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).

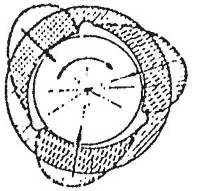


Fig. 436

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.

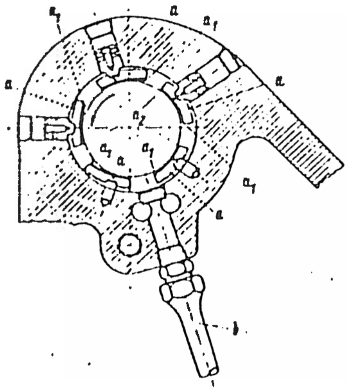


Fig. 437. "Filmatic" grinding spindle bearing (Cincinnati).

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.

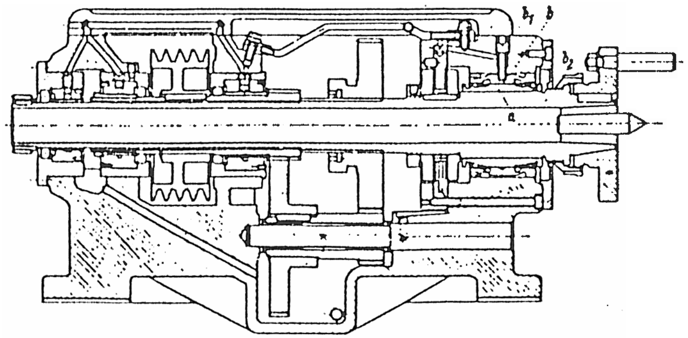


Fig. 438. Lathe spindle with ready-made plain bearings.

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 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.

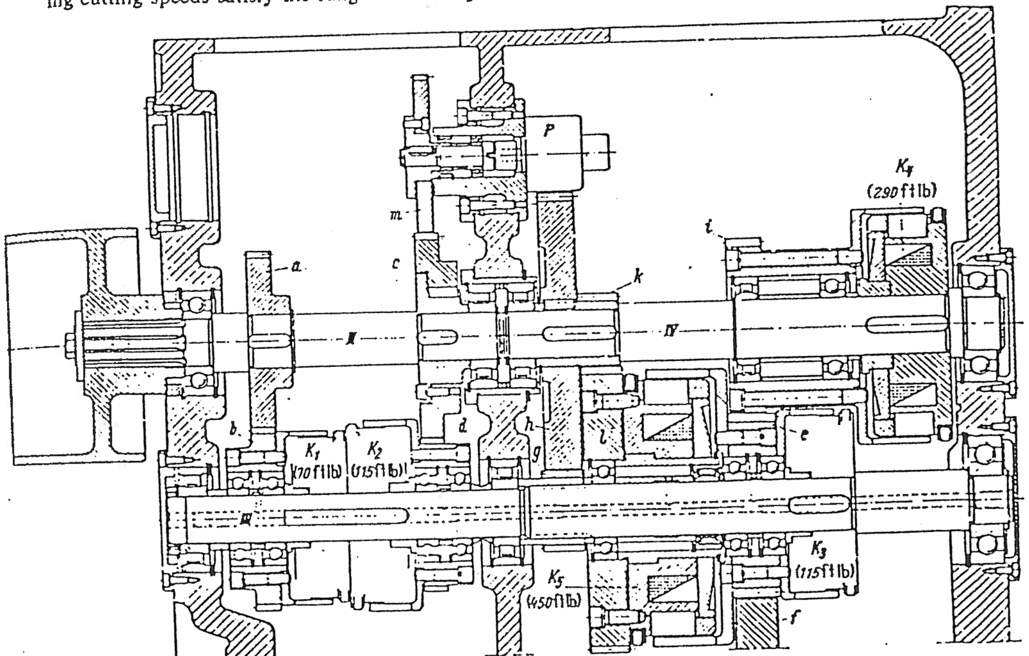


Fig. 439. Spindle head gearbox of a turret lathe (Gebr. Heinemann, St. Georgen, Germany).

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 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 1) 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.

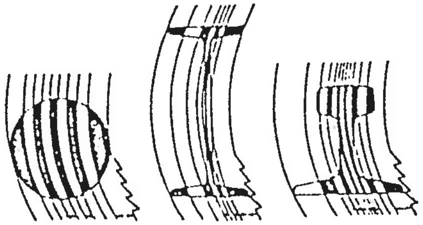


FIG. 460. Feed adjustment during a saw cut to suit the material section in front of the saw blade.

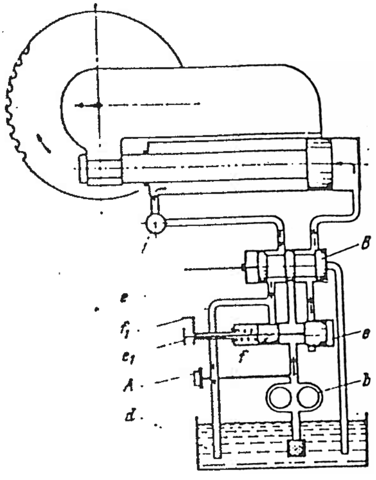


FIG. 461. Schematic lay-out of the hydraulic feed drive for a sawing machine (Gustav Wagner, Reutlingen, Germany).

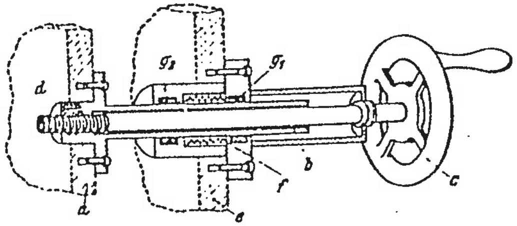


FIG. 462. "Inchworm" device.

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:

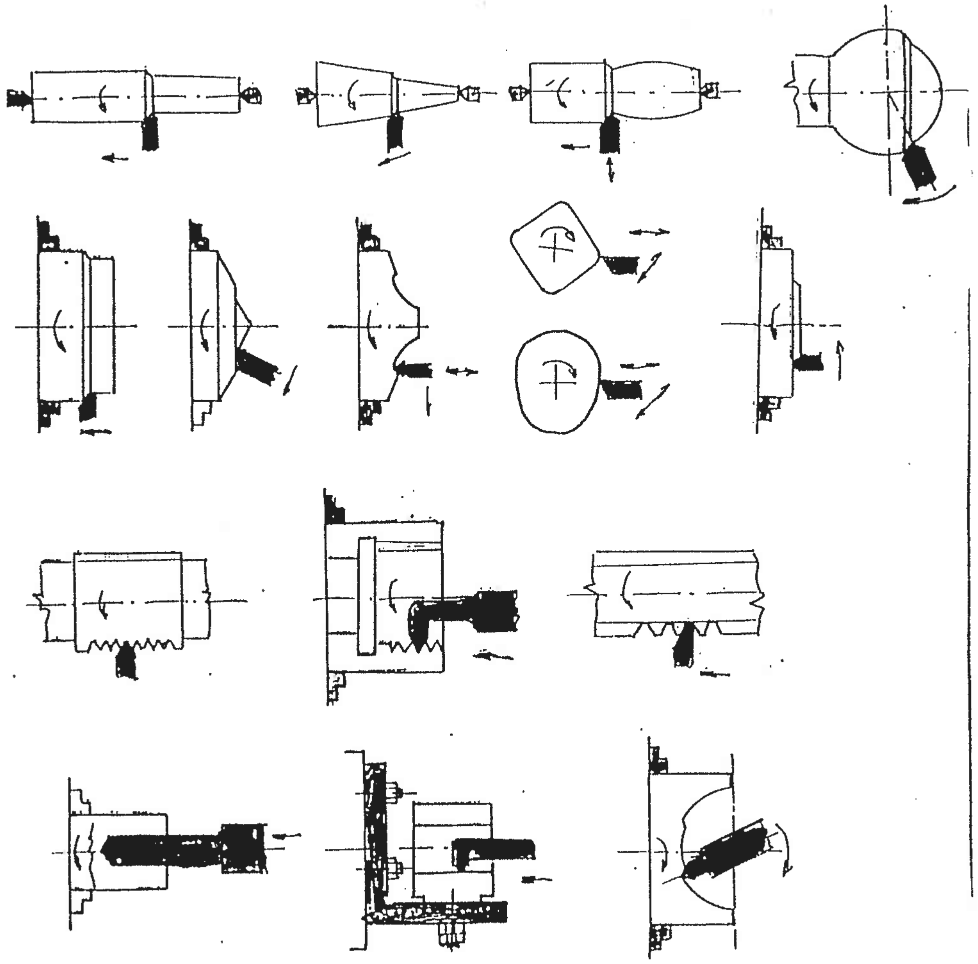
(a) Measuring
(b) Speed change
(c) Switching and locking
(d) Positioning and locking
(e) Lubrication.

(a) *Measuring*

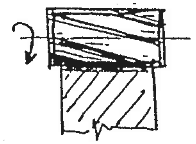
Optical instruments which serve for accurately measuring positioning movements have been used for many years and have been described in detail.[117] The main problem in their application lies in the need to fit them in such a manner that they form an organic whole with the machine to which they belong, 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.

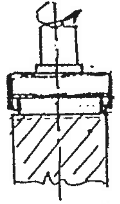
BASIC TURNING PROCESSES ON LATHES



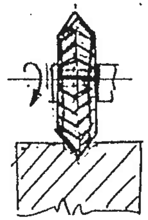
BASIC MILLING PROCESSES ON MILLING MACHINES



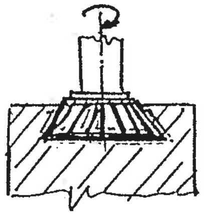
PLAIN



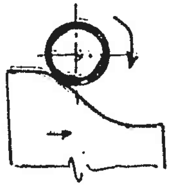
FACE



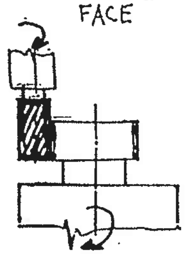
ANGLE



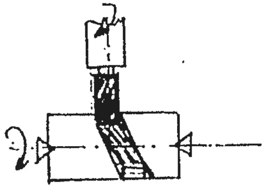
PRISMA



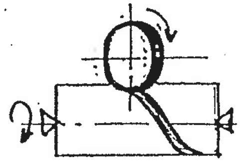
FORM, SURFACE



ROUND



WORM, THREAD



CAMS

EFFICIENT TOOL DURABILITY

$$T_E = (m_r - 1)\left(t_v + \frac{60 \cdot N_s}{M}\right) \text{ - } [\text{min}]$$

m_r - TOOL INFLUENCE GRADE
= 1.5 – 2.5 – ARTIFICIAL CORUND
3 – 6 – CEMENTED CARBIDE
6 – 8 – HIGH SPEED STEEL
8 – 10 – CARBON STEEL

t_v - [min] time of TOOL EXCHANGE AND SETTING
N_s - [\$] COSTS ON TOOL WEDGE
M - [-"-] COSTS OF MACHINE WORK (SALARY, MAINTANANCE,) (ENERGY, OVERHEADS)

CONVENTIONAL MACHINES $T_E \doteq 45 \div 90$ [min]
NC + CNC MACHINES $T_E \doteq 5 \div 15$ [min]

DEVELOPMENT TENDENCIES

RISE IN : P [kW] - MAIN DRIVE POWER $15 \div 80$ kW
n [min^{-1}] - SPINDLE SPEEDS 4000 - 8000 min^{-1}
s [mm min^{-1}] - FEEDS 200 - 1000 mm min^{-1}

OTHER FEATURES :

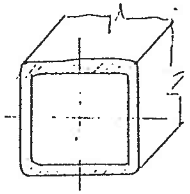
INFINITE SPEED VARIABILITY
TOOL OF HIGH POWER
TOOL DAMAGE INDICATION
HIGH EFFICIENCY OF COOLING, LUBRICATION, CHIP REMOVAL

STIFFNESS -

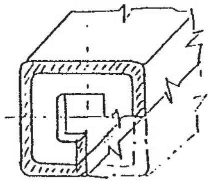
$$\text{STIFFNESS } C = \frac{\text{STATIC LOAD}}{\text{DEFORMATION}} = \frac{\text{FORCE}}{\text{DEFLECTION}} , \frac{\text{TORQUE}}{\text{ANGLE OF TWIST}}$$

RIGIDITY - DYNAMIC - BEHAVIOUR UNDER VIBRATION

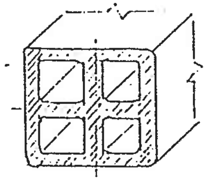
100 % - $C_B ; C_T = 1$
B - BENDING, T - TORQUE



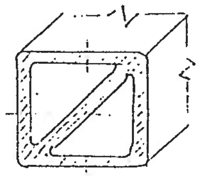
BASIC TYPE OF CROSS SECTION



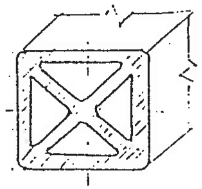
$C_B = 1,1$
$C_T = 1,3 \div 1,7$



$C_B = 1,2$
$C_T = 2,2$



$C_B = 1,5$
$C_T = 3$



$C_B = 1,78$
$C_T = 3,7$

OTHER POSSIBILITY OF STIFFNESS INCRESING :

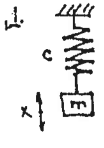
1. MATERIAL
   E - steel
   - cast Iron
   - composits
2. CROSS SECTION AREA
   $S = a \cdot b ;$
   ϕD
3. CONTACT - FINE SURFACE FINISH

PRE-LOADING

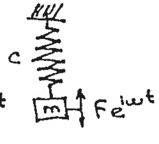
VIBRATIONS

$\Omega^2 = \frac{c}{m}$; $\vartheta = \frac{\varkappa}{2m}$ DAMPING COEFF. ; $\nu = \sqrt{\Omega^2 - \vartheta^2}$

1.



$$m\ddot{x} + cx = 0$$
$$x = \bar{X}_1 \cdot e^{i\Omega t} + \bar{X}_2 e^{-i\Omega t}$$

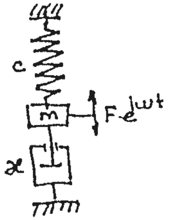


$$m\ddot{x} + cx = Fe^{i\omega t}$$
$$x = \underbrace{\bar{X}_1 \cdot e^{i\Omega t} + \bar{X}_2 e^{-i\Omega t}} + \frac{F}{c - m\omega^2}$$
$$x = \quad - \,'' \, - \quad + \frac{F}{c} \cdot \frac{\Omega^2}{\Omega^2 - \omega^2}$$

2.

$$m\ddot{x} + \varkappa\dot{x} + cx = 0$$
$$x = \bar{X}_1 e^{(-\vartheta + i\nu)t} + \bar{X}_2 e^{(\vartheta - i\nu)t}$$
$$-\vartheta \pm \sqrt{\vartheta^2 - \Omega^2} = -\vartheta \pm \nu$$
$$T = \frac{2\pi}{\nu}$$

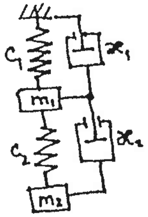
$\varkappa$ - DAMPING CONSTANT



$$m\ddot{x} + \varkappa\dot{x} + cx = Fe^{i\omega t}$$
$$x = \underbrace{\bar{X}_1 \cdot e^{i\Omega t} + X_2 \cdot e^{-i\Omega t}} + \frac{F}{c - m\omega^2 + \varkappa i\omega}$$
$$x = \quad - \cdot - \quad + \frac{\Omega^2}{\Omega^2 - \omega^2 + 2i\vartheta\cdot\omega} \cdot \frac{F}{c}$$

3.

$$m_1\ddot{x}_1 + c_1 x_1 + c_2(x_1 - x_2) = 0$$
$$m_2\ddot{x}_2 + c_2(x_2 - x_1) = 0$$
$$\frac{c_1}{m_1} = \Omega_{11}^2 \; ; \; \frac{c_2}{m_1} = \Omega_{21}^2$$

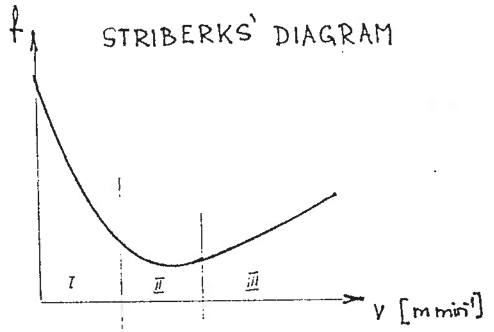


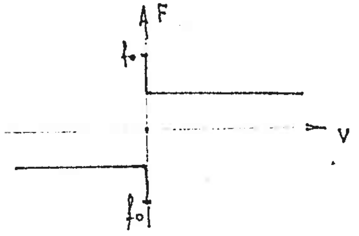
$$m_1\ddot{x}_1 + \varkappa_1\dot{x}_1 + c_1 x_1 + c_2(x_1 - x_2) + \varkappa_2(\dot{x}_1 - \dot{x}_2) = 0$$
$$m_2\ddot{x}_2 + \varkappa_2(\dot{x}_2 - \dot{x}_1) + c_2(x_2 - x_1) = 0$$
$$\vartheta_1 = \frac{\varkappa_1}{2m_1} \; ; \; \vartheta_2 = \frac{\varkappa_2}{2m_2}$$

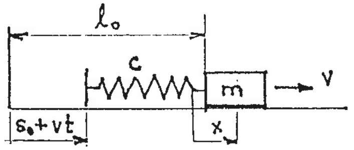
4.

