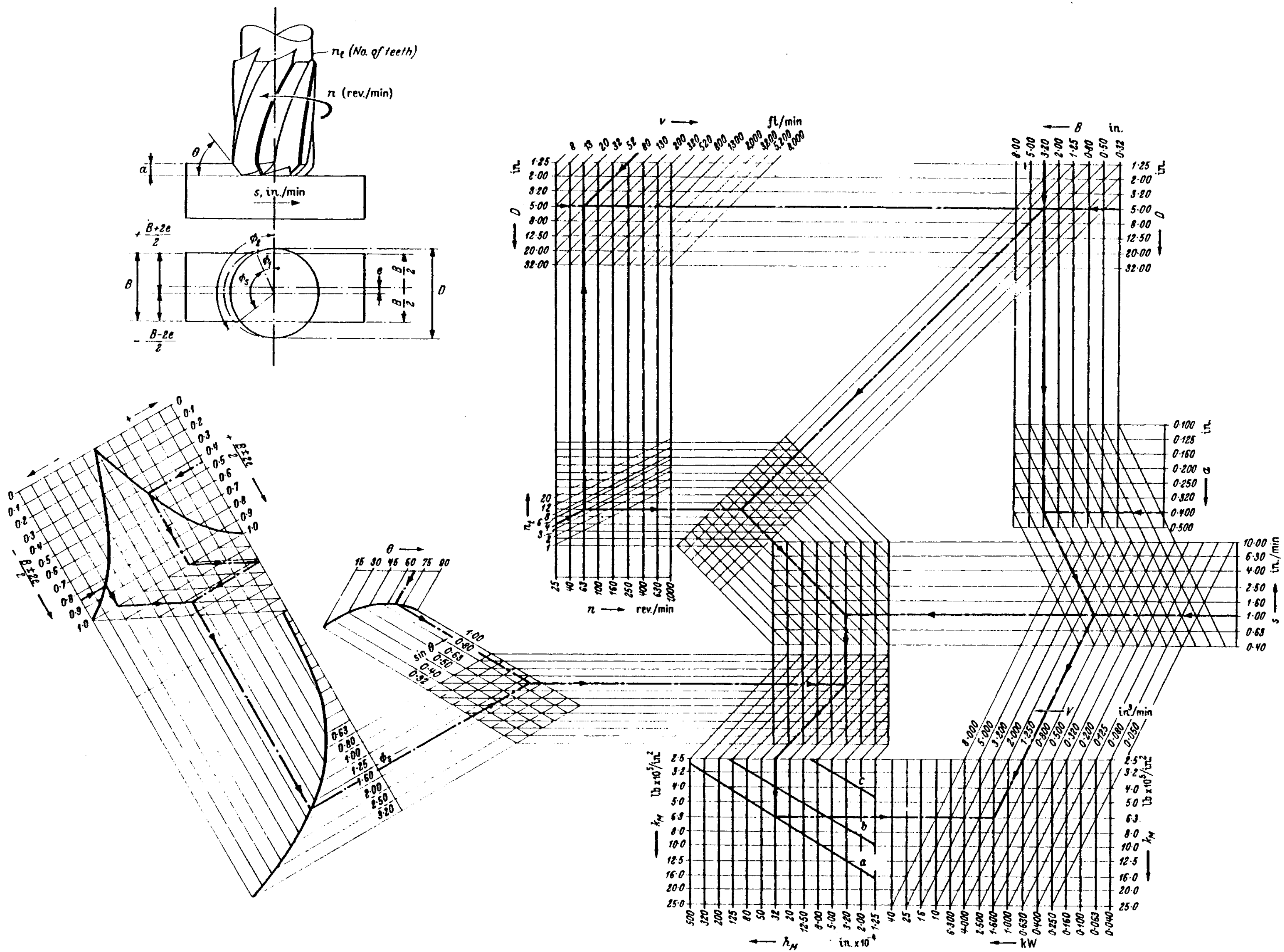
n_t (No. of teeth)
n (rev./min)
s, in./min
v ft/min
n rev./min
D in.
B in.
a in.
s in./min
V in.^{3}/min
k_M lb x 10^5/in.^{2}
h_M in. x 10^{-4}
kW
sin θ
ϕ_s

FIG. 28a



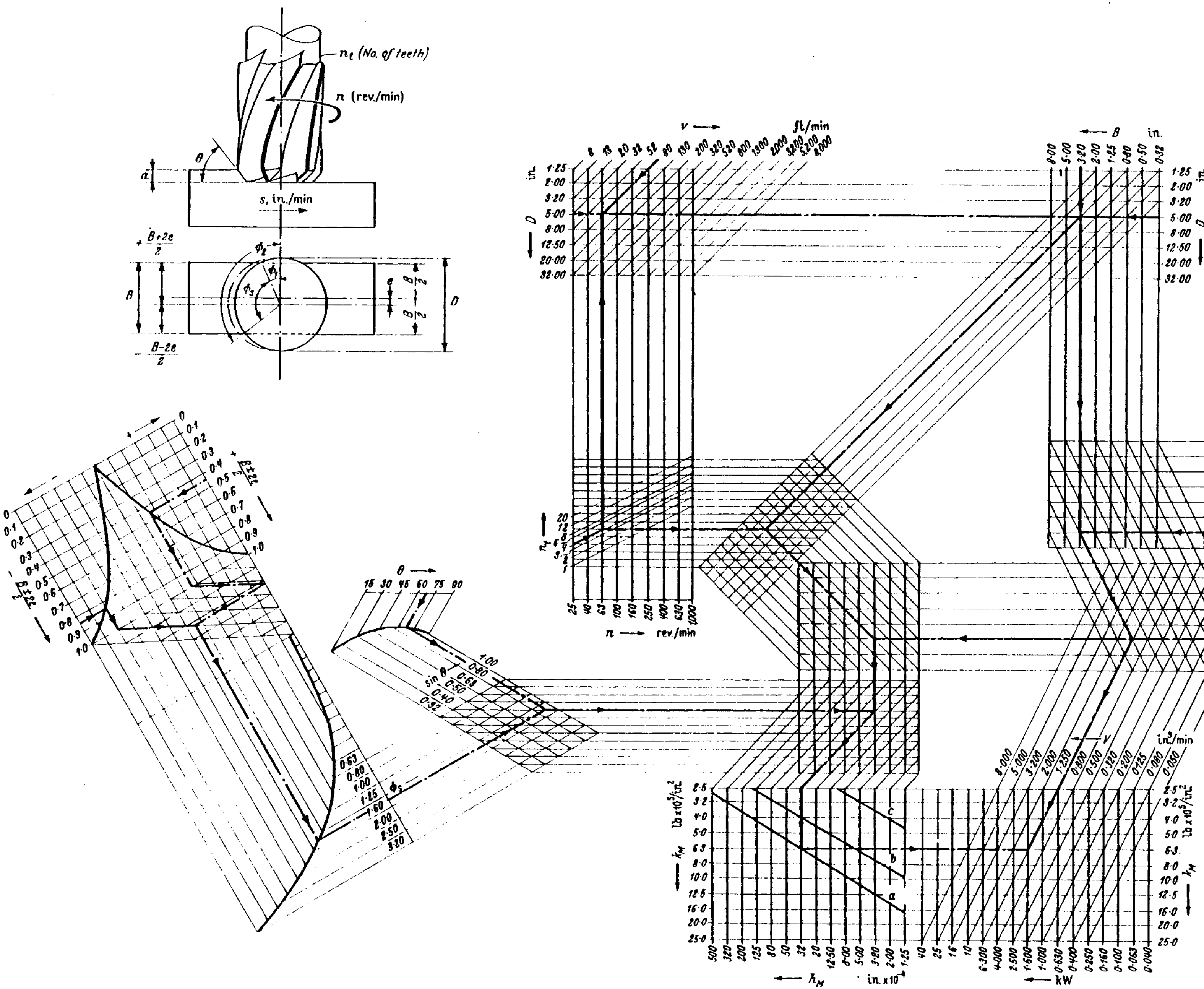
n_t (No. of teeth)
n (rev./min)
s, in./min
v
ft/min
B
in.
D
n
rev./min
n_t
a
s
in./min
θ
sin θ
ϕ_s
in.³/min
V
k_M
lb x 10³/in.²
h_M
in. x 10⁻⁴
kW

Fig. 28a



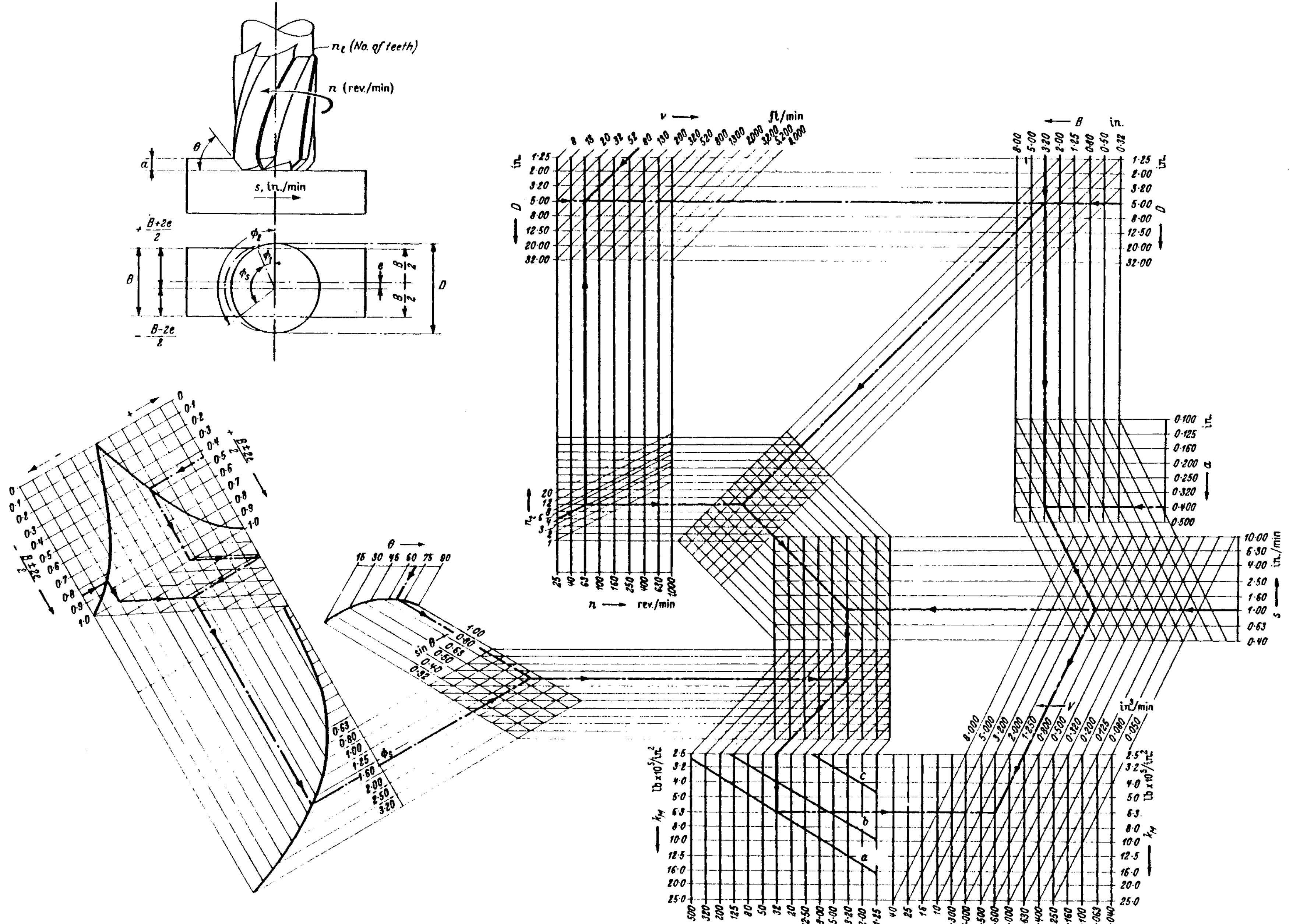
n_t (No. of teeth)
n (rev./min)
s, in./min
θ
a
$+\frac{B+2e}{2}$
$-\frac{B-2e}{2}$
B
D
e
ϕ_t
ϕ_s
v ft/min
D in.
B in.
n rev./min
n_t
a in.
s in./min
θ
$\sin \theta$
ϕ_s
k_M lb x 10^5/in.^{2}
h_M in. x 10^{-4}
kW
in.^{3}/min
V