STICK SLIP MOTION

STRIBERKS' DIAGRAM







$$m\ddot{x} + T - c\cdot(s_0 + vt - x) = 0$$
$$m\ddot{x} + cx = c\cdot s_0 - T + c\cdot vt$$
$$\ddot{x} + \frac{c}{m}\cdot x = \frac{\Delta T}{m} + \frac{c}{m}\cdot vt$$

$$\ddot{x} + \Omega^2 x = \frac{\Delta T}{m} + \Omega^2 \cdot vt$$

1. $\ddot{x} + \Omega^2 x = 0$; $x = \bar{X}\cdot e^{st}$

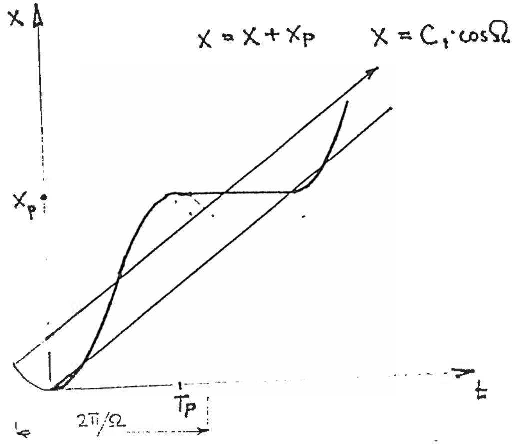
$$\dot{x} = s\cdot\bar{X}e^{st}$$
$$\ddot{x} = s^2\cdot\bar{X}e^{st}$$
$$s^2\cdot\bar{X}e^{st} + \Omega^2\bar{X}e^{st} = 0 \Rightarrow s_{1,2}$$
$$s_{1,2} = \pm i\Omega$$
$$x = C_1 \cdot \cos\Omega t + C_2 \sin\Omega t$$

2. $\frac{\Delta T}{m} + \Omega^2 \cdot vt = 0$; $a + bt = x_p$

$$b = \dot{x}_p$$
$$0 = \ddot{x}_p$$
$$\ddot{x}_p + \Omega^2 x_p = \frac{\Delta T}{m} + \Omega^2 \cdot vt$$
$$0 + \Omega^2(a + bt) = \frac{\Delta T}{m} + \Omega^2 \cdot vt$$
$$x_p = a + bt = \frac{\Delta T}{c} + v\cdot t \quad \rightarrow$$
$$a \,\hat{=}\, \frac{\Delta T}{c} \; ; \; b \,\hat{=}\, v$$

$$x = x + x_p \qquad x = C_1 \cdot \cos\Omega t + C_2 \sin\Omega t + \frac{\Delta T}{c} + vt$$

$$t = 0 \ldots x = 0 \; , \; \dot{x} = 0$$
$$C_1 = -\frac{\Delta T}{c} \; ; \; C_2 = -\frac{v}{\Omega}$$

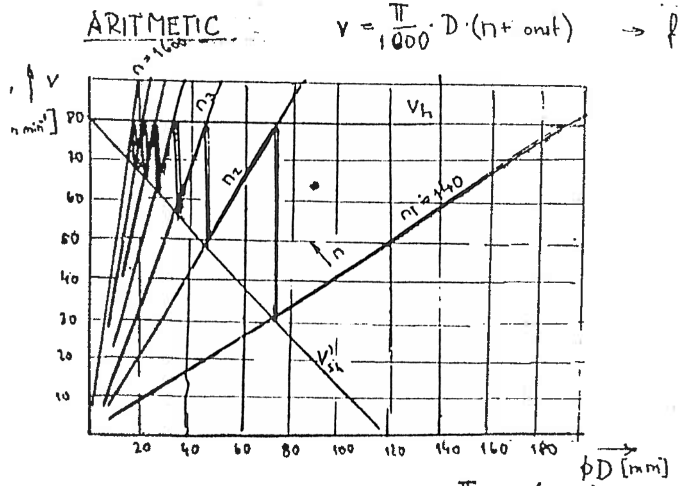


$$x = -\frac{\Delta T}{c}\cos\Omega t - \frac{v}{\Omega}\sin\Omega t + vt + \frac{\Delta T}{c}$$

PROGRESSIONS:

ARITMETIC

$$v = \frac{\pi}{1000} \cdot D \cdot (n + \text{onst}) \rightarrow f(n) = v - \underbrace{\text{ronst}}_{tg\,\kappa}$$



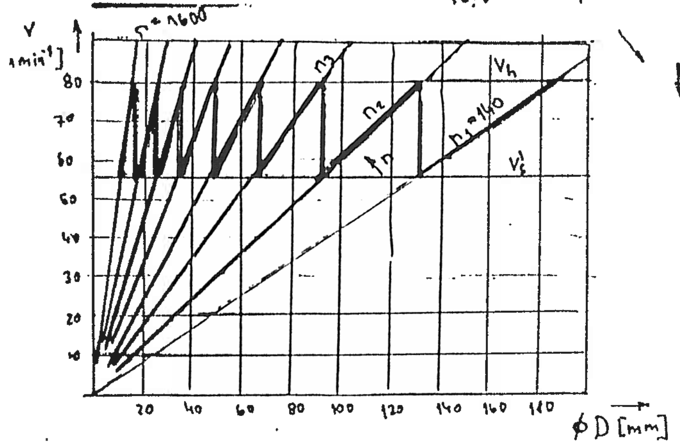
GEOMETRIC:

$$v = \frac{\pi}{1000} \cdot D \cdot (n \cdot \varphi)$$

$$- \varphi \triangleq R20 = \sqrt[20]{10} = 1{,}12$$

$$A_n = \frac{n_{max}}{n_{min}} = \varphi^{p-1} \Rightarrow \varphi = \sqrt[p-1]{A_n}$$

$$f(n) = v \cdot \frac{\text{konst}}{\varphi} = v \cdot \underbrace{\text{konst}}_{tg\,\alpha}$$

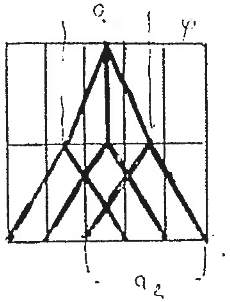


Designing a layout sliding gear diagram.
The possibility of alternatives are show on an example of gear box, for $p = 6$ spindle speeds (steps)

1. $p = 3 \cdot 2$

$a_1 = \varphi^2$

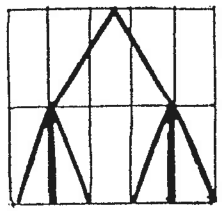
$a_2 = \varphi^3$



2.

$a_2 = \varphi^3$

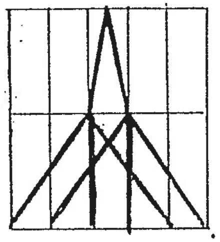
$a_1 = \varphi^2$



3. $p = 2 \cdot 3$

$a_1 = \varphi$

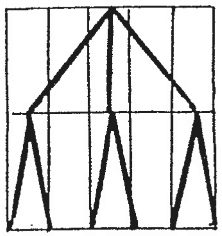
$a_2 = \varphi^4$



4.

$a_2 = \varphi^4$

$a_1 = \varphi$



Calculation - steps $p = k_1 \cdot k_2 \cdots k_r$

$$p_1 = k_1 \qquad A_1 = \underline{a_1} = \varphi^{p_1-1} = \varphi_1^{k_1-1} = \underline{\varphi^{k_1-1}}$$

$$p_2 = k_1 \cdot k_2 \qquad A_2 = a_1 \cdot a_2 = \varphi^{p_2-1} = \varphi^{k_1 k_2 - 1} = \varphi^{k_1-1} \cdot \varphi_2^{k_2-1}$$

$$\text{pro } \underline{a_2} \rightarrow \varphi_2^{k_2-1} = \varphi^{k_1 k_2 - 1 - (k_1-1)} = \underline{\underline{\varphi^{k_1(k_2-1)}}}$$

$$p_3 = k_1 \cdot k_2 \cdot k_3 \qquad A_3 = a_1 \cdot a_2 \cdot a_3 = \varphi^{p_3-1} = \varphi^{k_1 \cdot k_2 \cdot k_3 - 1} = \varphi^{k_1 k_2 - 1} \cdot \varphi_3^{k_3-1}$$

$$\text{Pro } \underline{a_3} \rightarrow \varphi_3^{k_3-1} = \varphi^{k_1 k_2 k_3 - 1 - \{k_1 k_2 - 1\}} = \underline{\underline{\varphi^{k_1 \cdot k_2 (k_3-1)}}}$$

$$p_r = k_1 \cdot k_2 \cdot k_3 \cdot \ldots \cdot p_r \, ; \; A_r = a_1 \cdot a_2 \cdot a_3 \cdot \ldots \underline{\underline{a_r}} = \varphi^{p_r-1} = \underline{\varphi^{k_1 \cdot k_2 \cdot k_3 \cdot \ldots (k_r-1)}}$$

GEAR BLOCKS WITH NOT MORE THAN THREE (3.) GEARS SHOULD BE USED.

IN ORDER TO SATISFY THE SMALLEST GEARBOX DIMENSION - (so the increase of TORQUE as even as possible) THE FOLLOWING CRITERIA SHOULD BE MET.

1. THE BLOCK SIZE SHOULD FOLLOW PROGRESSION

$$k_1 \geq k_2 \geq k_3 \; \dots \; \geq k_r$$

2. THE RANGE OF STEPS BETWEEN FOLLOWING SHAFTS SHOULD BE AS

$$a_1 < a_2 < a_3 \; \dots \; < a_r$$

EVERY RANGE CALULATED FROM

$a_1 = \varphi^{k_1 - 1}$

$a_2 = \varphi^{k_1 (k_2 - 1)}$

$a_3 = \varphi^{k_1 k_2 (k_3 - 1)}$ $\quad \dots\dots\dots \quad a_r = \varphi^{k_1 \cdot k_2 \cdot k_3 \cdot \dots (k_r - 1)}$

WITH THE SPEED v (n) DECREASING $n_{max} \to n_{min}$ THE TORQUE INGREASES. SO THE DIMENSIONS OF SHAFTS, BEARINGS, GEAR WHEELS GROW.

IT IS REASONABLE TO REALIZE EVEN, SMALL STEPS.

THE POWER $P\,[W] = F\,[N] \cdot v\,[ms^{-1}] = \text{constant}$

$$v = \frac{\pi \cdot D\,[mm] \cdot n\,[r\,min^{-1}]}{1000 \cdot 60} = [m\,s^{-1}] \quad ; \quad M = \frac{F \cdot D}{2}$$

$$P = \frac{F \cdot D}{2} \cdot \frac{n \cdot \pi}{1000 \cdot 30} = M \cdot \frac{n}{9550} = [W] \quad -$$

$$M = 9550 \cdot \frac{P\,[W]}{n\,[r\,min^{-1}]} \quad [Nm]$$

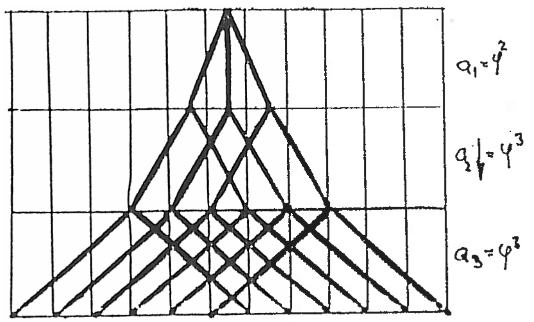
$; \; M_1 = CONST \cdot \dfrac{P}{n_1}$

$M_2 = CONST \cdot \dfrac{P}{n_2} = C \cdot \dfrac{P}{\frac{n_1}{\varphi}} = M_1 \cdot \varphi$

$M_3 = M_1 \cdot \varphi^2$

$\vdots$

THE POSSIBILITY OF GEAR LAYOUT FOR 12 STEPS



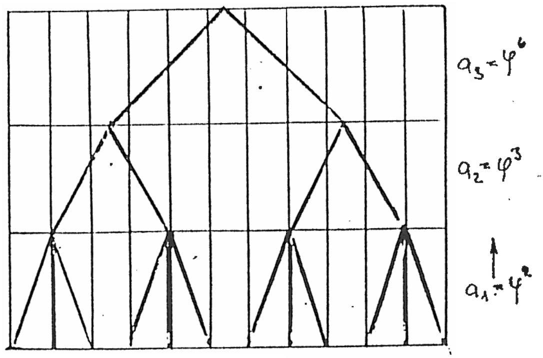
$p = 12 = 3 \cdot 2 \cdot 2$

$a_1 = \varphi^2$

$a_2 = \varphi^3$

$a_3 = \varphi^6$

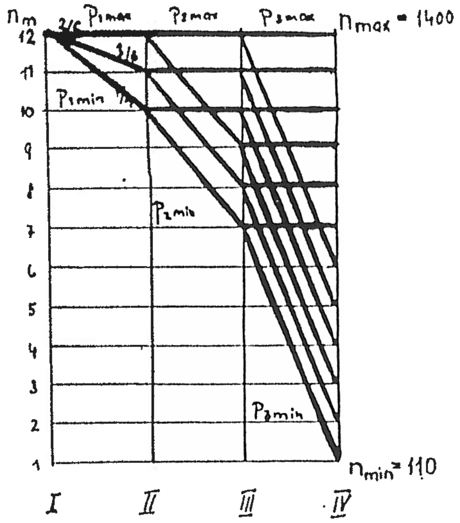
$A_1 = \varphi^2 \quad ; \quad A_2 = \varphi^5 \quad ; \quad A_3 = \varphi^{11}$



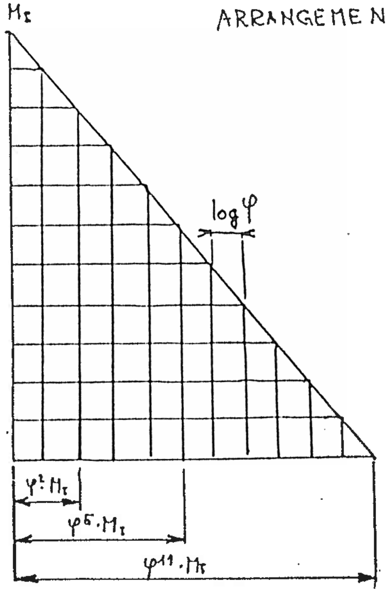
$p = 12 = 3 \cdot 2 \cdot 2$

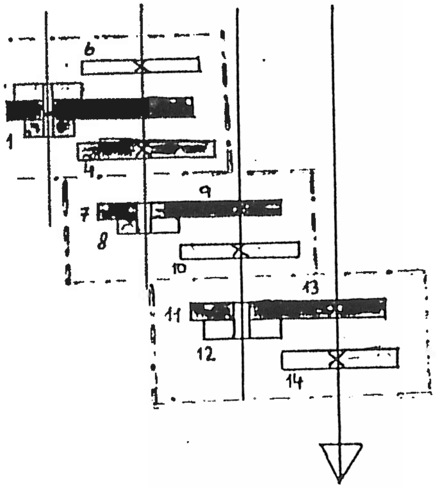
$A_1 = a_3 = \varphi^6 \quad ; \quad A_2 = a_3 \cdot a_2 = \varphi^9 \quad ; \quad A_3 = a_3 \cdot a_2 \cdot a_1 = \varphi^{11}$

SPEED DIAGRAM ;



TORQUE DISTRIBUTION & BASIC DRAWING OF POSSIBLE KINEMATIC ARRANGEMENT





GEAR WHEELS CALCULATION :

SELECT → $Z_{min} = Z_1 = 20$

$$p_{1min} = \frac{Z_1}{Z_4} = \frac{1}{\varphi^2} \Rightarrow Z_4 = Z_1 \cdot \varphi^2$$

$$Z_1 + Z_4 = 20 + 20\varphi^2 = 20(1+\varphi^2)$$

$$Z_2 = Z_5$$

$$Z_2 + Z_5 = 20(1+\varphi^2) \Rightarrow Z_2 = Z_5 = 10(1+\varphi^2)$$

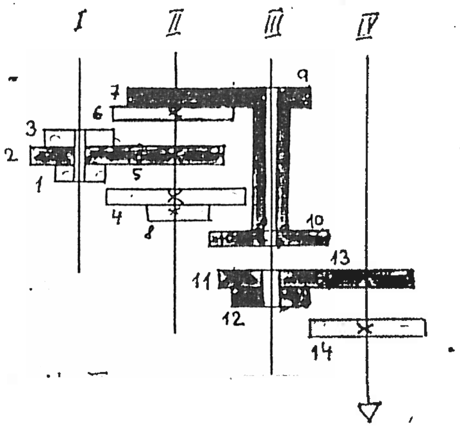
$$\frac{Z_3}{Z_6} = \frac{1}{\varphi} \quad ; \quad Z_3 + Z_6 = 20(1+\varphi^2)$$

$$\underset{\frac{1}{\varphi} \nearrow}{\frac{Z_3}{Z_6}} + 1 = \frac{20}{Z_6}(1+\varphi^2) \Rightarrow Z_1.$$

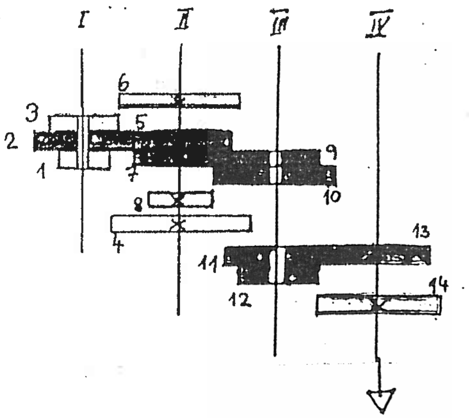
SELECTION CHOICES of Z_{min} :

$Z_{min} = 16$ - spindels ; $Z_{min} = 12 \div 14$ - feeds

RECOMENDED ! $Z_{min} = 18 \div 20$



1. CASE - SLIDING GEARS ARE PUT ON ONE SHAFT (III) ONLY



2. CASE - TWO GEAR WHEELS (5+7) ARE JOINED INTO ONE

LIMITS OF TRANSMITION RANGE -

$p_{max} = 2 \div 4$; $p_{min} = \frac{1}{4} \div \frac{1}{6}$

LIMIT OF THE REAL SPEEDS DEVIATION

$$\Delta = \frac{n_{Te} - n_{Re}}{n_{Te}} \cdot 100 \; [\%] \quad - \quad \pm 2\%$$

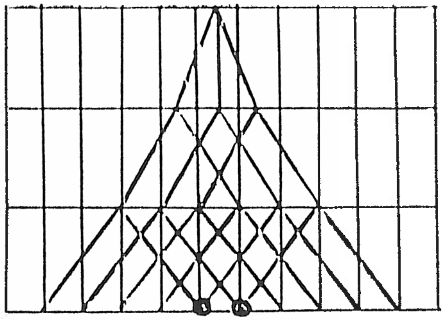
OVELAPING GEARS - x (The last number of steps is smaler of x steps)

$$A_x = \varphi^{p-1-x} \quad ; \quad a_x = \varphi^{k_1 \cdot k_2 \cdot\cdot (k_x - 1) - x}$$

SYMETRICAL PROGRESSION - WITH y - steps FREE ON THE OUTSKIRTS

$$A_y = \varphi^{p-1+2y} \quad ; \quad a_y = \varphi^{2(y+\frac{1}{2})}$$

EXAMPLES OF GEARS LAYOUT FOR p = 12



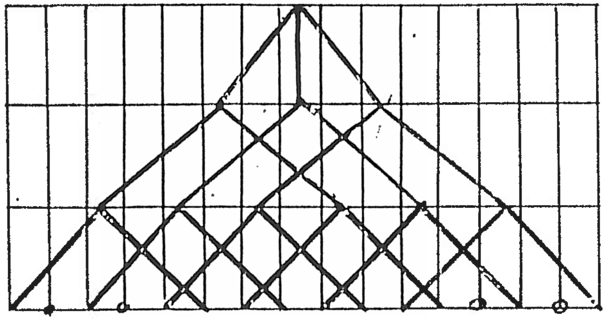
OVERLAPED STEPS

X = 2

$a_1 = \varphi^2$

$a_2 = \varphi^3$

$a_3 = \varphi^{2\cdot 3(2-1)-X} = \varphi^4$



SYMETRICALLY

LEFT FREE

Y = 2

$a_1 = \varphi^{2(k_1-1)} = \varphi^4$

$a_2 = \varphi^{2k_1(k_2-1)} = \varphi^6$

$a_3 = \varphi^{2(y+\frac{1}{2})} = \varphi^5$

GEAR WHEELS CALULATION

INFORMATIVE MODULUS CALCULATION

$$m = 10\sqrt[3]{\frac{2\,M_k}{z\cdot C\cdot\psi\cdot\pi}} \quad [mm]$$

M_k - [Nm] TORQUE

z - [NUMBER OF TEETH

C - $[MPa]\cdot 10^6$

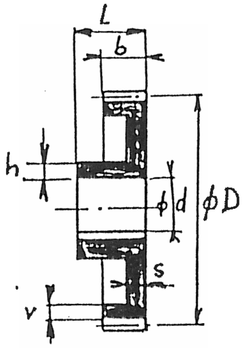
I. C = 10 ÷ 20 [MPa] - FIRST SHAFT

II C = 30 ÷ 45 [MPa] - LAST SHAFT

LAST C = 50 ÷ 60 [MPa) - GREAT STEP RANGE

GEAR WHEELS - DESIGN RECOMENDATIONS

1. TRASMITION RANGE $p = \frac{1}{4} \div 4$ in M.T. 97%
2. TEETH NUMBER $z = 20 \div 40$ in M.T. 83 %
3. MODULUS SIZE $m = 2.5 \div 5$ in M.T. 90 %
4. WIDTH RATIO $\psi = \frac{b}{m} = 4.3 \div 13$ → 6+8
5. MATERIAL CZ STANDARD - 14 220.9 90%
   C (0,14 ÷ 0,19) ; Mn (1,1 ÷ 1.4) ; Si (0.17 ÷ 0.37) ; Cr (0.8 - 1,1)
   DIN STANDARD - 1.7131
6. SPECIFIC RECOMENDINGS -
   v → over 4 $m s^{-1}$ - HARDENED, ROUND SURF.
   v ← up to 10 $m s^{-1}$ - STRIGHT GEARS
   v – over 20 up to 30 $m s^{-1}$ - HELICAL GEARS

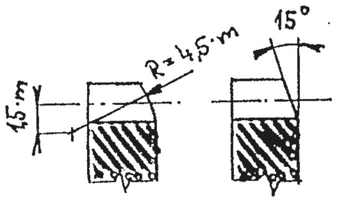


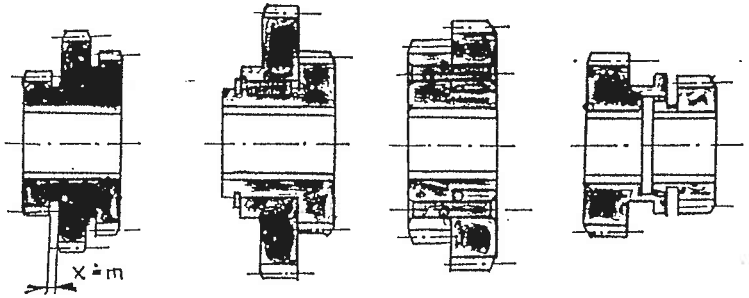
$v = 3\cdot m$ -[modulus]

$s = \frac{D}{20} + 5$

$h = (3 \div 4)\cdot m$ -[modulus]

$L = (0{,}75 + 1{,}5)\cdot d$ [shaft diameter]





SHAFTS DIMENSIONING - RECOMENDATIONS

FIRST DESIGN APPROACH :

BASIC CRITERIA - MAX. SHAFT DEFLECTION (IN BENDING)

$y \leq 0.0002 \cdot L$ - shaft length [mm]

$y \leq 0.035$ - absolute [mm]

$y \leq 0.01 \cdot m$ - modulus min [mm]

BENDING DEFLECTION IS CONSIDERED BECAUSE IT CAUSES

1. WORSTENING OF GEARING - INCREASE OF LOAD ON TOOTH EDGES, INCREASE OF WEAR, NOISE
2. CREATION OF ADDITIONAL AXIAL FORCE - PUSHING GEARS OUT OF GEARING
3. INCREASE OF LOAD ON EDGES IN PLAIN BEARINGS, PRESSURE GROWS WORSTENING LUBRICATION, GREATER WEAR, COLAPS OF BEARING
4. IN CASE OF ROLLER BEARINGS ADDITIONAL LOAD, ITS DURABILITY DECREASING.
5. VIBRATIONS, WHEN LOWER STIFFNES → GREAT AMPLITUDES CAUSE CLASHES, WEAR, NOISE

CALULATION CONDITIONS - SIMPLIFICATIONS

1. ROTATIONS DECREASE - FROM INPUT TO OUTPUT SHAFT

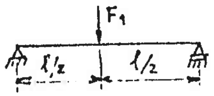
$n_1 > n_2 > n_3 > \dots > n_n$ THEN FORCES AND TORQUE INCRESE

$F_1 < F_2 < F_3 < \dots < F_n$

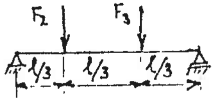
$M_1 < M_2 < M_3 < \dots < M_n$

2. FIRST SHAFT IS LOADED BY ONE FORCE, NEXT BY TWO AS SHOWN

I.



II



3. THE SHAFTS ARE CONSIDERED AS LOADED IN ONE PLAIN,
4. THE DIAMETERS OF PINIONS ARE ON ALL SHAFTS THE SAME
5. THE SHAFTS ARE CONSIDERED AS EQUALY LONG PRISMATIC BARS
6. DEFLECTIONS ARE CALCULATED IN POSITION UNDER ACTING FORCES

$M_1 \mathrel{\hat{=}} F_1$

$M_2 \mathrel{\hat{=}} M_1 \cdot a_1 \doteq M_1 \cdot \varphi^2 \mathrel{\hat{=}} F_2 \quad \Rightarrow \quad F_2 = F_1 \cdot \varphi^2 = F_1 \cdot a_1$

$M_3 = M_1 \cdot a_1 a_2 = M_1 \cdot \varphi^5 \mathrel{\hat{=}} F_3 \quad \Rightarrow \quad F_3 \doteq F_1 \cdot \varphi^6 = F_1 a_1 a_2$

$M_4 = M_1 \cdot a_1 a_2 a_3 = M_1 \cdot \varphi^{11} = F_4 \quad \Rightarrow \quad F_4 = F_1 \cdot \varphi^{11} = F_1 a_1 a_2 a_3$

GENERAL BAR DEFLECTION $y = \frac{F \cdot l^3}{\alpha \cdot E I}$; $I = \frac{\pi \cdot d^4}{64}$ =

$d^4 = \frac{64 \cdot F \cdot l^3}{y \cdot \pi \cdot E \alpha}$ substituting $y \leq 2 \cdot 10^{-4} \cdot l$ - →

$$d_1 = \sqrt[4]{const} \cdot \sqrt[4]{F \cdot l^2}$$

$$d_2 = \underbrace{\sqrt[4]{const}}_{\beta} \cdot \sqrt[4]{F l^2} \cdot \sqrt[4]{a_1 + \frac{7}{8}}$$

$$d_3 = \beta \cdot \sqrt[4]{F \cdot l^2} \cdot \sqrt[4]{a_1 (a_2 + \frac{7}{8})}$$

$$d_4 = \beta \cdot \sqrt[4]{F l^2} \cdot \sqrt[4]{a_1 a_2 (a_3 + \frac{7}{8})}$$

TAKING POWER or by substituting $F = const \cdot \frac{P}{D \cdot n}$

$$d = \sqrt[4]{const} \cdot \sqrt[4]{\frac{l^2}{D}} \cdot \sqrt[4]{\frac{P}{n}} \quad \text{or} \quad d = \sqrt[4]{const} \cdot \sqrt[4]{M_k}$$

3,56 1,8

FOR MOST COMMON DIMENSIONS, $L = 150 \div 300$ [mm] ; $D = 47 \div 90$ [mm]

$\sqrt[4]{\frac{L^2}{D}} \doteq 2{,}25 \div 3{,}73$

$$d \doteq 10 \sqrt[4]{\frac{P}{n}} \; \frac{[kW]}{[min^{-1}]} \; [cm]$$

STEPPED DRIVES. - BASIC SYSTEMS

1. EXCHANGABLE GEAR-WHEELS }
2. SLIP GEARS
3. SLIDING GEAR BLOCKS
4. CLUTCHES (DOG ; FRICTION)
5. BACK GEARS
6. NORTON TYPE GEARBOX
7. "MEANDER GEARBOX
8. DRAW-KEY TYPE GEARBOX

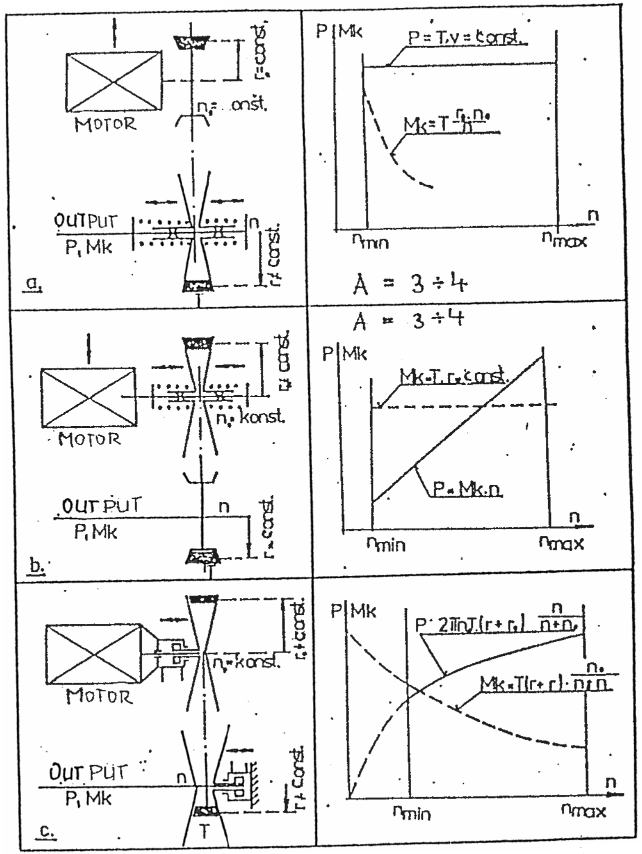
DESIGN RECOMENDATIONS :

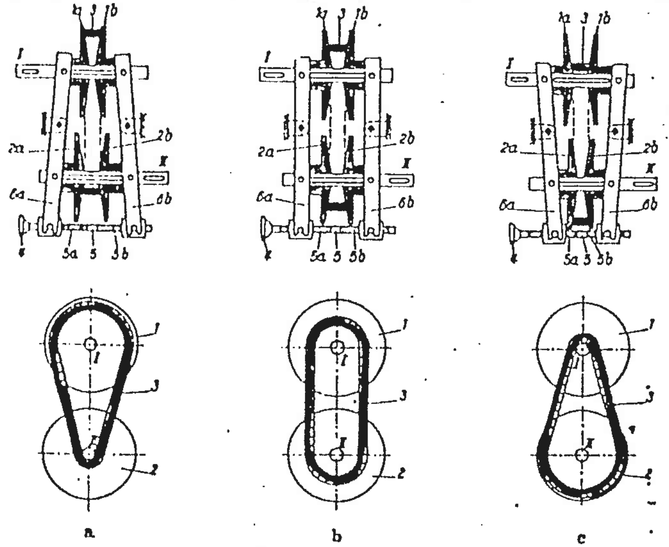
1. USE MINIMUM NUMBER OF GEAR WHEELS AS POSSIBLE
2. USE MINIMUM NUMBER OF SHAFTS AS POSSIBLE
3. APPLY MINIMUM LENGTH OF SHAFTS AS POSSIBLE

STEPLESS DRIVES -

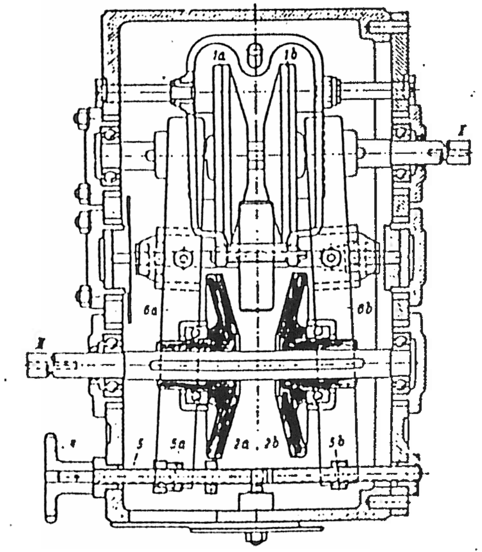
1. FRICTION DRIVES
2. POSITIVE INFINITELY VARIABLE DRIVES (P.I.V.)