FIG. 28a



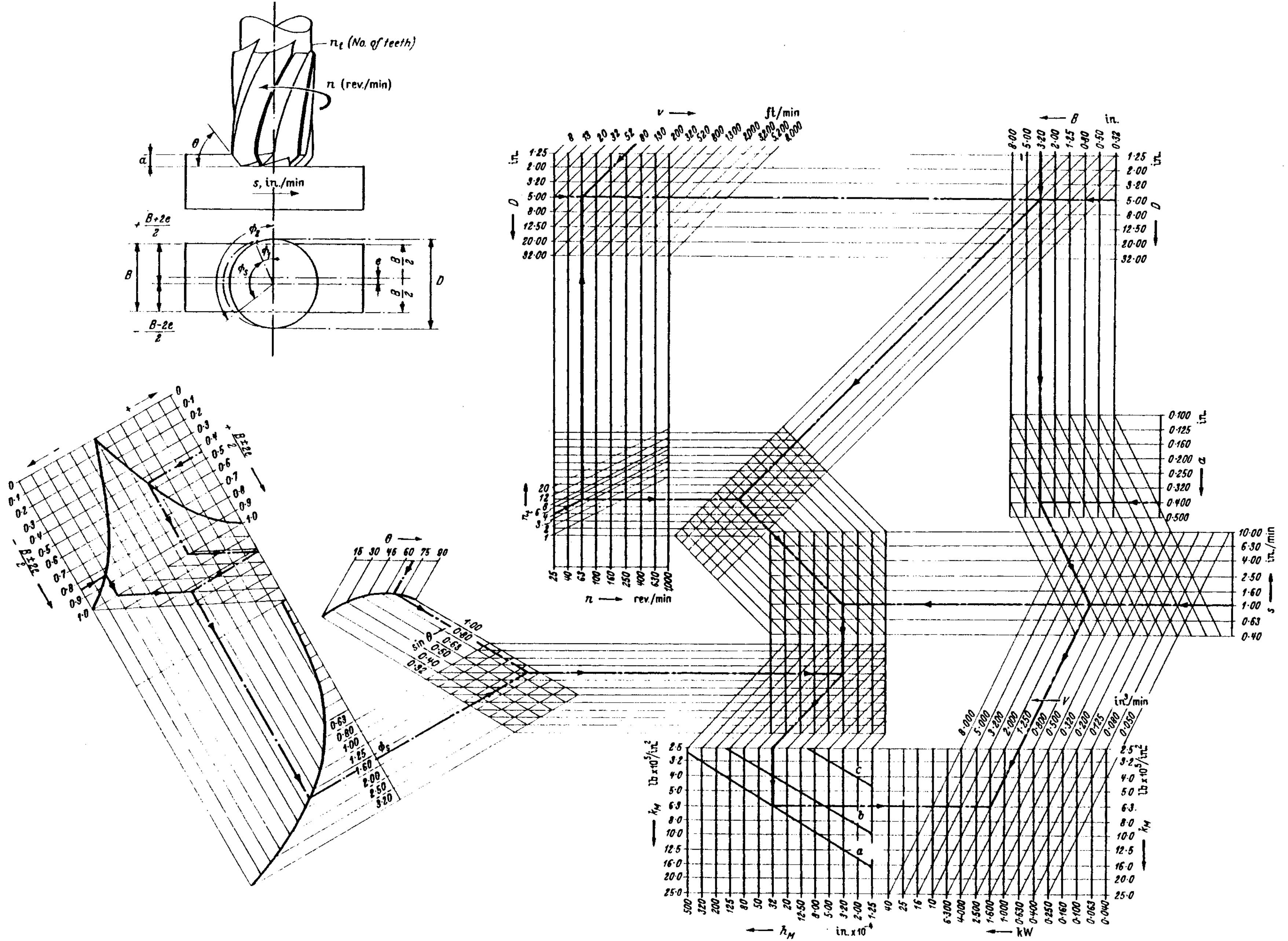
n_t (No. of teeth)
n (rev./min)
s, in./min
θ
a
B
D
e
$\frac{B+2e}{2}$
$\frac{B-2e}{2}$
$\frac{B}{2}$
ϕ_e
ϕ_s
v ft/min
D in.
B in.
n rev./min
a in.
s in./min
θ
sin θ
ϕ_s
k_M lb x 10^5/in^2
h_M in. x 10^{-4}
kW
in^3/min

FIG. 28a



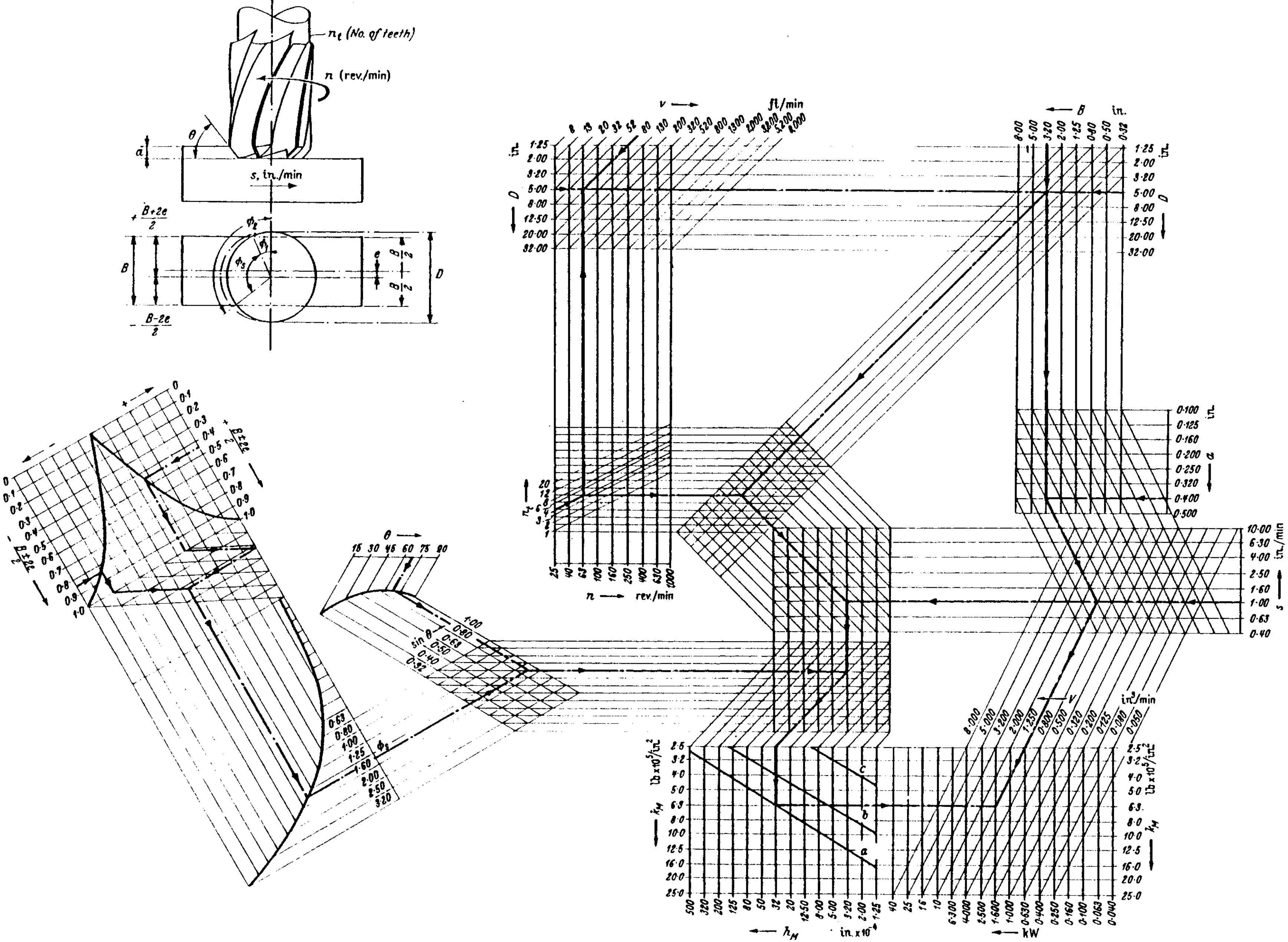
n_t (No. of teeth)
n (rev./min)
s, in./min
ft/min
rev./min
in./min
in.³/min
kW
lb x 10⁵/in.²
in. x 10⁻⁴
sin θ