BELT P.I.V. DRIVES

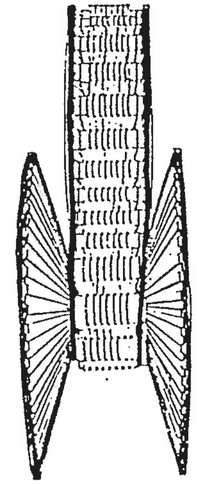


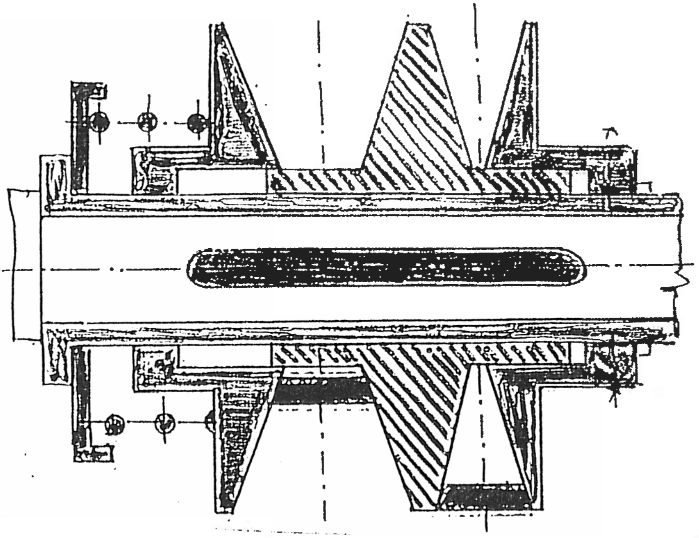


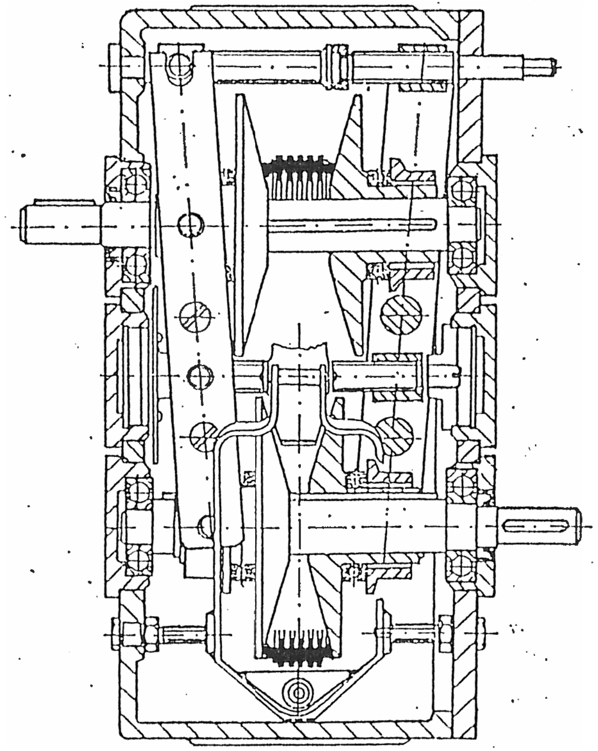
Working principle of the P.I.V. drive. a—Speed increase; b—1:1 transmission; c—Speed reduction.

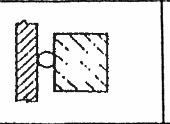
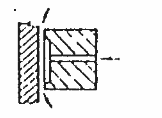
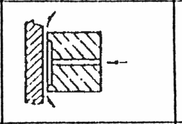
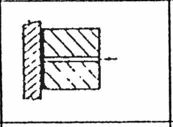
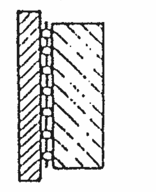
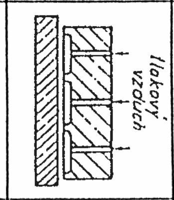
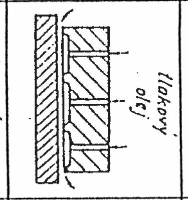
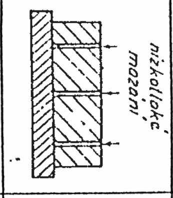
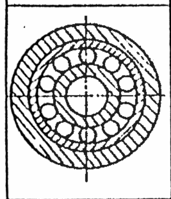
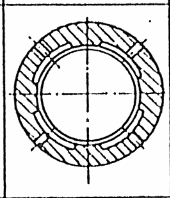
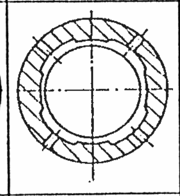
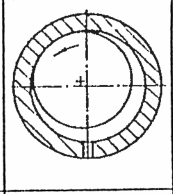
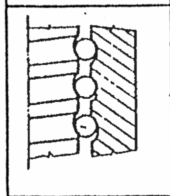
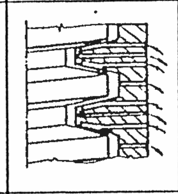
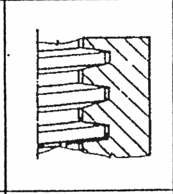
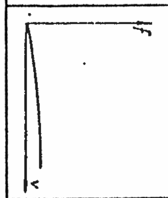
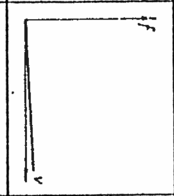
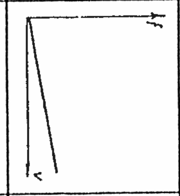
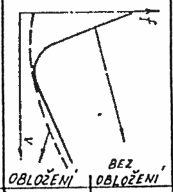


P.I.V. drive: *I*—Driving shaft (input); *II*—Driven shaft (output).







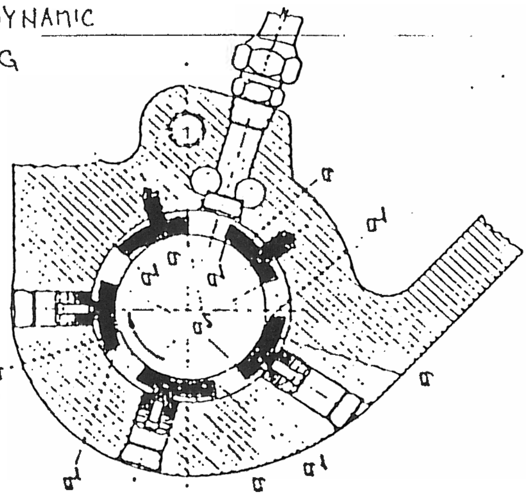
ROLLING	AIR-PNEUST.	HYDROSTATIC	FRICTION		TYPE		
					ELEMENT		
					LINEAR CIRCULAR	SLIDE	MOVABLE JOINTS
					ROTATIONAL	SUPPORT	
	not used yet				SCREW + NUT WORM + RACK		
					FRICTION		
0.001 - 0.005	$1\cdot10^{-7}$	$5\cdot10^{-6}$	0.01 ÷ 0.1	0.12 ÷ 0.3	FRICTION COEFFICIENT		
No	No	No	No	YES	STICK-SLIP		
GREAT	SMALL	GREAT	GREAT	GREAT	STAT. STIFFNESS		
SMALL	VERY SMALL	VERY SMALL	GREAT	GREAT	IN MOVEMENT D.	DAMPING	
SMALL	FAIR	GREAT	GREAT	GREAT	IN LOAD DIR.		
GOOD - GREAT	GREAT	GREAT	SMALL	SMALL	LIFE - TIME		
GOOD - GREAT	GREAT	GREAT	GOOD	GOOD	ACCURACY		
NO	NO	NO	YES	YES	GAP		
NO	NO	NO	NO	YES	WELDING TENDENCY		

SPEED v [$m\,s^{-1}$] VALUES IN TURNING

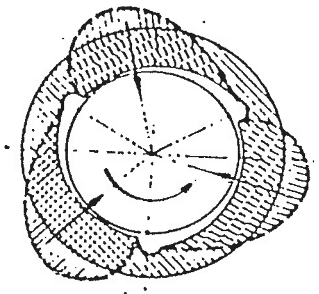
FEED s [$mm\,rot^{-1}$]

MACHINED MATERIAL	TOOL MATERIAL											
	HSS- HIGH SPEED STEEL				CC- CEMENTED CARBIDE				CERAMICS			
	ROUGHING		FINISHING		ROUGHING		FINISHING		ROUGHING		FINISHING	
	v	s	v	s	v	s	v	s	v	s	v	s
STEEL UP TO 500 [MPa]	0,30 - 0,40	0,8 to 1,6	0,50-0,75	to 0,2	0,65-1,35	0,8 to 2	3,30-6,65	to 0,2	4,15-8,30	0,2 to 0,6	5,00-10,0	to 0,2
STEEL 500 ÷ 700 [MPa]	0,25-0,30		0,40-0,60		0,60-1,30		2,50-4,15		3,30-5,8		4,15-6,6	
STEEL 700 ÷ 850 [MPa]	0,20-0,35		0,25-0,30		0,40-0,80		1,65-3,00		2,50-4,15		3,00-5,0	
STEEL 850 ÷ 1000 [MPa]	0,13-0,20		0,15-0,25		0,30-0,50		1,40-2,30		1,65-2,50		2,30-3,15	
STEEL OVER 1000 [MPa]	0,10-0,13		0,13-0,20		0,30-0,40		1,15-1,65		-		-	
CAST IRON UP TO 220 [HB]	0,25-0,30		0,30-0,40		0,10-0,15		1,15-1,65		1,30-2,50		2,00-3,30	
Cu ALLOYS	0,30-0,65	to 2	0,50-0,80	to 0,2	1,65-3,30	to 2	2,50-5,00	to 0,2	1,65-3,00	to 0,1	5,00-8,30	to 0,1
Al ALLOYS	1,65-3,30		1,65-3,30		1,65-16,5		2,50-16,5		-		-	

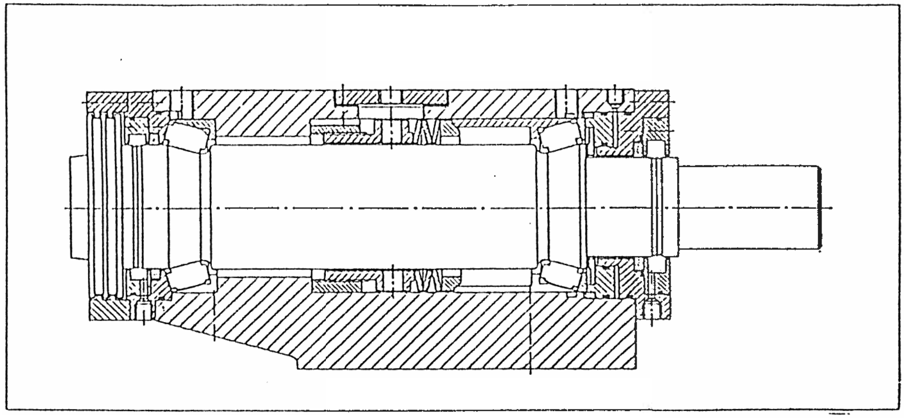
HYDRODYNAMIC BEARING



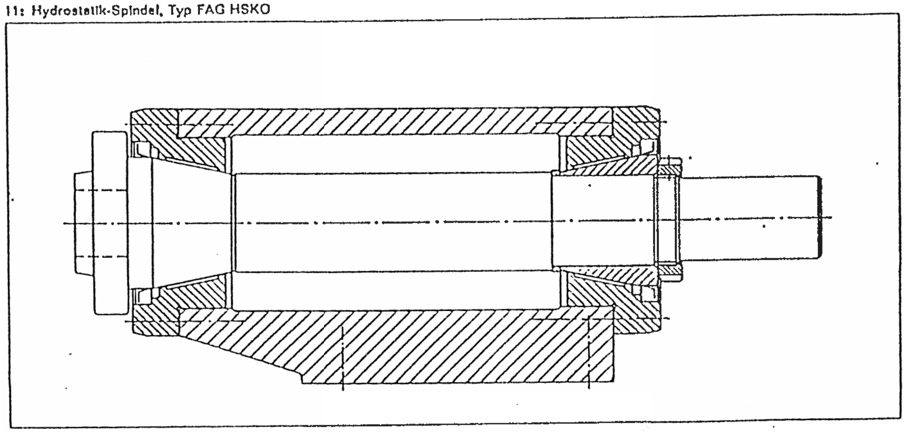
A PRESSURE DISTRIBUTION

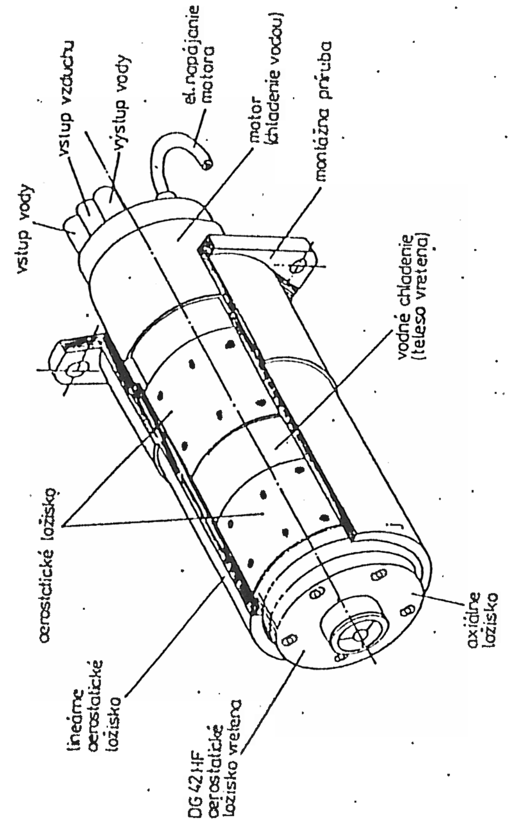


10: Hydrodynamik-Spindel, Typ FAG HDKO



11: Hydrostatik-Spindel, Typ FAG HSKO





ROLLING BEARINGS

PRECISSION - STANDARD - P0

P6 - HIGHER

P5 - } SPECIAL

P4 - }

GAP; C1, C2 ; .... C5 ;

LOWER NOISE. - C6

SPECIAL - C7, 8, 9

MAX. ROT $n_{MAX} \cdot d_s \leq (1 \div 1{,}5) \cdot 10^5$ [mm min^{-1}]

$d_s \rightarrow \frac{D+d}{2}$

BEARINGS - BALL ; ROLLER ; NEEDLE ;

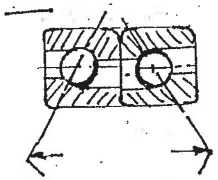
ONE ROW ; DOUBLE ROW

ANGLE β - 15° ÷ 25° TYPE 70... ; 72... ; 73... ONE ROW

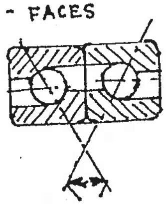
β - 32° TYPE 32... ; 33 DOUBLE ROW

PAIR COMBINATIONS :

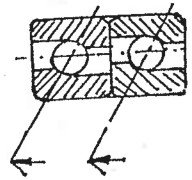
"O" - BACKS



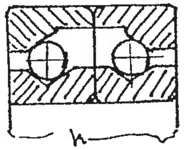
"X" - FACES



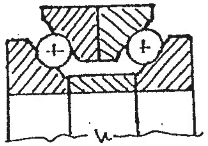
"T" - FOLLOW



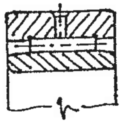
EX-CELL-O



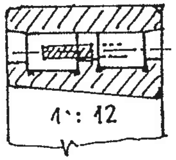
SKF. - AXIAL β = 60°



NEEDLE NA 49.. V



DOUBLE ROLLER

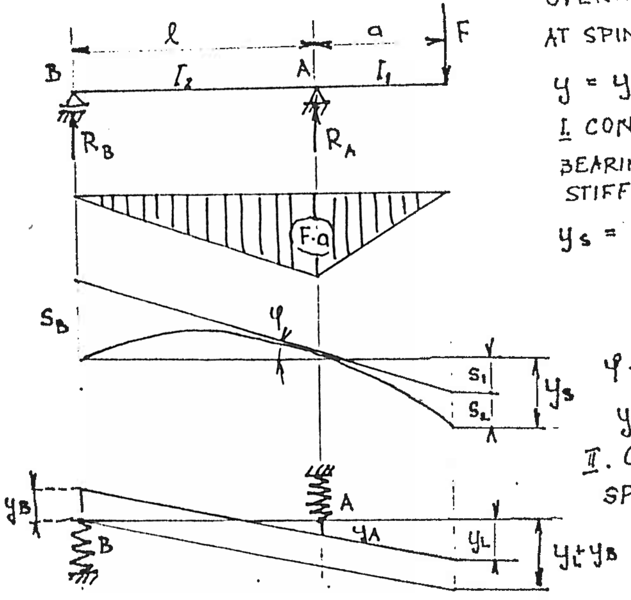


NN 30... K

/SP - SUPER PRECISSION

/UP - ULTRA PRECISSION

SPINDLE SUPPORT - OPTIMUM BEARING DISTANCE



OVERAL DEFLECTION
AT SPINDLES' END

$$y = y_s + y_L \; ; \; R_A = F\left(1+\frac{a}{l}\right)$$

$$R_B = -F\frac{a}{l}$$

I. CONSIDERED - BEARING SUPPORT INFINITE STIFF

$$y_s = S_1 + S_2$$

$$S_1 = \varphi \cdot a$$

$$S_2 = \frac{F \cdot a^3}{3EI_1}$$

$$\varphi = \frac{S_B}{l} \; ; \; S_B = \frac{Fa \cdot l^2}{3EI_2}$$

$$y_s = \frac{Fa^2}{3E}\left(\frac{l}{I_2} + \frac{a}{I_1}\right)$$

II. CONSIDERED SPINDLE BAR INFINITE STIFF

$$\frac{y_L + y_B}{l+a} = \frac{y_A + y_B}{l}$$

$$y_L = y_A \cdot \left(1+\frac{a}{l}\right) + y_B\left(\frac{a}{l}\right)$$

$$y_A = R_A / C_A \; ; \; y_B = R_B / C_B$$

$$\frac{1}{C} = \frac{7{,}68 \cdot 10^{-5} \cdot Q^{0.9}}{\cos\alpha \cdot l_m}$$

$$Q = \frac{5 \cdot R}{z \cdot \cos\alpha}$$

$$z = q_2\left(\frac{D+d}{d_o}\right)$$

$$d_o = q_1(D-d)$$

$l_m =$

$q_1 ; q_2$ - see TABLE

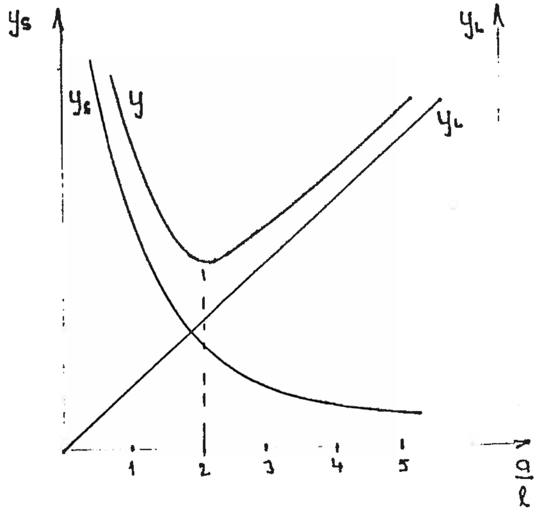


TABLE - BEARING COEFFICIENTS

BEARING	q_1		q_2	
RADIAL BALL - 1 row	0,216	0,33	0,99	0,89
BALL - 2 rows	0,2	0,28	1,39	1,19
ONE ROW ANGLE	0,25	0,32	1,4	1,24
DOUBLE ROW ANGLE	0,241	0,29	1,48	1,25
INCLINING	0,217	0,238	1,33	1,07
ROLLER	0,205	0,257	1,24	0,97
BARREL	0,259	0,289	1,36	1,15
BARREL	0,233	0,278	1,4	1,15
	0,22	0,28	1,6	1,3
NEEDLE - NO CARRIBE	0,13	0,21	1,57	1,57
—n— WITH	0,13	0,21	0,78	1,0
AXIAL				
BALL	0,318	0,386	1,42	1,19
BARREL	0,237	0,253	1,12	1,07
ANGLE (BALL)	0,34	0,38	1,41	1,23
ROLLER	0,27	0,35	0,85	1,2

1. Bearing systems for machine tool spindles

The spindle can be considered to be the heart of the machine tool and the quality of the spindle has a major influence on machine performance.

The most common requirements for a spindle are:

- High running accuracy
- High speed capability
- Great stiffness
- Low and even running temperature
- Minimum need of maintenance

Some of these requirements conflict with each other and it is thus not possible to achieve the best of everything in one and the same spindle. Therefore, when designing a spindle, a careful evaluation as to which of the parameters are most important has to be made. It may be necessary to know the end user´s requirements as a machine might be used in different ways by different end users. Designing the machine in a way that makes it possible to accept alternative spindles may be to advantage in many cases.

A range of bearing types and arrangements are available in order to meet the requirements of the various parameters. The major types of spindle bearings are shown in Fig. 2, which also illustrates the relative proportions of the bearings.

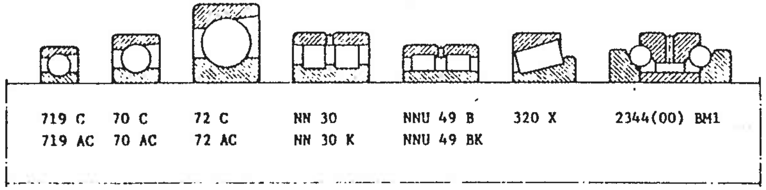


Fig. 2 Major types of spindle bearings

1.1 Bearing system I

A combination of two double row cylindrical roller bearings of series NN 30 K and one angular contact thrust ball bearing of series 2344(00) gives a very stiff bearing system. The common characteristics of these bearing series are that they have the same low sectional height and that they have a large number of rolling elements.

Their low sectional height makes the spindle design compact, i.e. permits the use of a large diameter without the spindle external dimensions being too big. The large number of rolling elements is a factor that is essential for the great stiffness of these bearings, accentuated by the line contact of series NN 30 K and the wide contact angle for series 2344(00). These two bearings are normally mounted at the work side of the spindle with another NN 30 K at the opposite side. The bearings of series NN 30 K have a tapered bore in order to enable accurate adjustment of the preload on mounting. With the help of a distance sleeve, bearings of series 2344(00) are accurately preadjusted to the proper preload in the factory.

The characteristic features of this bearing system, in addition to its great stiffness, are that radial and axial loads are supported individually by the bearings and that all clearance in the bearings and between the bearings and the housing can be eliminated as thermal displacements can be accomodated inside the cylindrical bearings. This system is suitable for spindles working in a wide speed range and is commonly used for medium sized and heavy lathes, machining centers, fine boring machines and surface grinders. Fig. 3 shows a spindle for a CNC lathe with bearings NN 3034 K/SPW33 and 234434 BM1/SP at the work side and NN 3032 K/SPW33 at the drive side.

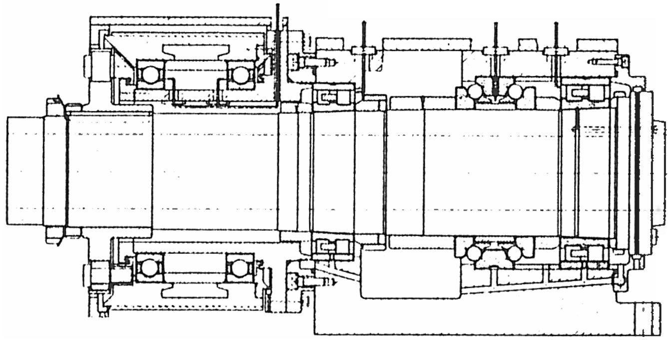


Fig. 3 Spindle for CNC lathe

The latest generation of series NN 30 K bearings has for the most common sizes been equipped with separate polyamide 6,6 cages for each row of rollers. The properties of this cage enable cooler and quiter running of the bearings, Fig. 4. The latest generation of series 2344(00) bearings has a cage of a design that enables the bearings to be run at higher speeds than before with grease lubrication, Fig. 5.

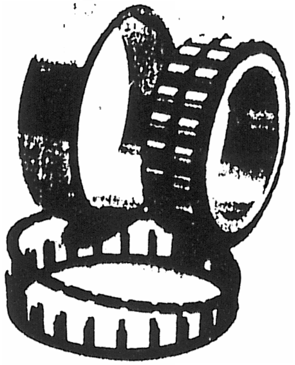


Fig. 4 Bearing NN 30 K with polyamide 6,6 cages - designation TN9 and TN

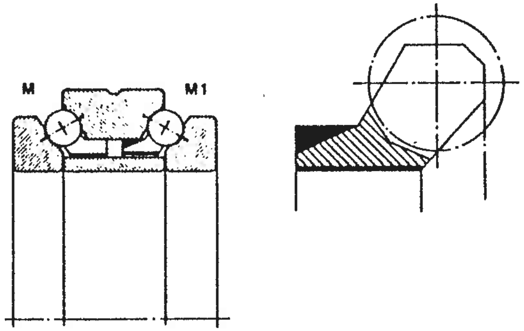


Fig. 5 Bearing 2344(00) with cage allowing high speeds - designation H1

Another series of double row cylindrical roller bearings, NNU 49 BK, with lower sectional height than NN 30 K is used especially for large spindles, in order to reduce the external dimensions. In addition to the difference in sectional height, the design differs in that the series NN 30 K bearings have the guiding flanges on the inner ring and the series NNU 49 BK have them on the outer ring.

1.2 Bearing system II

If higher speeds than those that can be obtained with the bearing system I are required then angular contact ball bearings are used. These bearings can be arranged in many different ways, the arrangement is determined by the stiffness and speed requirements. When great stiffness is necessary, though not as great as with bearing system I, then an arrangement with three angular contact ball bearings is used at the work side. These bearings are available either as a matched set or as single universally paired bearings, in both cases with three different degrees of preloads.

Spindles requiring a set of three bearings normally also need good axial stiffness and axial load carrying capacity and therefore bearings with a contact angle of 25^{o} are most commonly used for this arrangement.

A bearing of series NN 30 K is commonly used at the opposite side of the spindle so that thermal displacement can be accomodated inside the bearing.

High speed lathe and machining center spindles are typical applications for this bearing system. Fig. 6 shows a high speed spindle for a vertical machining center using bearing system II. The bearings at the work side are a matched set 7013 AC/P4TBTB and the bearing at the drive side is NN 3011 K/TNSPW33.

1.3 Bearing system III

When the stiffness requirement is not so stringent, and higher speed is more important, bearing systems with only one or two angular contact ball bearings at each side of the spindle are used.

A typical system is shown in Fig. 7 consisting of a set of two bearings arranged back-to-back on each side of the spindle. The bearing set on the drive side is then free to float axially. This bearing system is commonly used for light, high speed lathes and grinding machines. The spindle shown in the figure is for a small surface grinding machine and the bearing set on each side is

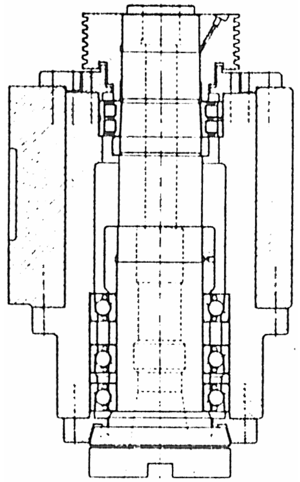


Fig. 6 Spindle for vertical machining center

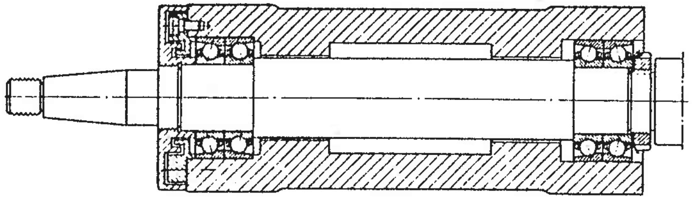


Fig. 7 Spindle for a surface grinding machine

7006 C/P4DBA. For this kind of spindle the axial carrying capacity and stiffness is not so important and bearings with a contact angle of 15° can be used. If the spindle is to be subject to heavy axial loads it may be preferable to have bearings with 25° contact angle at the work side.

Bearing systems with one angular contact ball bearing at each side of the spindle are used when the highest speeds are required. The spindles will not be as stiff, but as spindles with this system are supposed to work at more or less constant high speeds with light loading the stiffness is not of major importance. Normally bearings with 15° contact angle are used and in order to ensure maintenance of the proper bearing preload, and without risking neither too heavy a preload or clearance in the bearings, the system is preloaded by a spring. The spring then acts on the outer ring of one of the bearings, normally the drive side bearing. When there is a requirement for higher load carrying capacity and/or greater stiffness than can be obtained by one bearing at each side, then two bearings in tandem can be applied on each side. This extremely high speed system is typically used for internal grinding machine spindles, Fig. 8. This spindle has one bearing set 7207 C/P4DGA at each side.

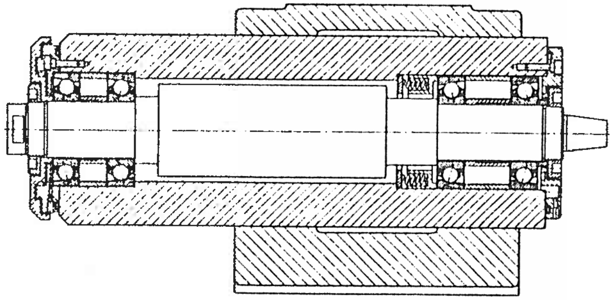


Fig. 8 Internal grinding machine spindle

Instead of using a spring for preloading the bearing system, this can be achieved by hydraulic means, making it possible to adjust the hydraulic pressure and thus the preload to suit the operating conditions. This method is used for the high speed milling machine spindle in Fig. 9 with bearing set 7010 C/PA9DT at the work side and bearing set 7008 C/PA9DT at the drive side.

1.4 Other bearing systems

A system that has proved to be very successful when great radial stiffness is required together with high speeds uses a bearing of series NN 30 K at the work side and a set of two angular contact ball bearings at the opposite side. It is then assumed that the axial stiffness is of minor importance. The distance from the spindle nose to the bearings locating the spindle axially becomes of course longer than for the other bearing systems that have been discussed, which is a disadvantage from the thermal expansion point of view. However, it may not be a disadvantage in cases where the temperatures can be kept at low levels or the spindle is running at

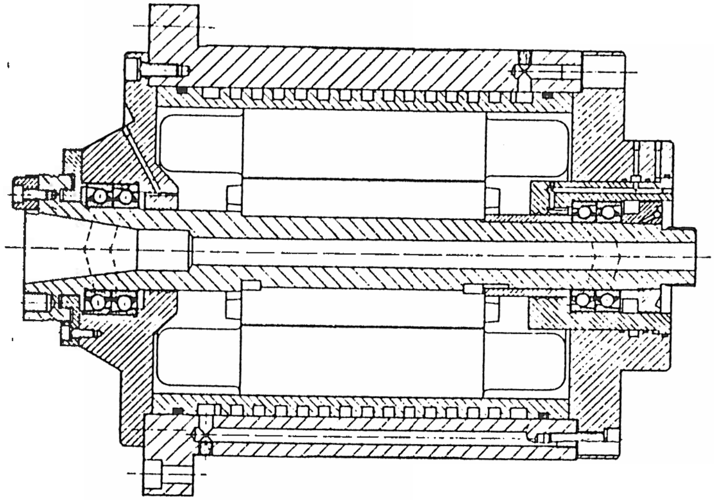


Fig. 9 High speed milling machine spindle

more or less constant speeds giving only small fluctuations in the temperature.

Systems with taper roller bearings can be used for spindles working at moderate speeds and in a narrow speed range. The most common system has one bearing at each side of the spindle and the two bearings arranged back-to-back. A system often used for long spindles has two taper roller bearings in back-to-back arrangement at the work side and another bearing at the opposite side. The taper roller bearings then determine the performance of the spindle and the other bearing acts as a support bearing, mainly to accomodate the drive forces. Fig. 10 shows a spindle for a lathe with one bearing at each side.

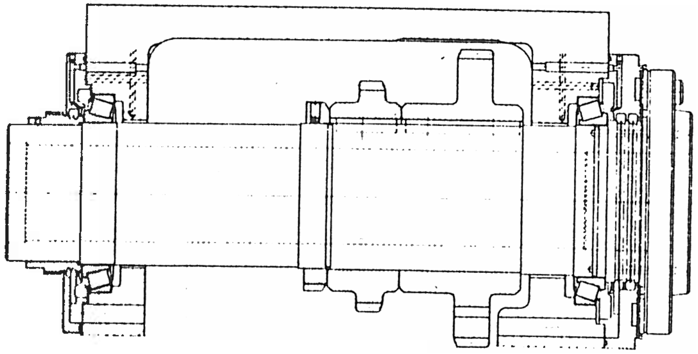


Fig. 10 Spindle with taper roller bearings

2. Bearing systems for positioning screws

Quick and accurate positioning of workpieces and tools requires that the translation mechanisms of the machine tool must be of high quality. Ball or roller screws of high precision are utilized for CNC machines for the linear movements.

In most cases bearings of special design are required to support the screws in order to benefit from the quality of the screws. The bearings made for this purpose are one row angular contact ball bearings. As the axial stiffness is an important factor they incorporate a large number of balls and have a steep contact angle, 60°. The basic designation of these bearings is BDAB 6342(00). Depending on stiffness and accuracy requirements, these bearings can be arranged in a number of different systems. They can be arranged in sets of two bearings either back-to-back, face-to-face or in tandem or with three or four bearings in these arrangements, Fig. 11.

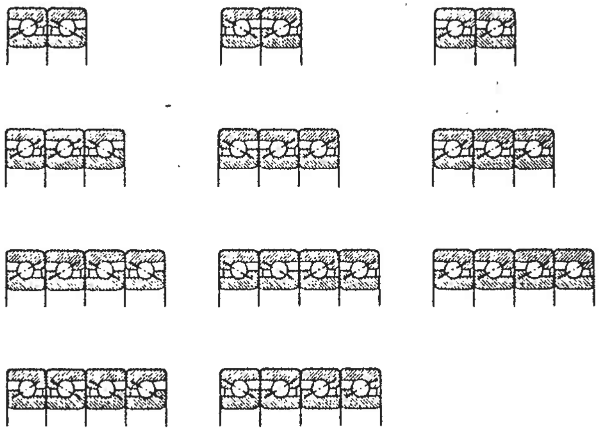


Fig. 11 Arrangements of screw support bearings

The bearings are available either as matched sets or as bearings for universal pairing. In both cases the bearings become preloaded to a predetermined value when arranged back-to-back or face-to-face.

Short screws are supported at one end only. The number of bearings and their arrangements are determined by the required stiffness and the principal load direction. Long screws require a support bearing at the other end, to avoid critical speeds. This support bearing is typically a normal deep groove ball bearing. Most commonly positioning is determined by the number of revolution of the screw and is then dependent on its lead accuracy. If the screw is long and is axially located at one end only the increase in the temperature of the screw during operation may have a detrimental effect on the accuracy. An improvement in accuracy is then achieved by using a set of the special screw support bearings at each end of the screw and by tensioning the screw on mounting, Fig. 12.