FIG. 28a



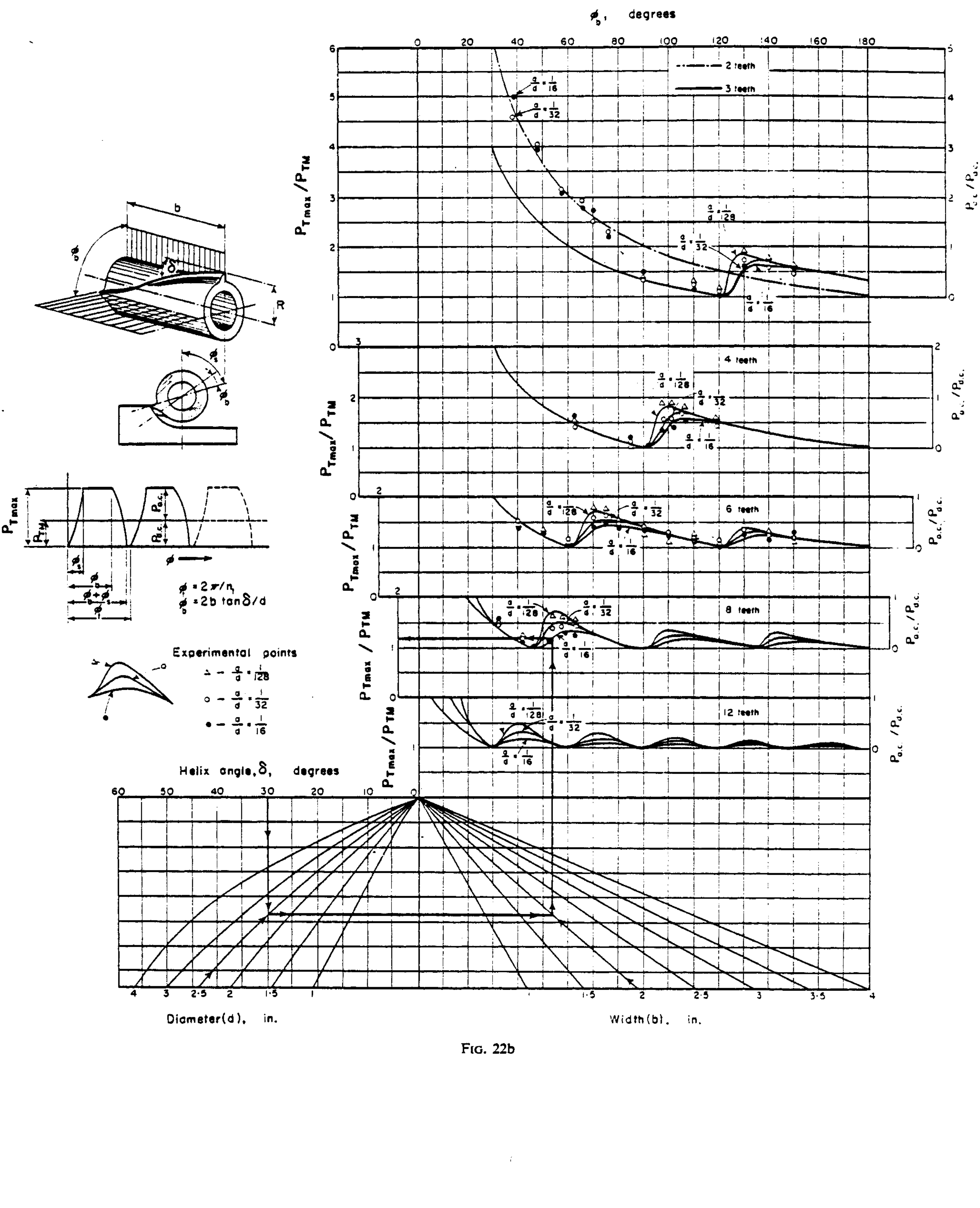
n_t (No. of teeth)
n (rev./min)
θ
a
s, in./min
$\frac{B+2e}{2}$
$\frac{B-2e}{2}$
ϕ_2
ϕ_1
ϕ_s
B
e
$\frac{B}{2}$
D
v ft/min
8 13 20 32 52 80 130 200 320 520 800 1300 2000 3200 5200 8000
D in.
1·25 2·00 3·20 5·00 8·00 12·50 20·00 32·00
B in.
8·00 5·00 3·20 2·00 1·25 0·80 0·50 0·32
n_t
20 12 8 6 4 3 2 1
n rev./min
25 40 63 100 160 250 400 630 1000
a in.
0·100 0·125 0·160 0·200 0·250 0·320 0·400 0·500
s in./min
10·00 6·30 4·00 2·50 1·60 1·00 0·63 0·40
θ
15 30 45 60 75 90
$\sin\theta$
1·00 0·80 0·63 0·50 0·40 0·32
ϕ_s
0·63 0·80 1·00 1·25 1·60 2·00 2·50 3·20
0 0·1 0·2 0·3 0·4 0·5 0·6 0·7 0·8 0·9 1·0
V in.³/min
8·000 5·000 3·200 2·000 1·250 0·800 0·500 0·320 0·200 0·125 0·080 0·050
k_M lb x 10⁵/in.²
2·5 3·2 4·0 5·0 6·3 8·0 10·0 12·5 16·0 20·0 25·0
a b c
h_M in. x 10⁻⁴
500 320 200 125 80 50 32 20 12·50 8·00 5·00 3·20 2·00 1·25
kW
40 25 16 10 6·300 4·000 2·500 1·600 1·000 0·630 0·400 0·250 0·160 0·100 0·063 0·040

FIG. 28a



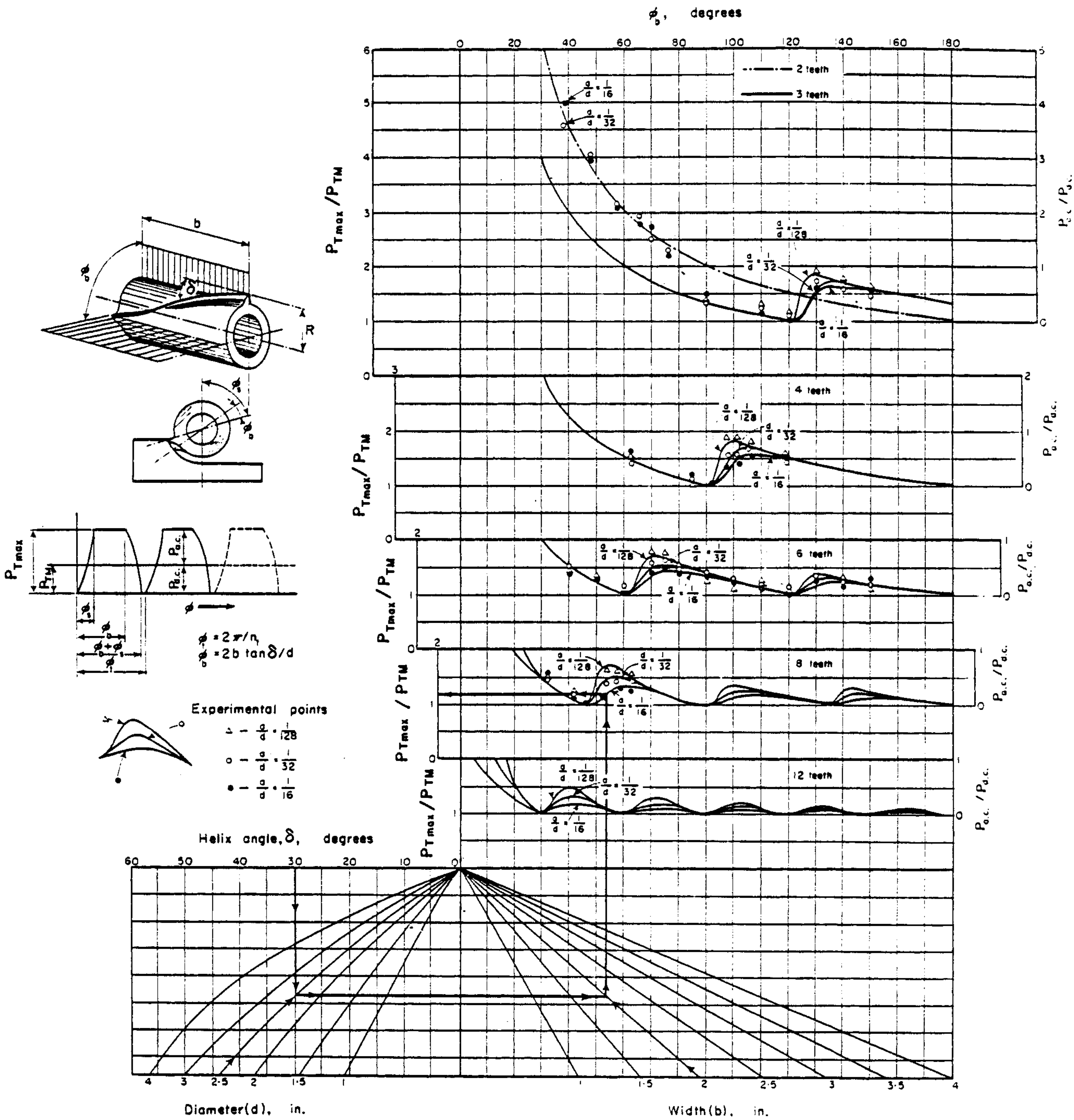
n_t (No. of teeth)
n (rev./min)
θ
a
s, in./min
B
D
e
$\frac{B+2e}{2}$
$\frac{B-2e}{2}$
$\frac{B}{2}$
ϕ_s
ϕ_1
ϕ_2
$\frac{B \pm 2e}{2}$
sin θ
v ft/min
D in.
B in.
a in.
n_t
n rev./min
s in./min
v in.^{3}/min
k_M lb x 10^5/in.^{2}
h_M in. x 10^{-4}
kW
a
b
c