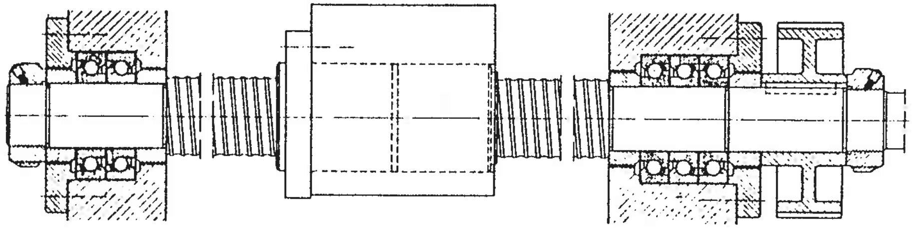
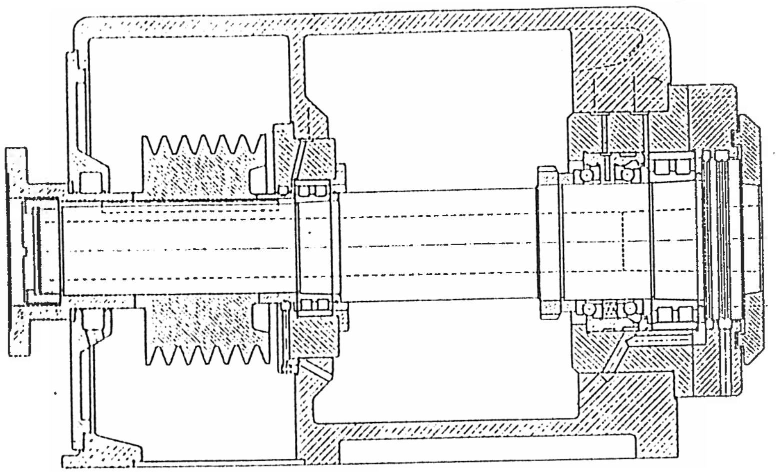
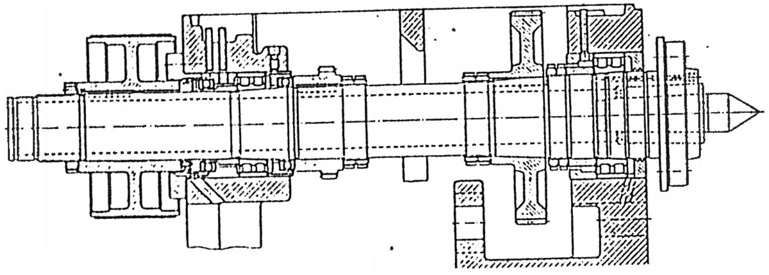


Fig. 12 Screw mounted in tension

When the temperature of the screw increases in operation the length of the screw will increase but this elongation will take place within the preload. Thus the changes in lead will be smaller than if the screw had been free to elongate in a specific direction. The capacity of the bearings must be considered when determining the screw tension, as well as the fact that some tension shall remain in the screw when the maximum elongation occurs.

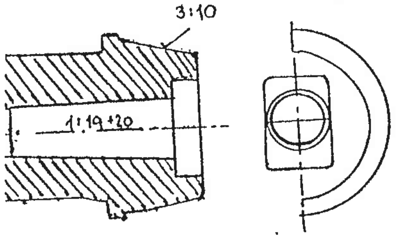
LATHE SPINDLE BEARINGS ARRANGEMENT

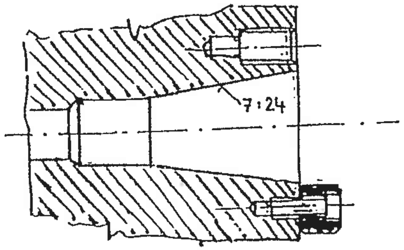




SPINDLE (TOOL) ENDS

1 : 30	- α ≙ 1° 54' 34"	- DRILLS
1 : 20	- α ≙ 2° 51' 52"	- METRIC
1 :(19 ÷ 20)	α ÷ 2° 51' ÷ 3°	- MORSE
7 : 24	α ≐ 16° 35' 40"	- ISO
3 : 10	α ÷ 17° 03' 42"	





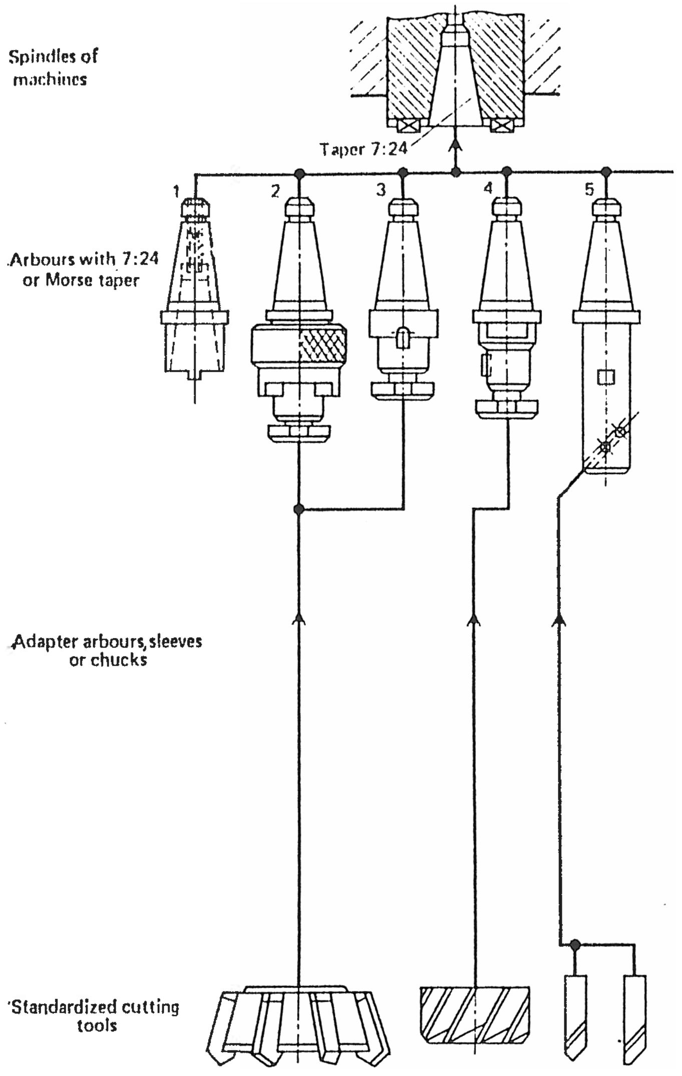
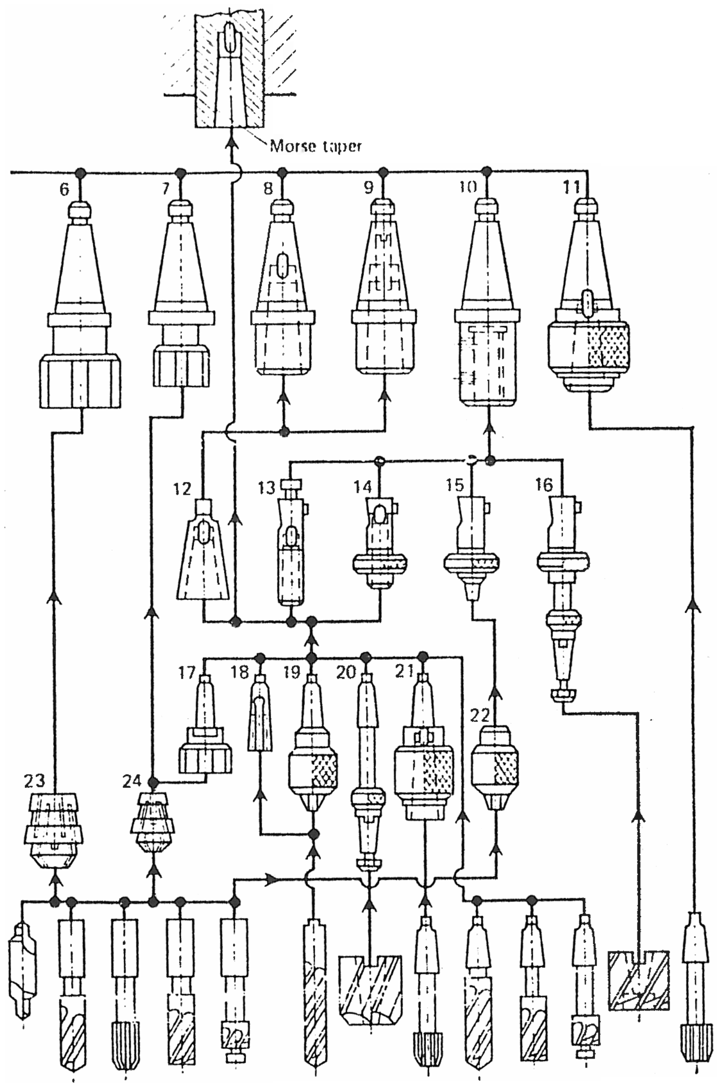


Fig. 92. Tool sets for N/C machine tools

reamers with taper shanks; *12*—adapter sleeves with external and internal Morse tapers; *13*—long round adjustable adapter sleeves for mounting and positioning of tools with Morse taper shanks; *14*—short cylindrical adapter sleeves for mounting and position adjustment of tools with Morse taper; *15*—adjustable arbours for mounting



shell core-drills and reamers; *17*—collet chucks with Morse tapers for mounting tools with straight shanks 5 to 20 mm in diameter; *18*—split sleeves with external Morse tapers for mounting drills and other tools with shanks 3 to 18 mm in diameter; *19*—three-jaw drill chucks with arbours for mounting drills 1 to 13 mm in diameter; *20*—arbours having Morse tapers for mounting shell core-drills and

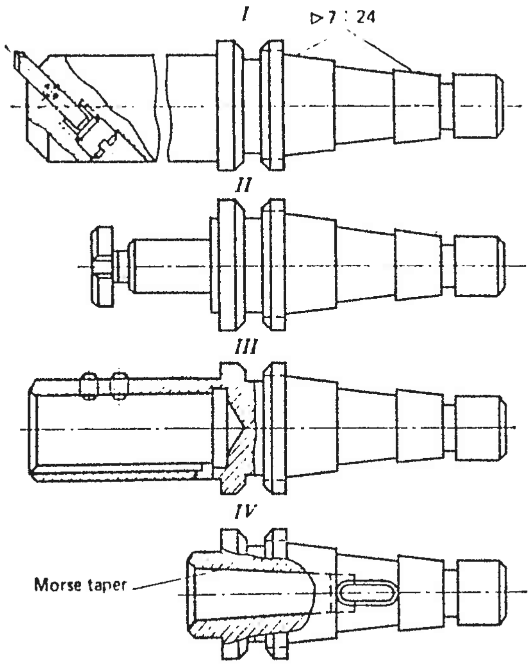


Fig. 93. Arbours with taper (7 : 24) shanks

version I — for boring tools; *version II* — for shell cutting tools with cylindrical or taper mounting hole; *version III* — for cutting tools with adapter arbours (see Fig. 94) and straight shanks; *version IV* — for cutting tools with Morse taper shanks

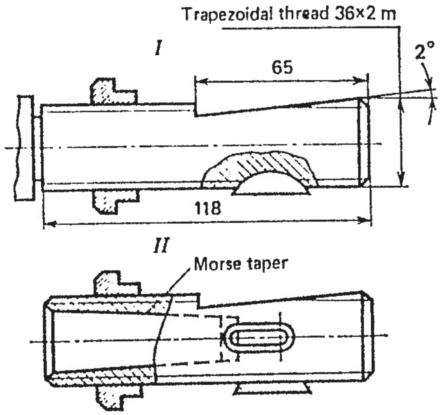
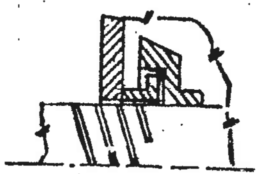
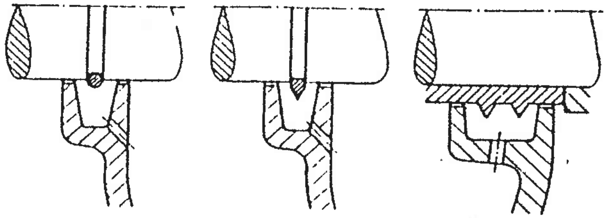


Fig. 94. Adjustable shanks for end cutting tools

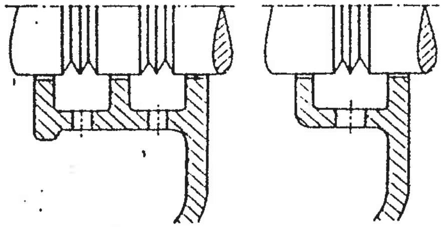
version I — for boring bars, end cutting tools and adapter arbours; *version II* — adapter arbours for cutting tools with Morse taper shanks

TÉSNĚNÍ - SEALINS

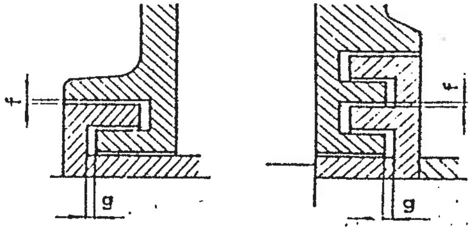




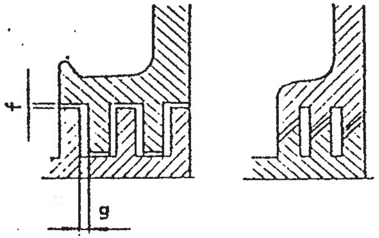
Obr. 38



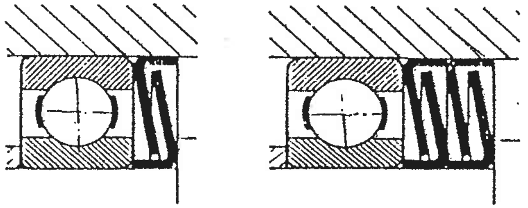
Obr. 39



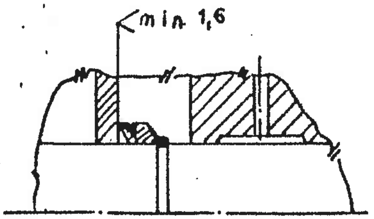
Obr. 40

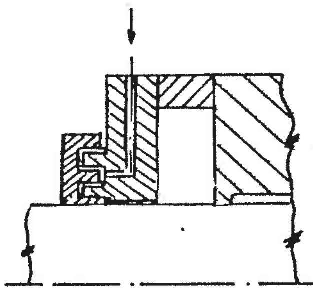


Obr. 41



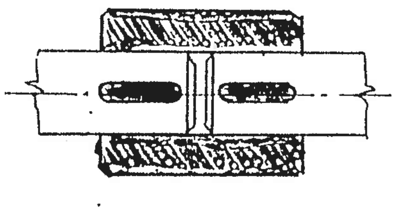
Obr. 42

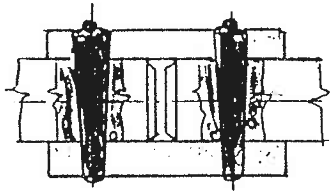


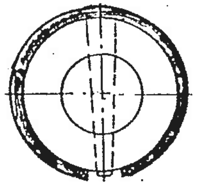


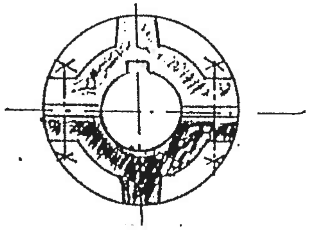
CLUTCHES :

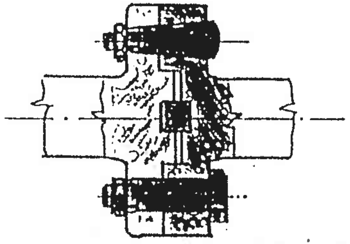
FIXED

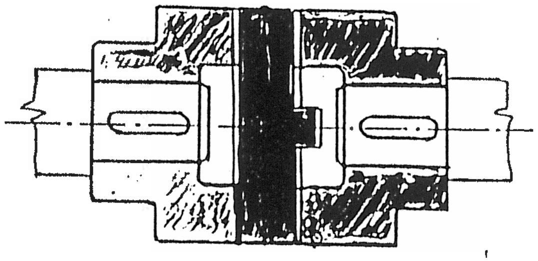


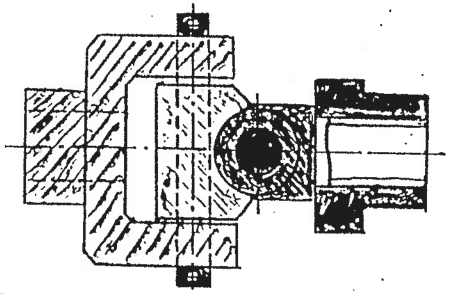




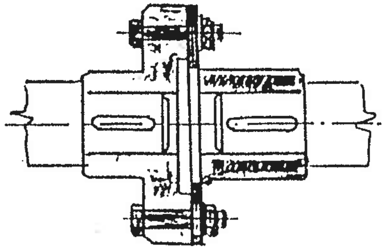


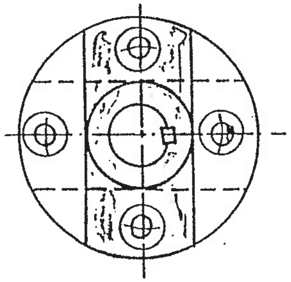




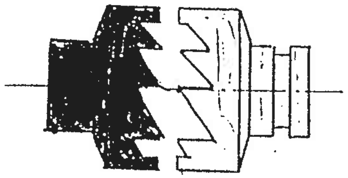


- FLEXIBLE

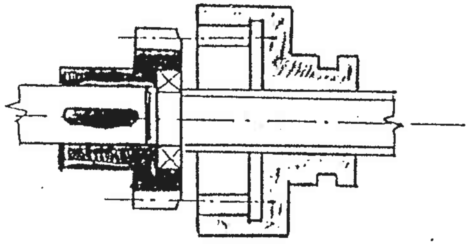




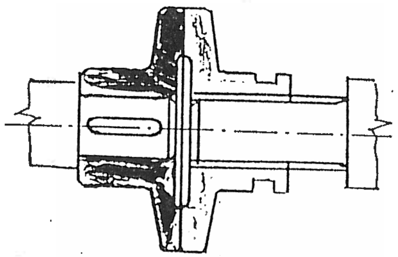
DOG

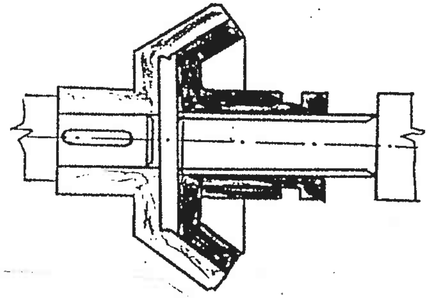


GEAR



FRICTION

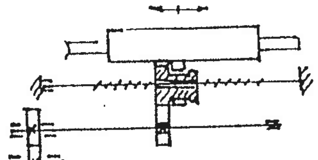




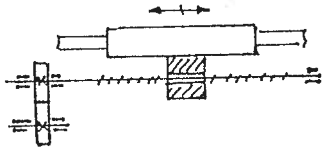
MECHANISMS OF LINEAR MOTION .