FIG. 28a



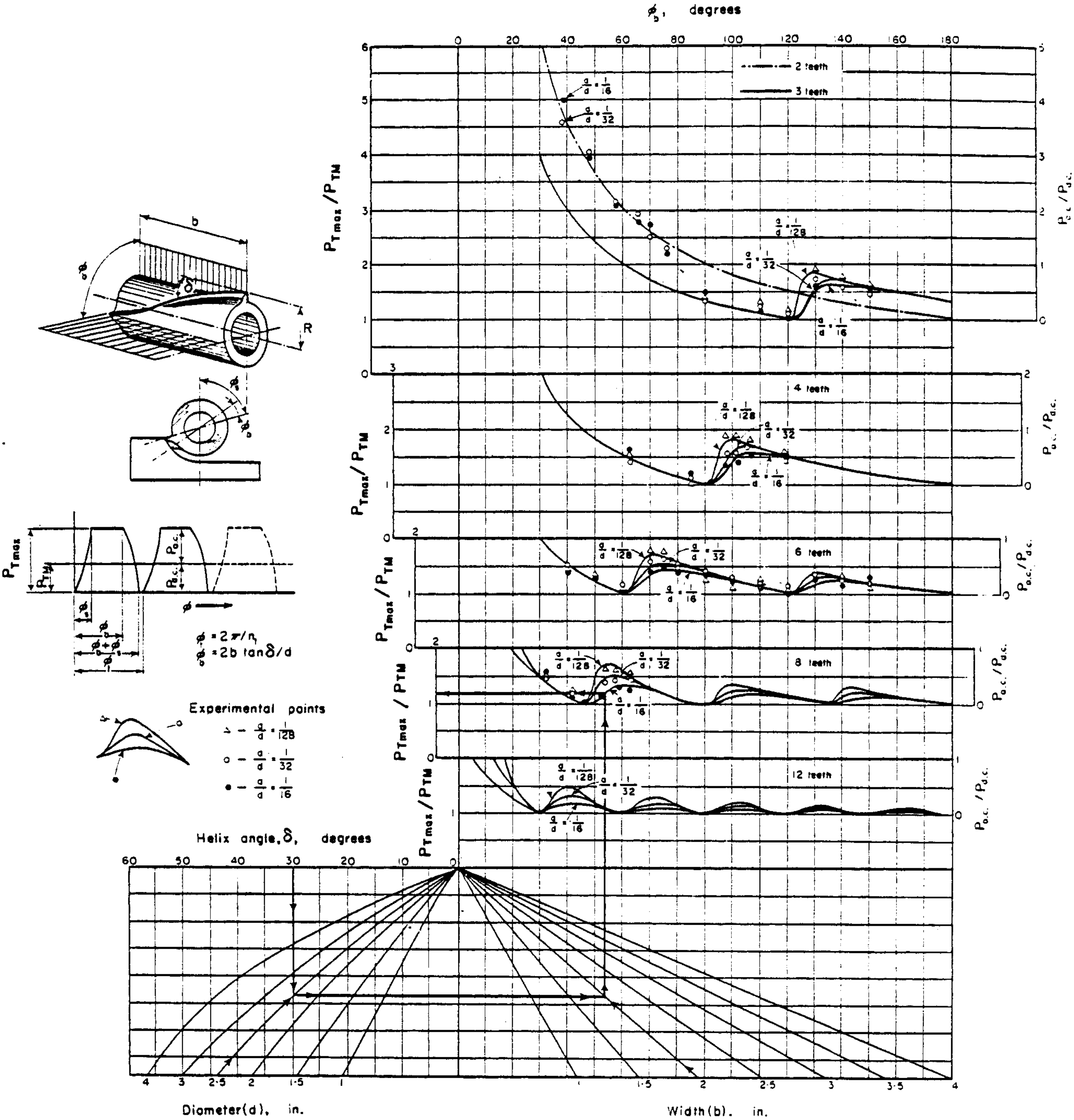
ϕ_b, degrees
2 teeth
3 teeth
4 teeth
6 teeth
8 teeth
12 teeth
P_Tmax / P_TM
P_a.c. / P_d.c.
Experimental points
Helix angle, δ, degrees
Diameter(d), in.
Width(b), in.
ϕ_t = 2π/n_t
ϕ_b = 2b tan δ/d

Fig. 22b



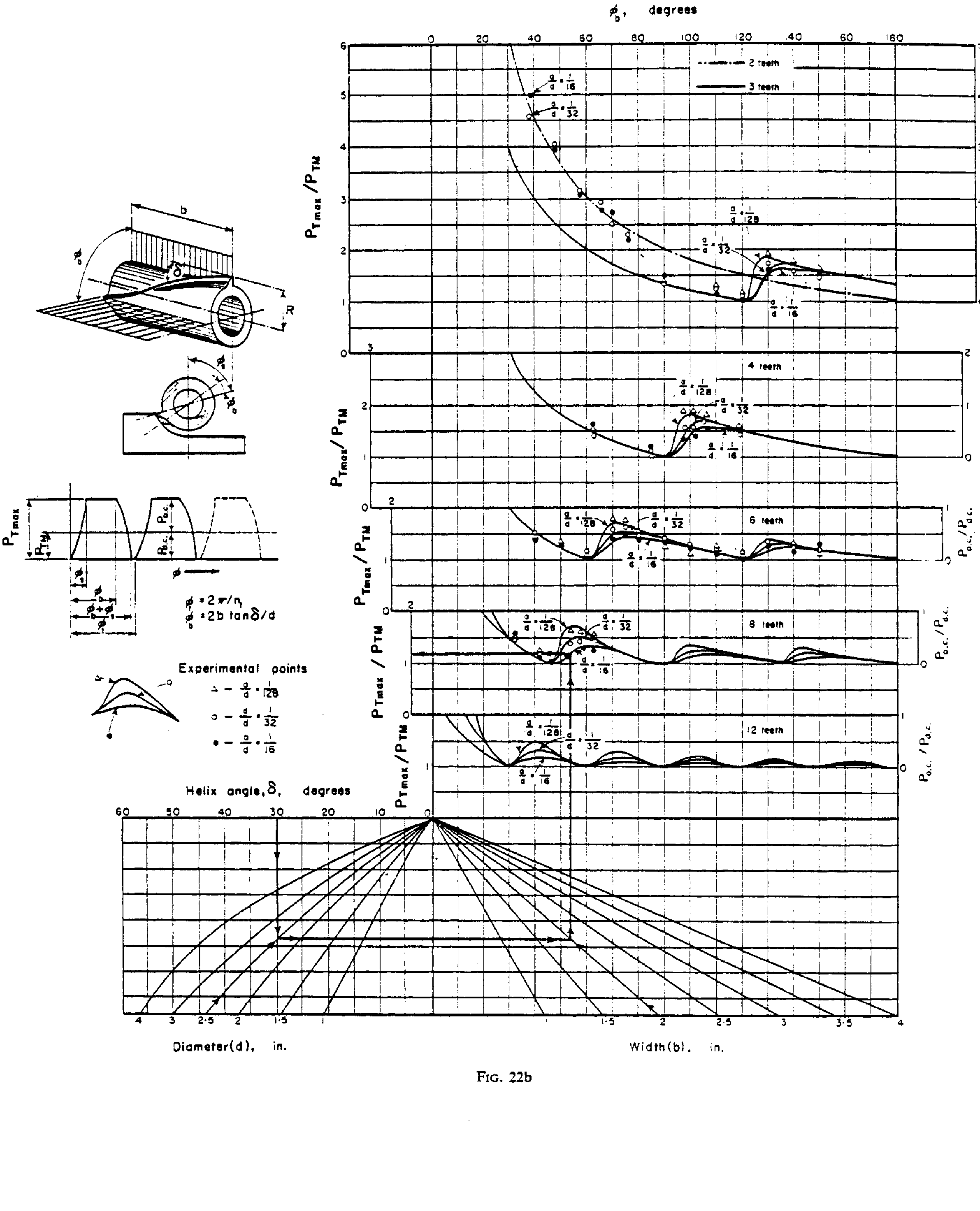
ϕ_b, degrees
2 teeth
3 teeth
4 teeth
6 teeth
8 teeth
12 teeth
$P_{T\max}/P_{TM}$
$P_{a.c.}/P_{d.c.}$
$\frac{a}{d}=\frac{1}{16}$
$\frac{a}{d}=\frac{1}{32}$
$\frac{a}{d}=\frac{1}{128}$
Experimental points
$\phi_t=2\pi/n_t$
$\phi_b=2b\tan\delta/d$
Helix angle, δ, degrees
Diameter (d), in.
Width (b), in.

FIG. 22b



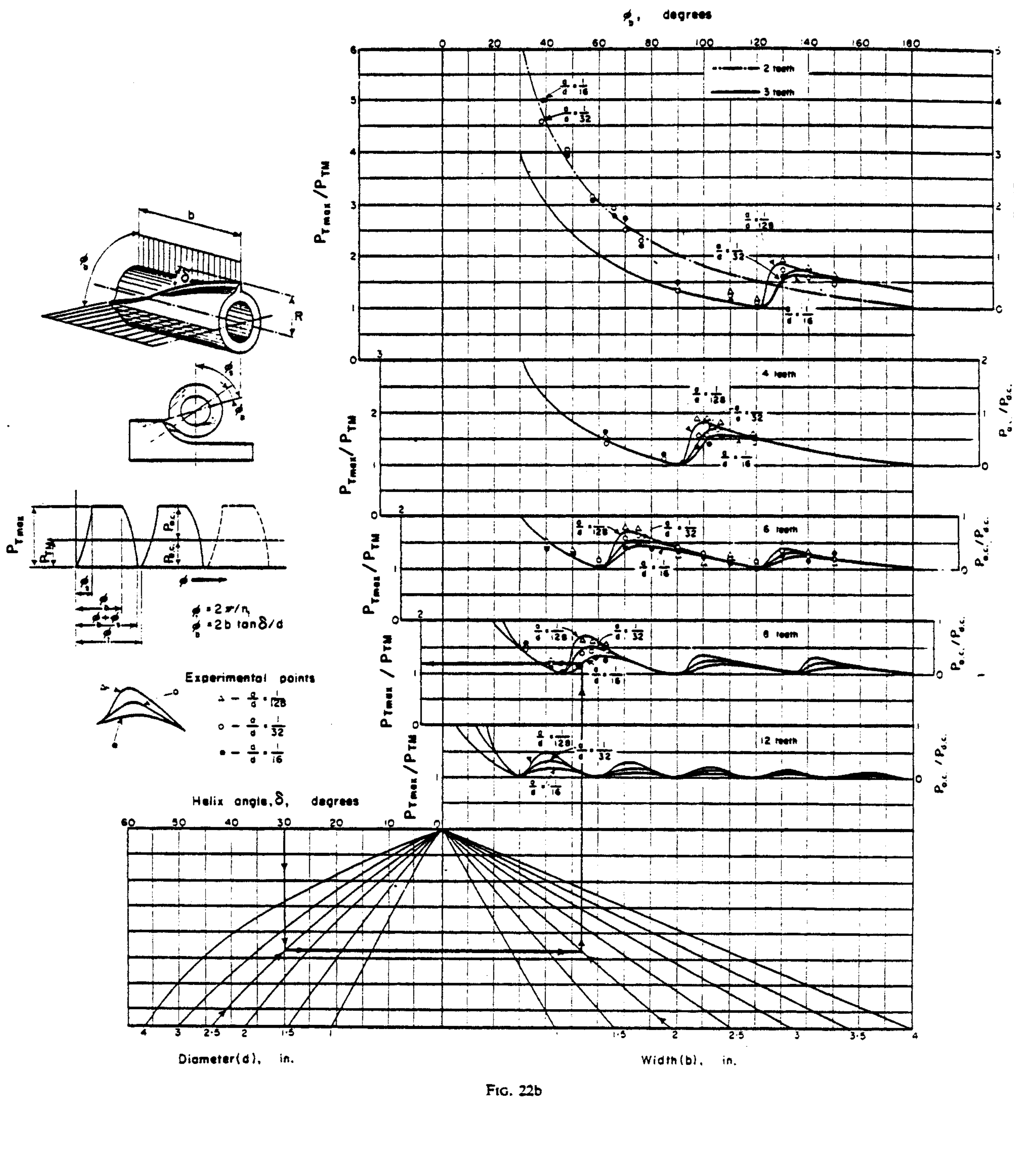
ϕ_b, degrees
2 teeth
3 teeth
4 teeth
6 teeth
8 teeth
12 teeth
$P_{T\max}/P_{TM}$
$P_{a.c.}/P_{d.c.}$
$\frac{a}{d}=\frac{1}{16}$
$\frac{a}{d}=\frac{1}{32}$
$\frac{a}{d}=\frac{1}{128}$
b
R
δ
ϕ_a
ϕ_b
$P_{T\max}$
P_{TM}
$P_{a.c.}$
$P_{d.c.}$
$\phi = 2\pi/n_1$
$\phi_a = 2b\tan\delta/d$
Experimental points
Helix angle, δ, degrees
Diameter (d), in.
Width (b), in.

FIG. 22b



ϕ_b, degrees
P_{Tmax}/P_{TM}
$P_{a.c.}/P_{d.c.}$
2 teeth
3 teeth
4 teeth
6 teeth
8 teeth
12 teeth
Experimental points
$\triangle$ – $\frac{a}{d} = \frac{1}{128}$
$\circ$ – $\frac{a}{d} = \frac{1}{32}$
$\bullet$ – $\frac{a}{d} = \frac{1}{16}$
$\phi = 2\pi/n_1$
$\phi_d = 2b \tan\delta/d$
Helix angle, δ, degrees
Diameter(d), in.
Width(b), in.

FIG. 22b



φb, degrees
2 teeth
3 teeth
4 teeth
6 teeth
8 teeth
12 teeth
P_Tmax / P_TM
Pa.c. / Pd.c.
Experimental points
Helix angle, δ, degrees
Diameter(d), in.
Width(b), in.
φ = 2π/n
φb = 2b tanδ/d

FIG. 22b

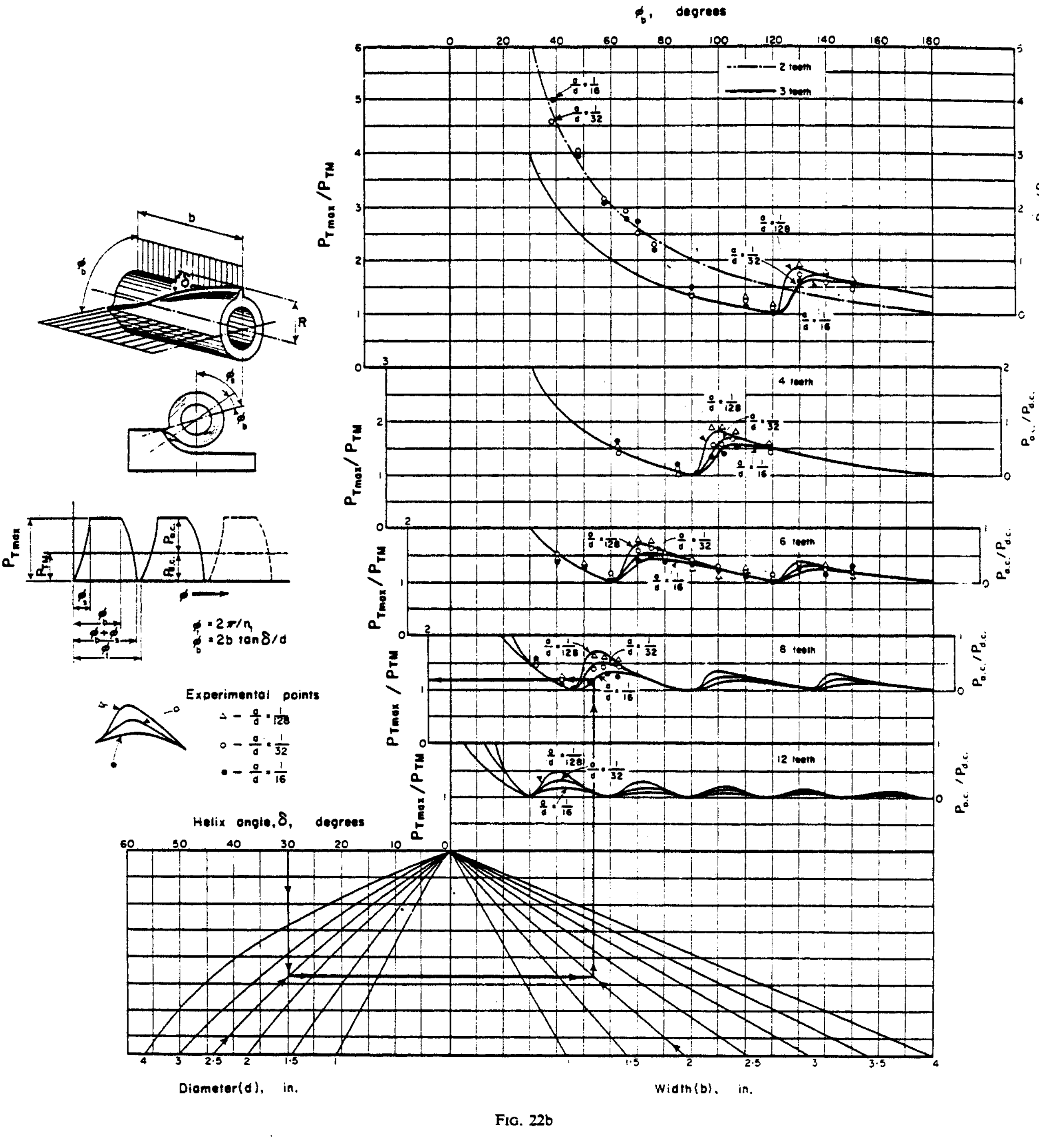
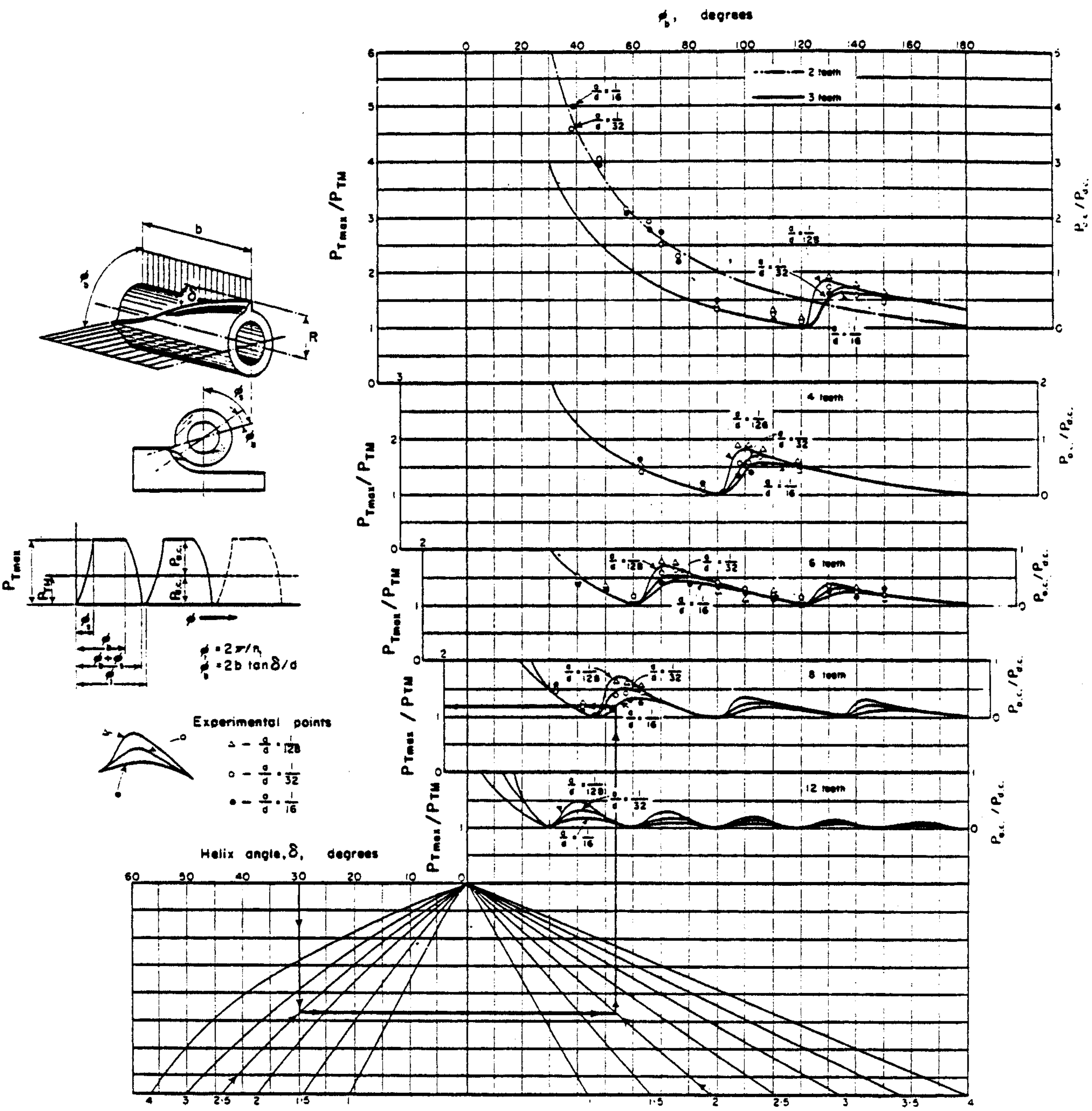
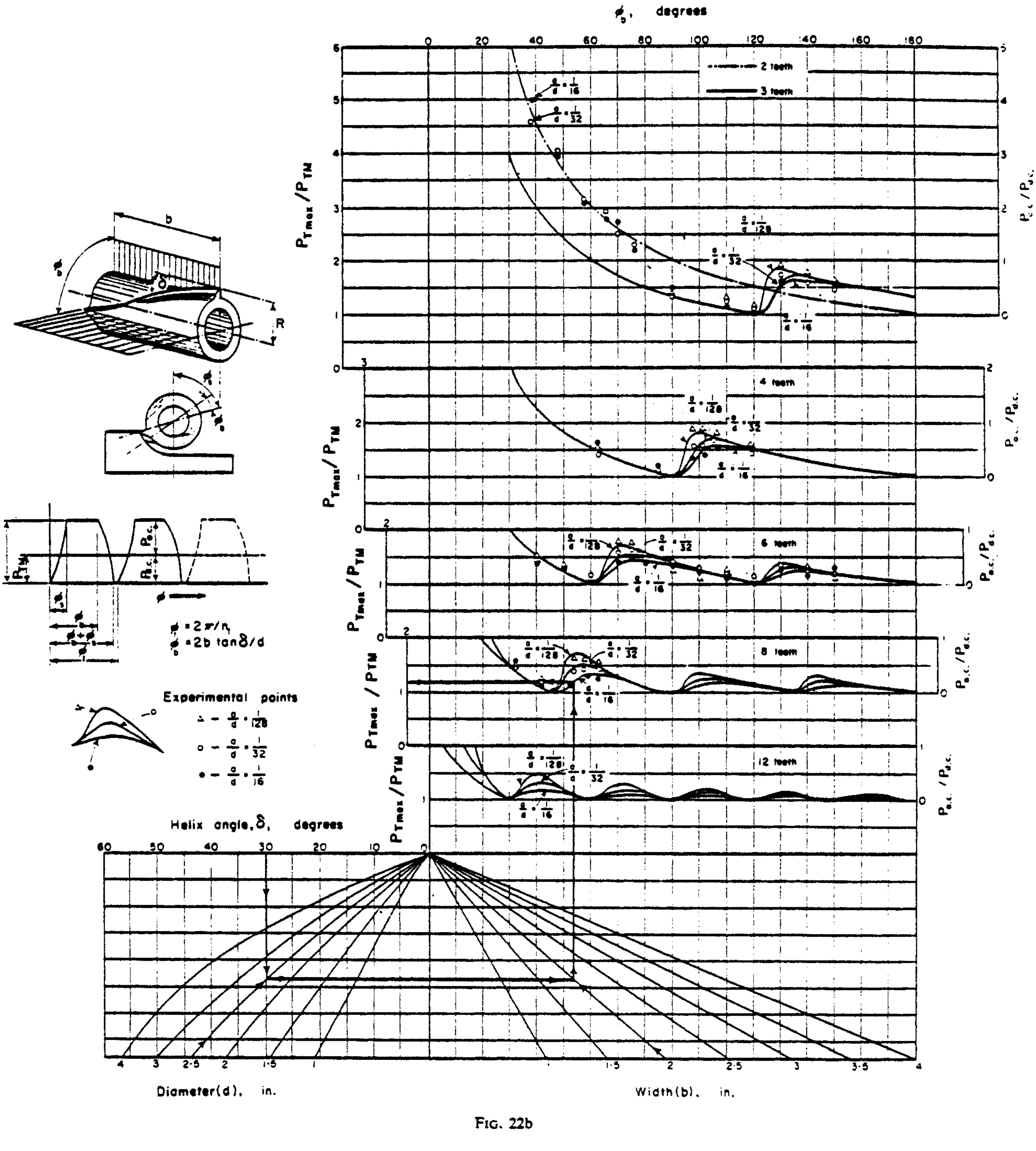