1. SCREW AND NUT
2. SCREW AND RACK
3. WORM AND WORM RACK
4. PINION AND RACK
5. CAMS
6. CRANK MECHANISMS

i.e. 1.



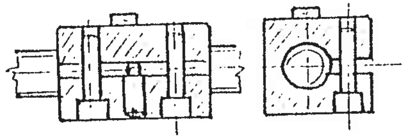
FIXED SCREW

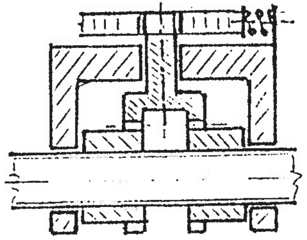


SCREW ROTATES

BACKLASH ELIMINATION

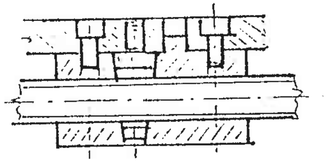
AXIAL

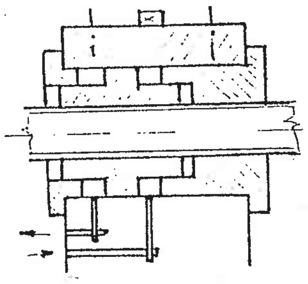




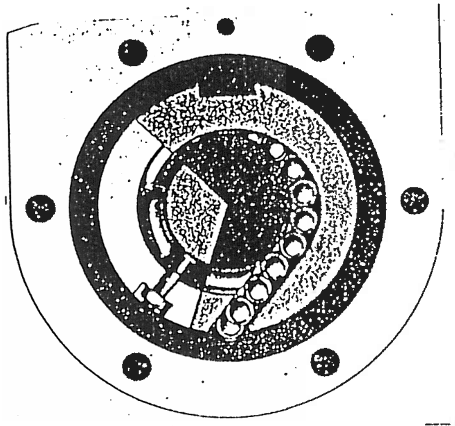
PRELOADED - ROTATION

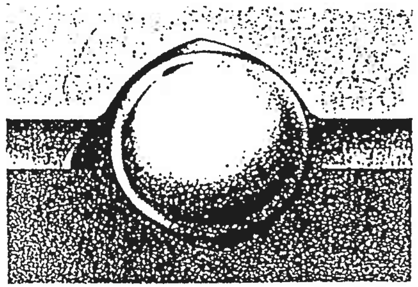
RADIAL

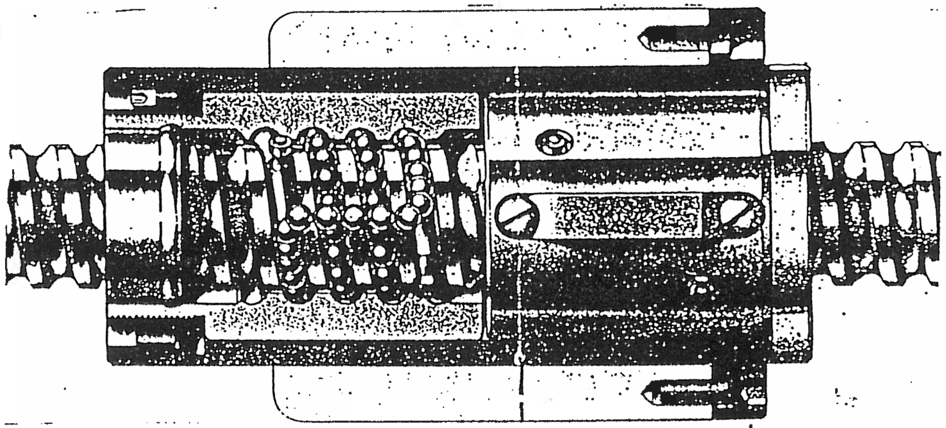


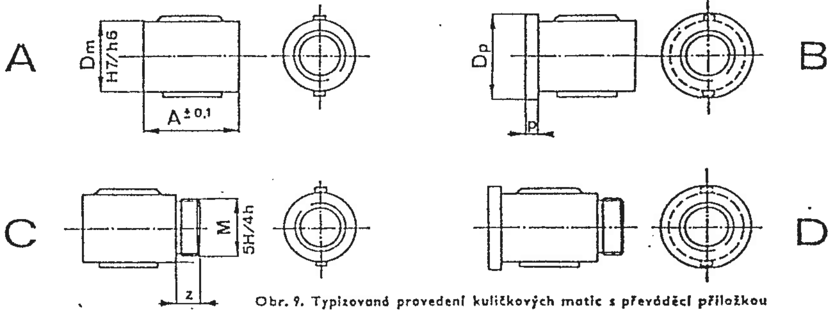


HYDRAULIC

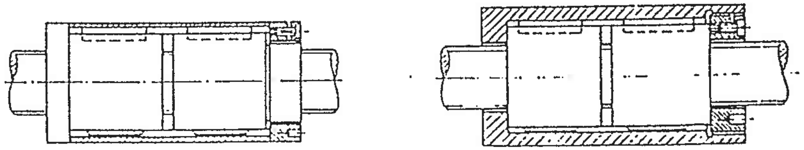




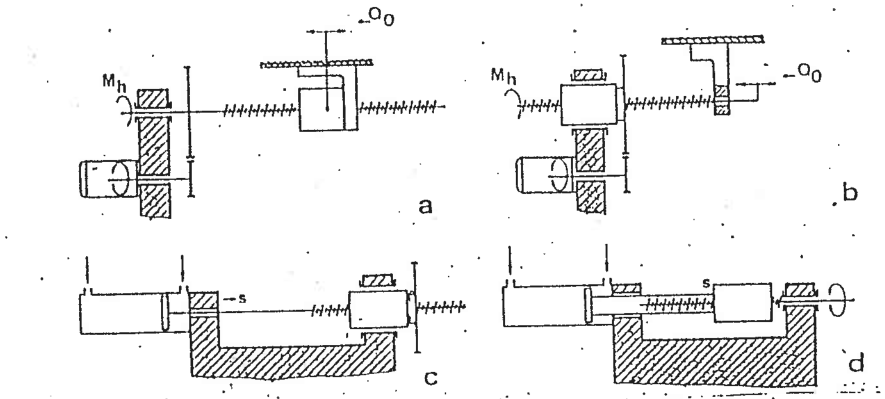


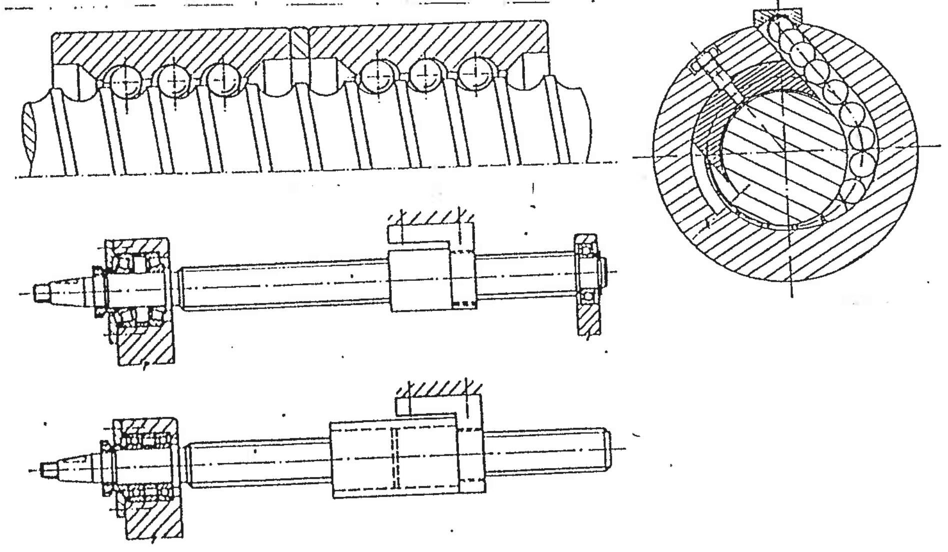


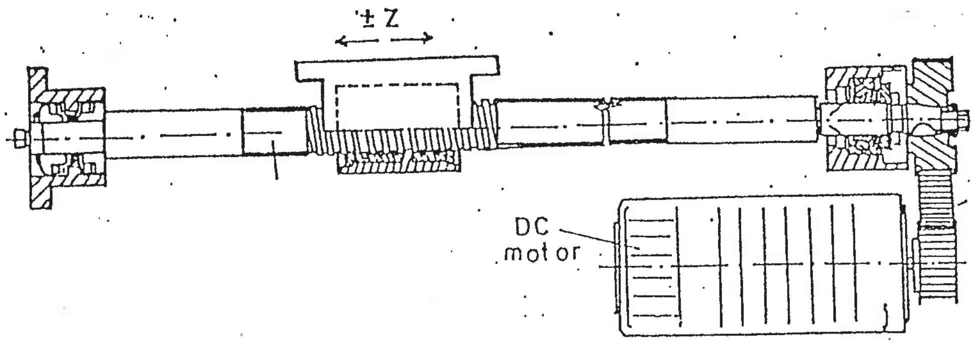
Obr. 9. Typizovaná provedení kuličkových matic s převáděcí příložkou

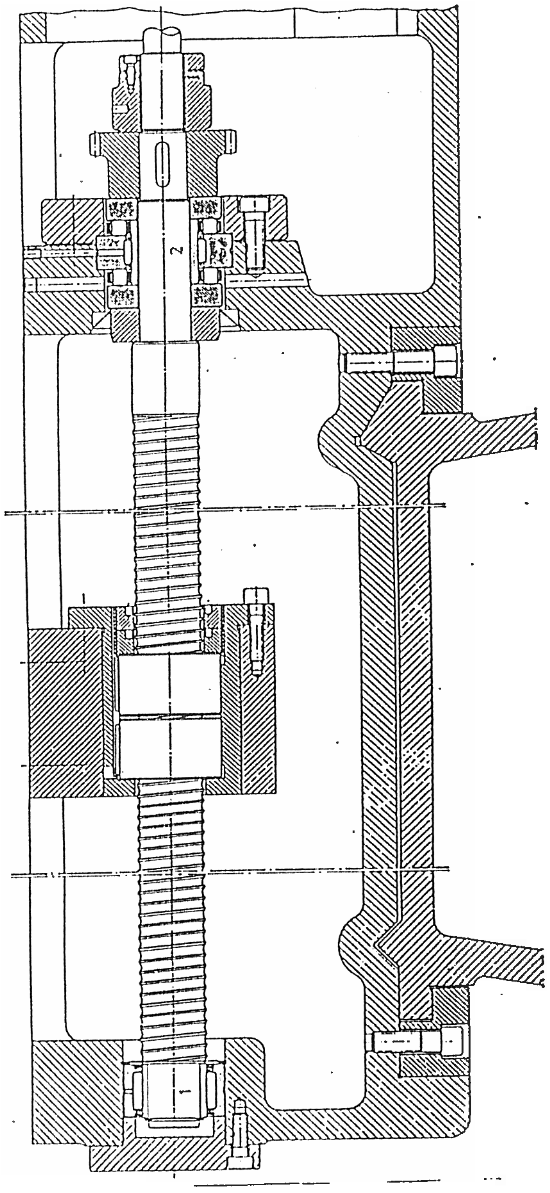


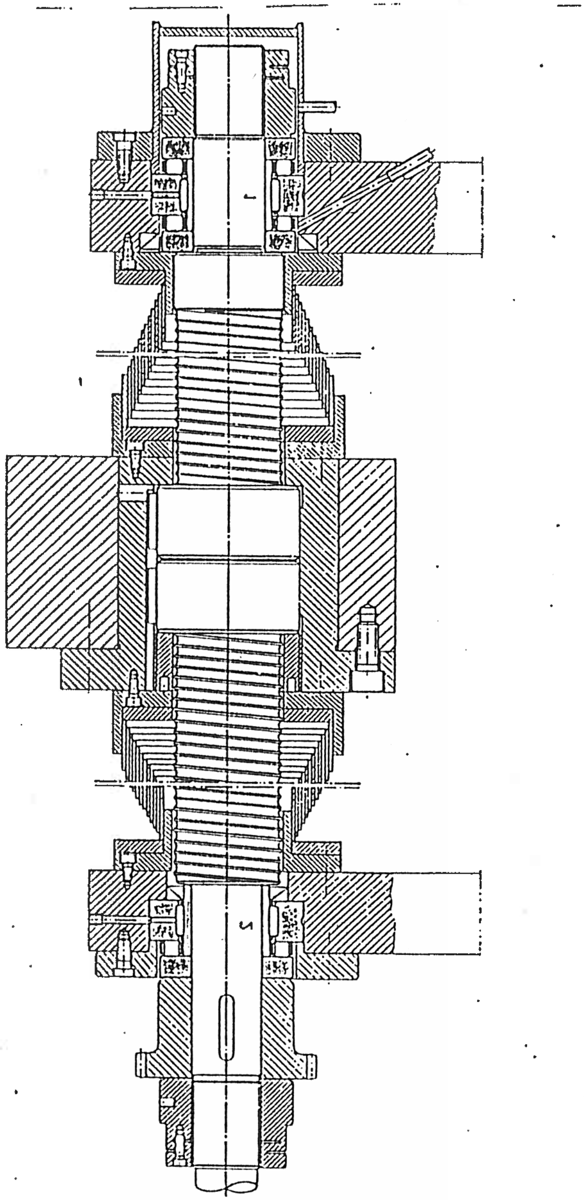
Obr. 14. Uložení dvojice matic s vymezováním vůle a nastavením předpětí pomocí závitového stavěcího kroužku


O0
Mh
a
Mh
O0
b
s
c
s
d

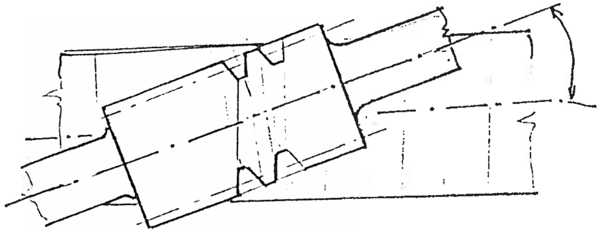



±Z
DC motor


2

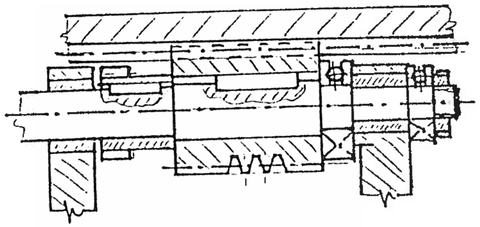


SCREW AND RACK



RARE USE ;
SMALE EFICIENCY
HIGH WEAR

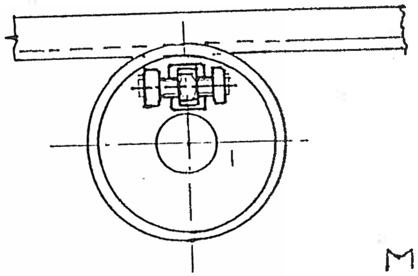
WORM AND WORM RACK



GREAT STIFFNESS

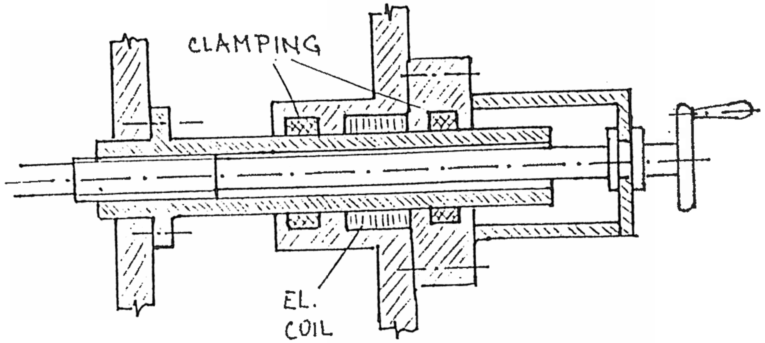
WORM STEEL p =
RACK - BRASS p - 8 ÷ 12 MPa
CAST IRON p - 4 ÷ 6 MPa

PINION AND RACK



LONG TRAVELS
QUICK MOTION
BACKLASH ELIMNATION
(POSITION MEASUREMENT)

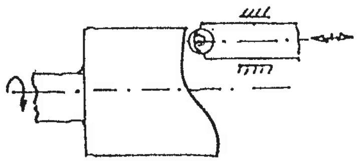
MAGNETO-SHRINKING 10^{-3}

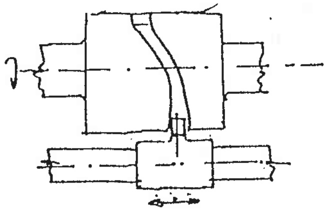


CAMS :

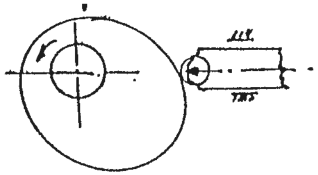
MATERIÁL 12 020 ; {15 251 ; 14 340}

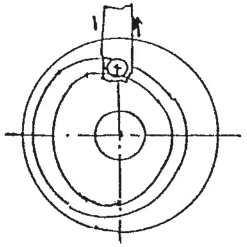
DRUM.