φb, degrees
2 teeth
3 teeth
4 teeth
6 teeth
8 teeth
12 teeth
PTmax / PTM
Pa.c. / Pd.c.
Experimental points
φ = 2π/n
φb = 2b tan δ/d
Helix angle, δ, degrees
Diameter (d), in.
Width (b), in.

FIG. 22b



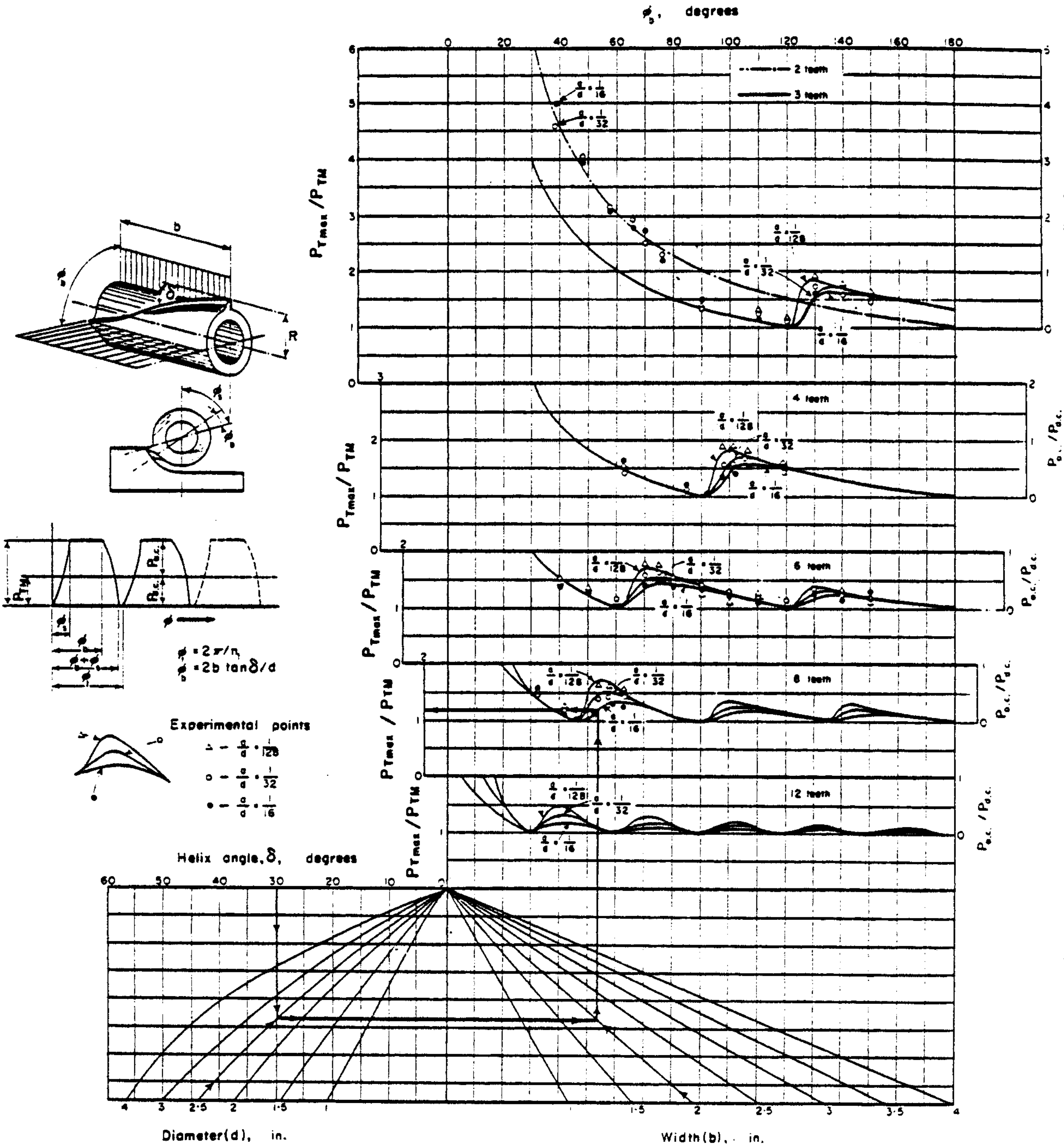
ϕ_b, degrees
2 teeth
3 teeth
4 teeth
6 teeth
8 teeth
12 teeth
P_{Tmax}/P_{TM}
$P_{a.c.}/P_{d.c.}$
$\phi = 2\pi/n$
$\phi_s = 2b \tan\delta/d$
Experimental points
Helix angle, δ, degrees
Diameter(d), in.
Width(b), in.

FIG. 22b



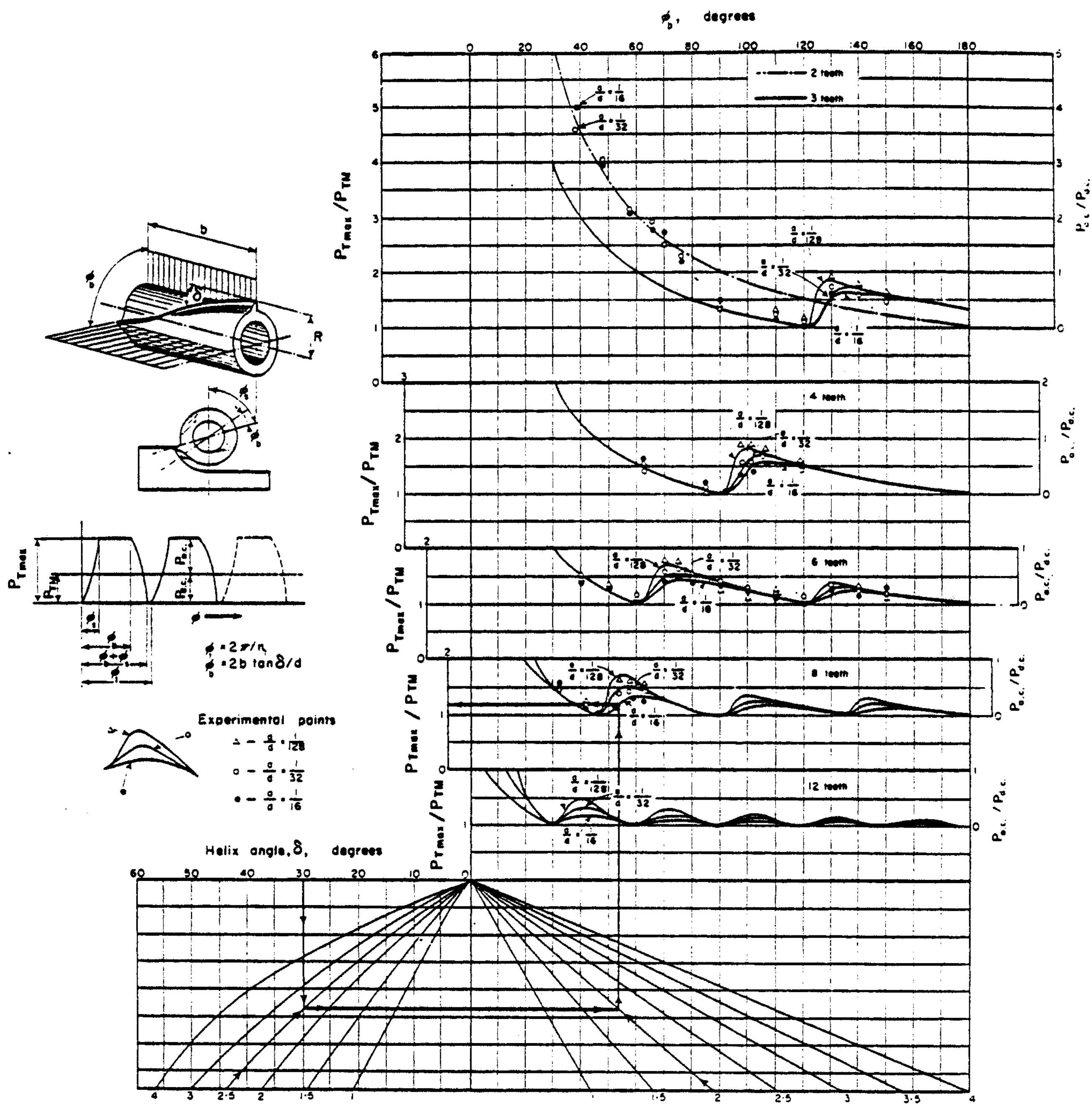
ϕ_b, degrees
2 teeth
3 teeth
4 teeth
6 teeth
8 teeth
12 teeth
P_{Tmax}/P_{TM}
$P_{a.c.}/P_{d.c.}$
Experimental points
Helix angle, δ, degrees
Diameter(d), in.
Width(b), in.
$\phi = 2\pi/n_t$
$\phi_b = 2b \tan\delta/d$

FIG. 22b



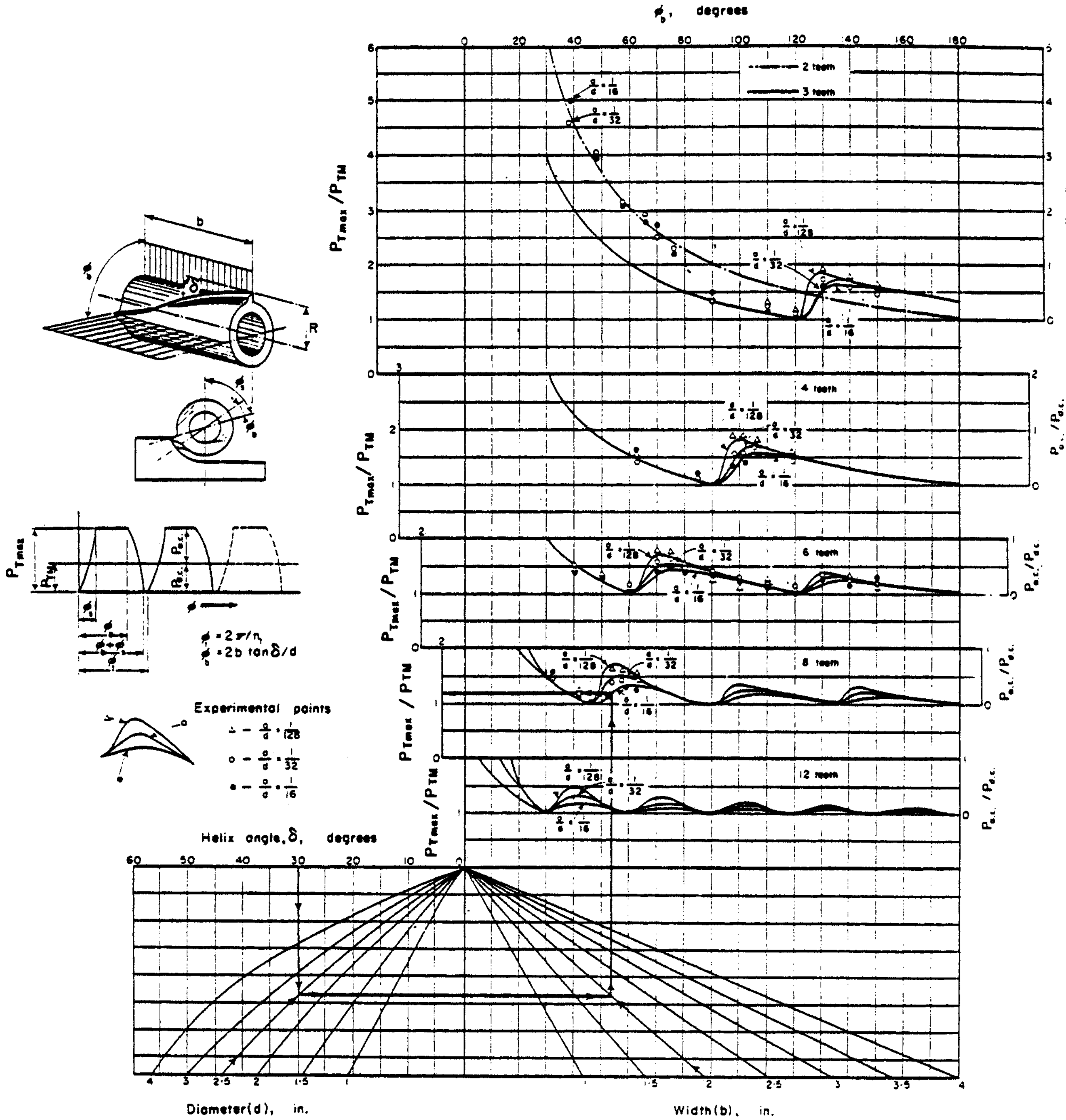
ϕ_b, degrees
2 teeth
3 teeth
4 teeth
6 teeth
8 teeth
12 teeth
$P_{T max}/P_{TM}$
$P_{a.c.}/P_{d.c.}$
$\phi_s = 2\pi/n$
$\phi_b = 2b \tan\delta/d$
Experimental points
Helix angle, δ, degrees
Diameter (d), in.
Width (b), in.

FIG. 22b



φ_b, degrees
2 teeth
3 teeth
4 teeth
6 teeth
8 teeth
12 teeth
P_Tmax / P_TM
P_a.c. / P_d.c.
Experimental points
Helix angle, δ, degrees
Diameter(d), in.
Width(b), in.

FIG. 22b



ϕ_b, degrees
2 teeth
3 teeth
4 teeth
6 teeth
8 teeth
12 teeth
P_Tmax / P_TM
P_o.c. / P_o.c.
Experimental points
Helix angle, δ, degrees
Diameter (d), in.
Width (b), in.
ϕ = 2π/n
ϕ = 2b tan δ/d

FIG. 22b

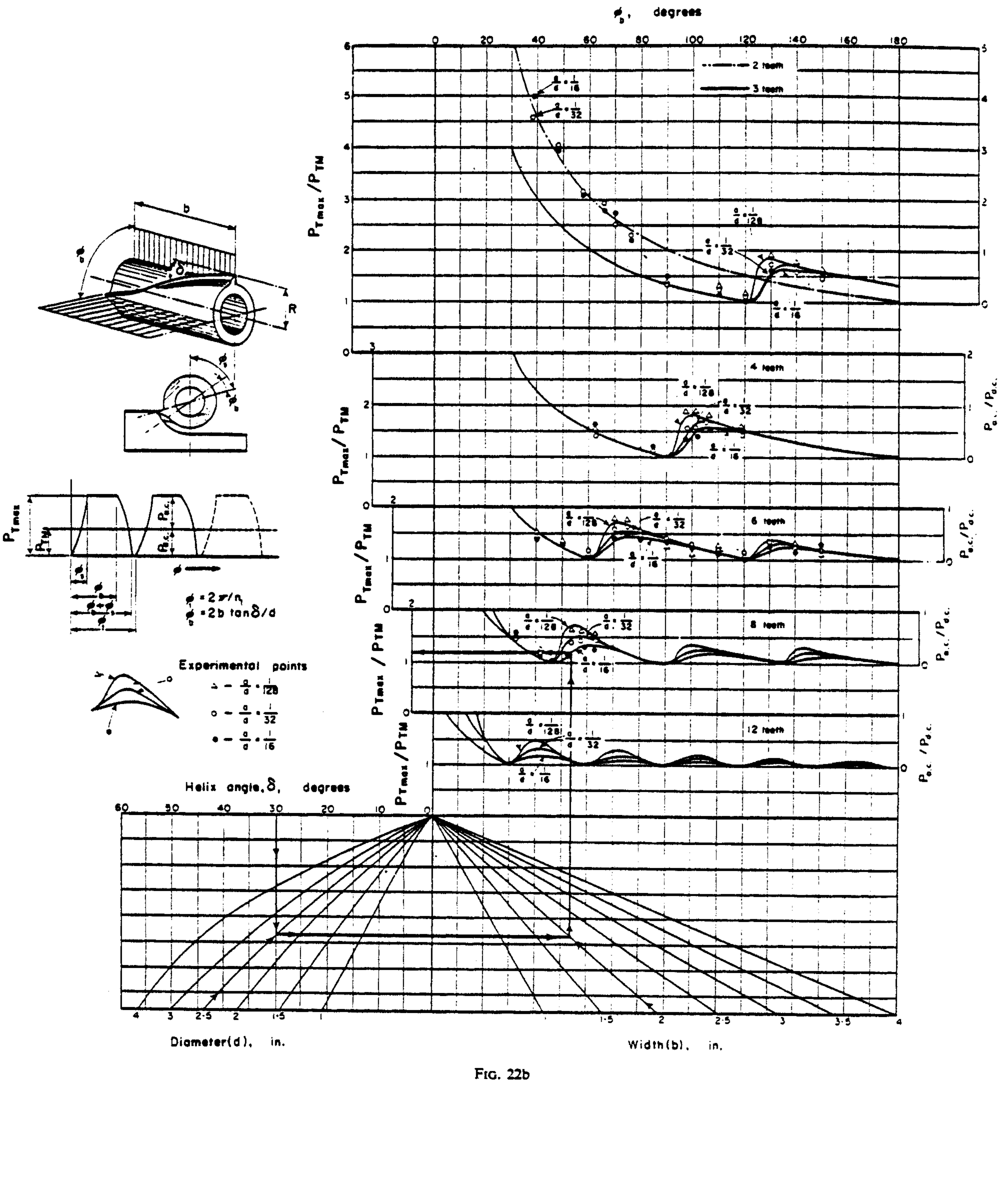
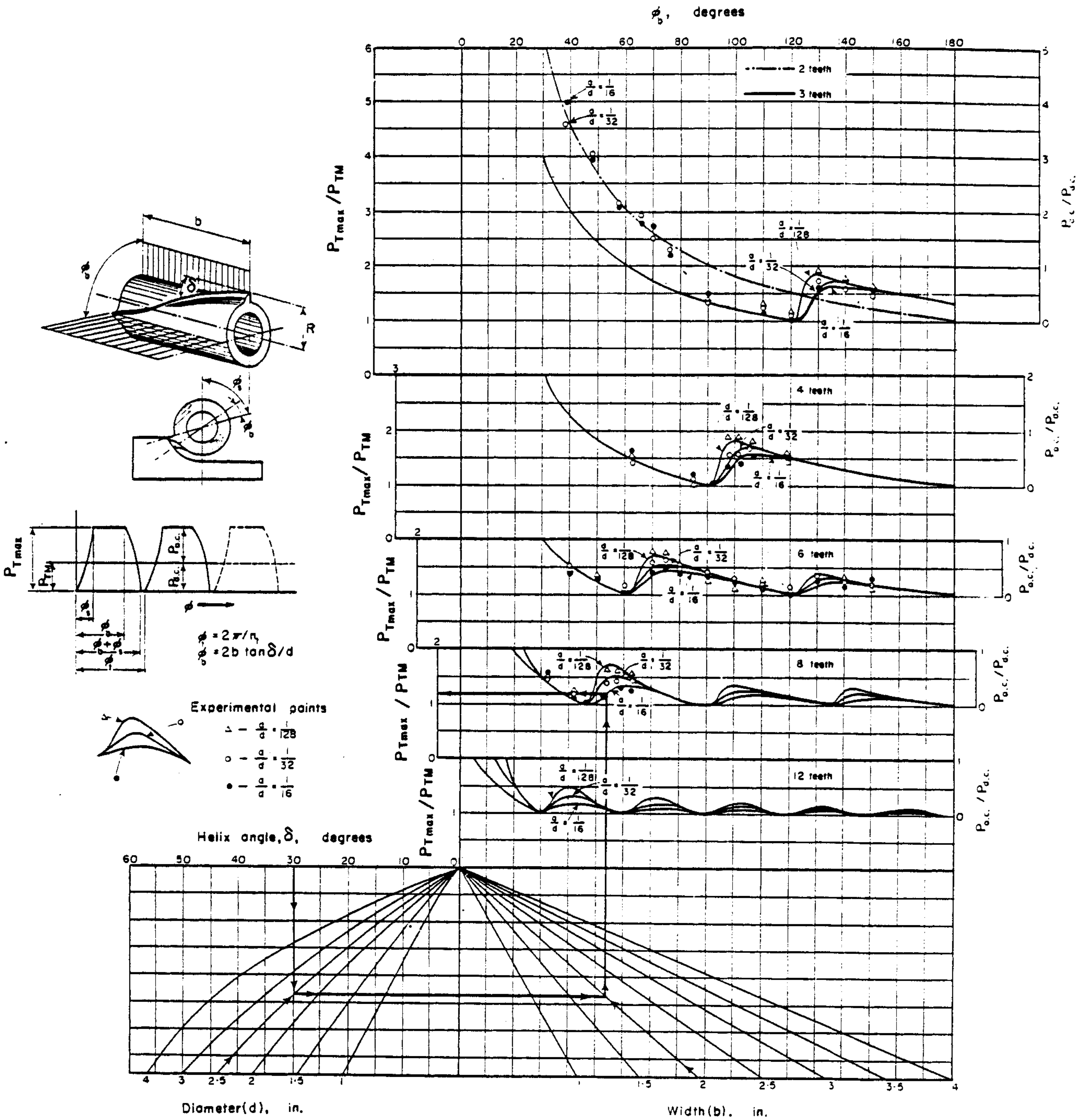
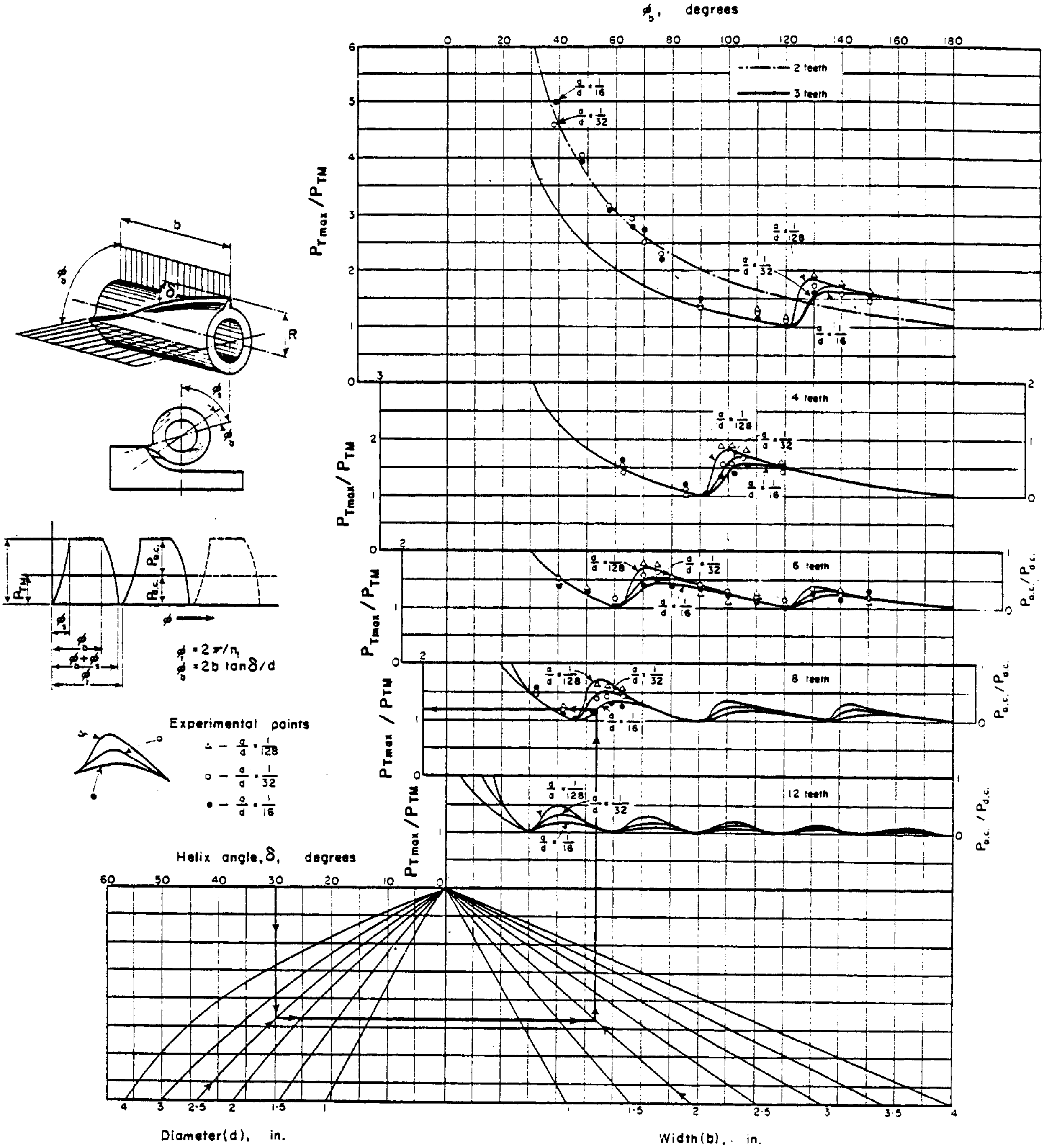

ϕ_b, degrees
P_{Tmax}/P_{TM}
$P_{a.c.}/P_{a.c.}$
2 teeth
3 teeth
4 teeth
6 teeth
8 teeth
12 teeth
b
R
P_{Tmax}
P_{TM}
$\phi = 2\pi/n_t$
$\phi_b = 2b\tan\delta/d$
Experimental points
$a/d = 1/128$
$a/d = 1/32$
$a/d = 1/16$
Helix angle, δ, degrees
Diameter (d), in.
Width (b), in.

FIG. 22b



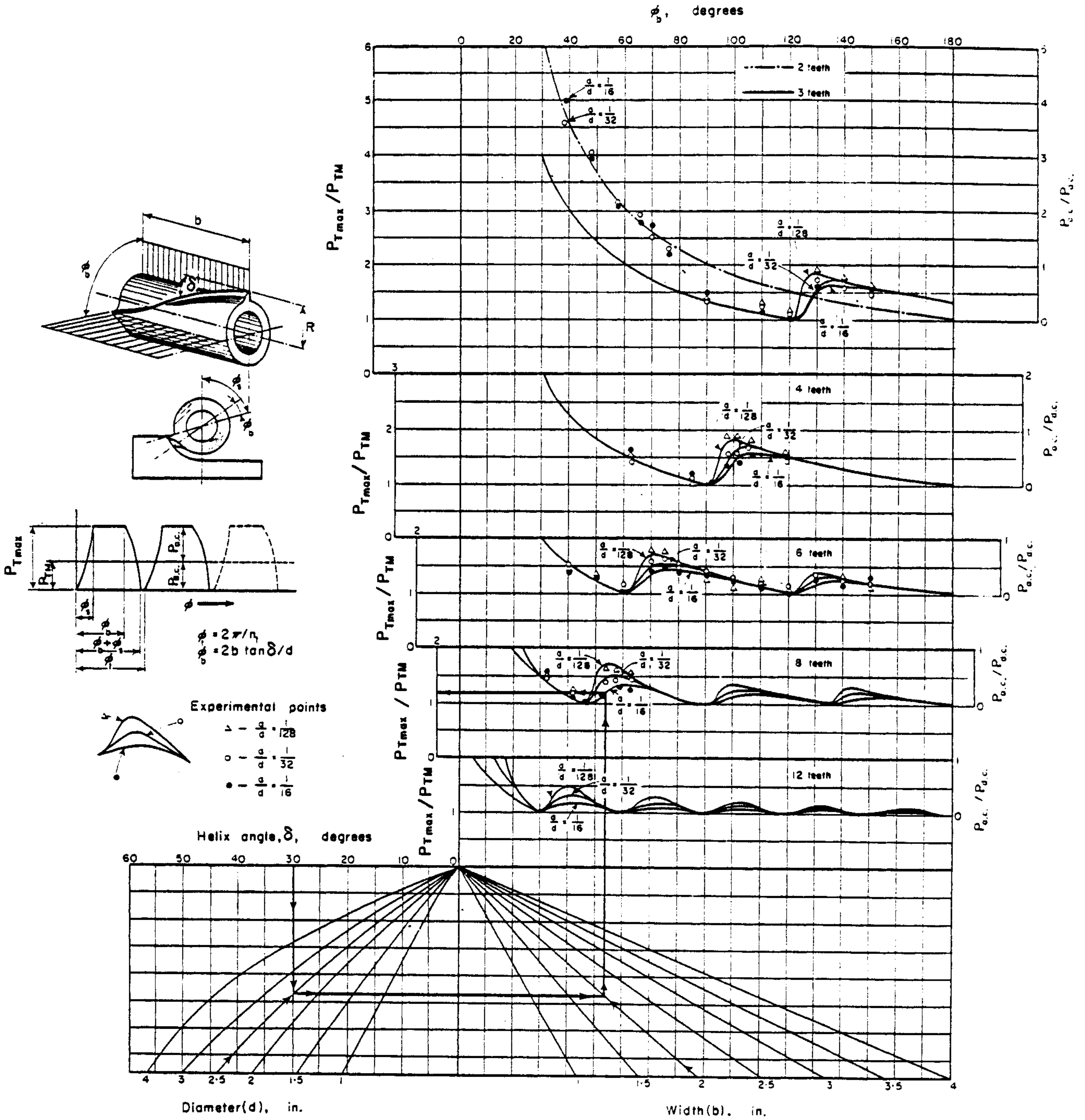
ϕ_b, degrees
2 teeth
3 teeth
4 teeth
6 teeth
8 teeth
12 teeth
$P_{T\,max}/P_{TM}$
$P_{a.c.}/P_{d.c.}$
$\frac{a}{d}=\frac{1}{16}$
$\frac{a}{d}=\frac{1}{32}$
$\frac{a}{d}=\frac{1}{128}$
$\phi = 2\pi/n_t$
$\phi_b = 2b\tan\delta/d$
Experimental points
Helix angle, δ, degrees
Diameter(d), in.
Width(b), in.

FIG. 22b



φb, degrees
2 teeth
3 teeth
4 teeth
6 teeth
8 teeth
12 teeth
PTmax / PTM
Pa.c. / Pd.c.
Experimental points
Helix angle, δ, degrees
Diameter(d), in.
Width(b), in.

FIG. 22b



ϕ_b, degrees
2 teeth
3 teeth
4 teeth
6 teeth
8 teeth
12 teeth
$P_{T\,max}/P_{TM}$
$P_{a.c.}/P_{d.c.}$
Experimental points
Helix angle, δ, degrees
Diameter (d), in.
Width (b), in.
$\phi = 2\pi/n_t$
$\phi_b = 2b\tan\delta/d$

FIG. 22b