PLAIN:





$\rho = g(\psi)$; $y = g(\psi)$.

$V = \frac{dy}{dt} = \omega \cdot \frac{dg(\psi)}{d\psi}$

$a = \frac{dv}{dt} = \omega^2 \frac{d^2g(\psi)}{d\psi^2}$

MOTION $y = f(x) = u f(t)$

SPEED $v = \frac{dy}{dt} = u \frac{df(x)}{dt}$

ACCELER. $a = \frac{dv}{dt} = u^2 \frac{d^2 f(x)}{dt^2}$

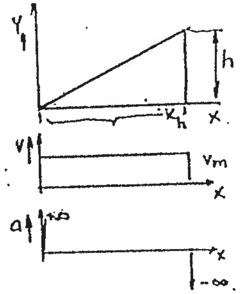
LINEAR :

$x = u \cdot t = \omega \cdot nt$

$y = v \cdot t = \varepsilon \cdot x$

$v = \text{konst}$

$a = \infty$

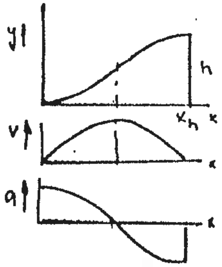


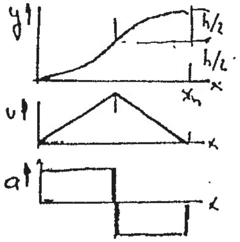
COSINUSOIDAL

$y = \frac{h}{2}\left(1 - \cos \frac{\pi x}{x_h}\right)$

$v = \frac{\pi \cdot u \cdot h}{2 x_h} \cdot \sin \frac{\pi x}{x_h}$

$a = \frac{\pi^2 u^2 h}{2 x_h^2} \cdot \cos \frac{\pi x}{x_h}$





PARABOLIC $0 < x < \frac{x_h}{2}$

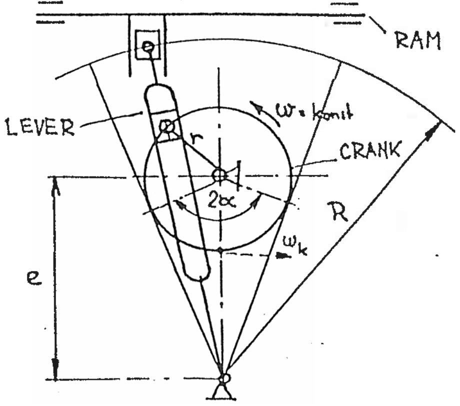
$y = \frac{a \cdot t^2}{2} = \frac{a x^2}{2u^2}$

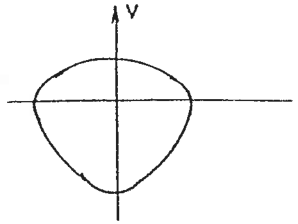
$v = \frac{a}{u} \cdot x$

$a = |a|$

CRANK MECHANIZMS

1. SYMETRICAL





$\frac{2\pi - 2\alpha}{t_h} = \omega \Rightarrow$

$t_h = \frac{2(\pi - \alpha)}{\omega}$

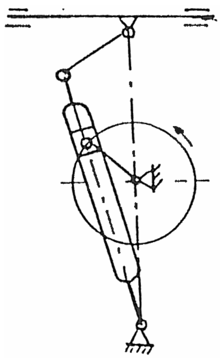
$\frac{2\alpha}{t_v} = \omega \Rightarrow$

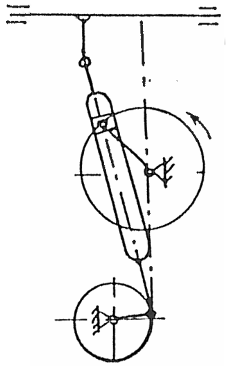
$t_v = \frac{2\alpha}{\omega}$

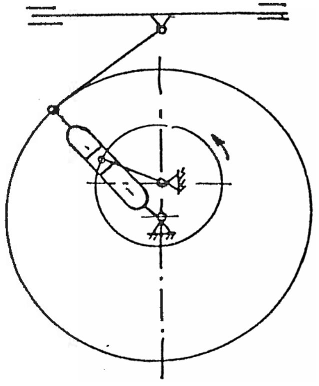
$\omega_k \cdot (e - r) = \omega \cdot r \rightarrow \omega_{k\,max} = \frac{\omega \cdot r}{(e - r)}$

$v_{max} = \omega_{k\,max} \cdot R$

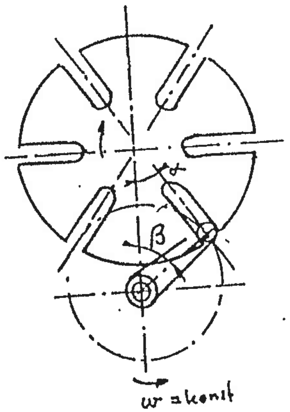
2. ASYMETRICAL

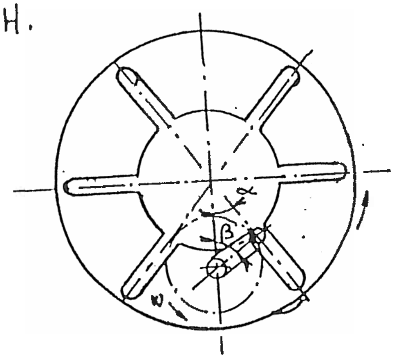






(GENEVA) MECH.





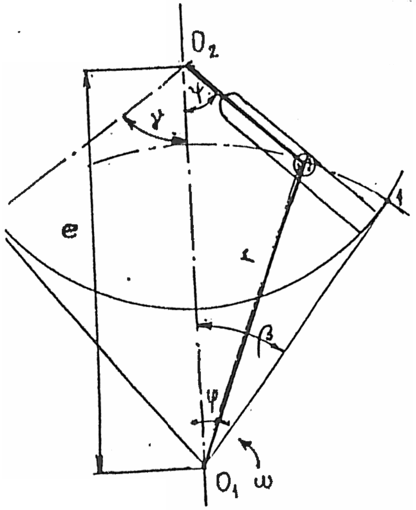
$$t_v = (2\pi - 2\beta)\frac{1}{\omega}$$
$$t_h = (2\beta)\frac{1}{\omega}$$

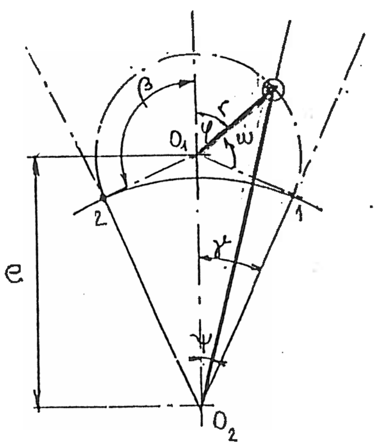
$$2\alpha = \frac{2\pi}{z} \quad | \quad \beta = \frac{\pi}{2} - \alpha$$
$$t_v = \frac{2\beta}{\omega}\ ; \ t_h = \frac{2\pi - 2\beta}{\omega}$$

$\frac{t_v}{t_h}$ - min

$$\frac{\varepsilon}{\omega^2} = 5{,}4$$

$$\frac{r}{e} = \lambda$$





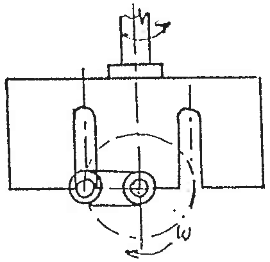
$$\frac{r}{e} = \frac{\sin\psi}{\sin\{\underbrace{\pi - (\varphi+\psi)}\}} = \lambda$$

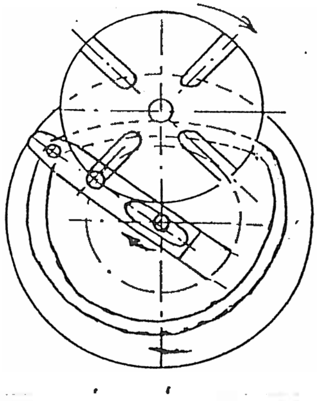
$$tg\,\psi = \frac{r\cdot\sin\varphi}{e - (r\cdot\cos\varphi)} = \frac{\lambda\cdot\sin\varphi}{1 - \lambda\cos\varphi}$$

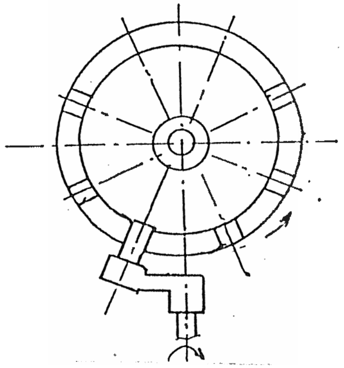
$$\frac{r}{e} = \frac{\sin\psi}{\sin(\pi - \psi - (\pi - \varphi)} = \frac{\sin\psi}{\sin(\varphi+\psi)}$$

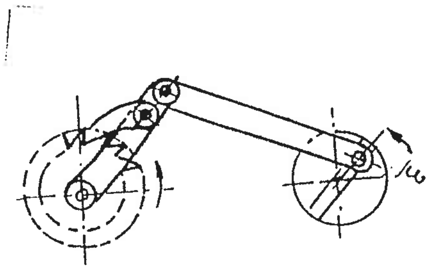
$$tg\,\psi = \frac{r\cdot\sin\varphi}{e + (r\cdot\cos\varphi)} = \frac{\lambda\cdot\sin\varphi}{1 + \lambda\cos\varphi}$$

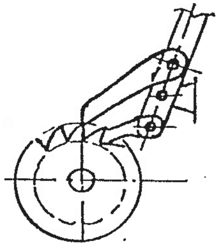
SPECIAL CASES OF GENEVA MECHANISMS

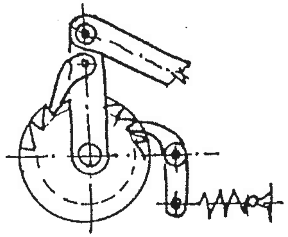




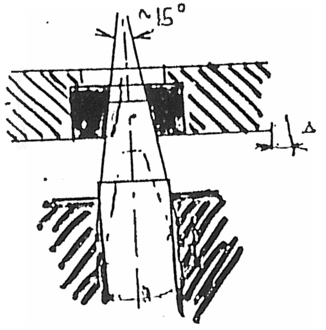


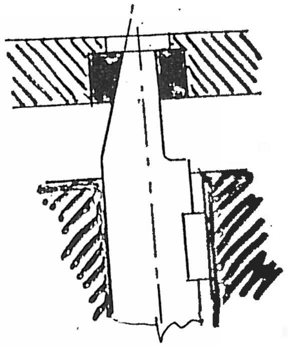


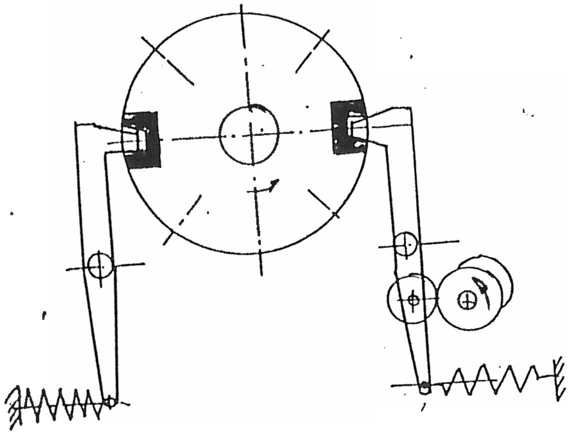




INDEXING (POSITIONING) DEVICES

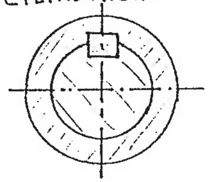




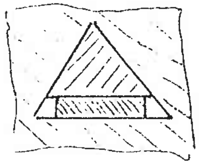


SLIDE WAYS

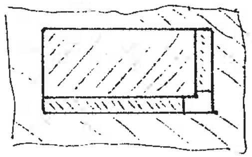
CYLINDRICAL



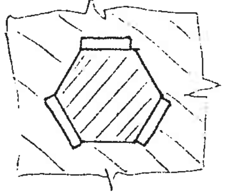
TRIANGEL



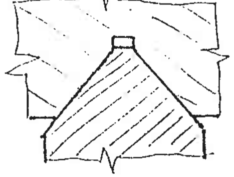
RETANGULAR



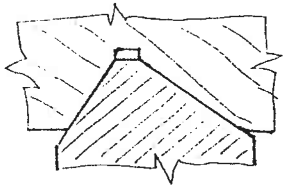
MORE-SIDES



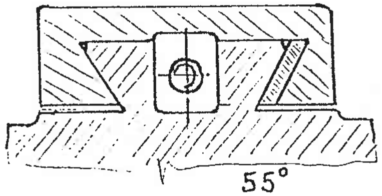
PRISMATIC



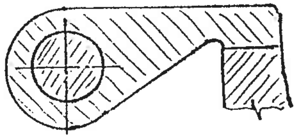
90°
SYMERIC

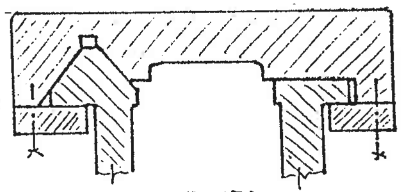


70° ÷ 120°
ASYMETRIC

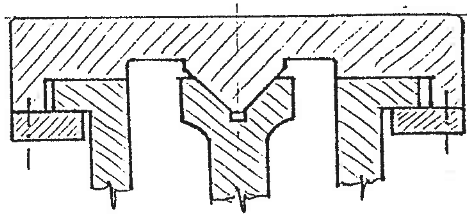


55°



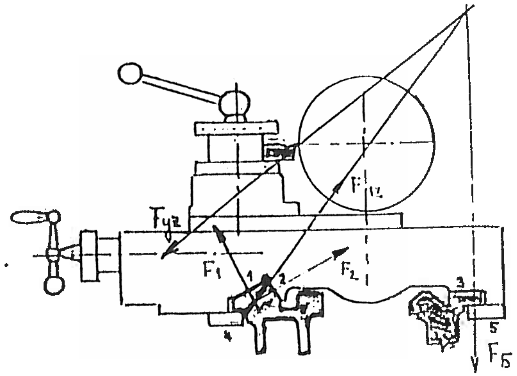


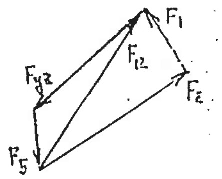
COMBINED

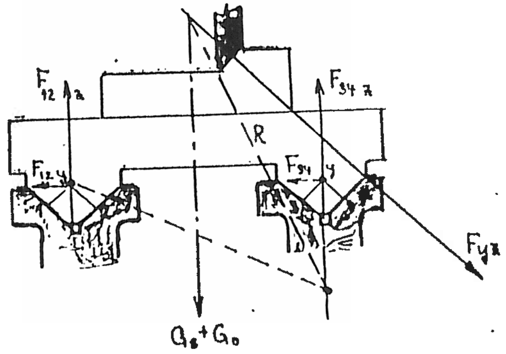


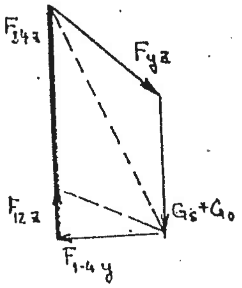
MULTIPLE

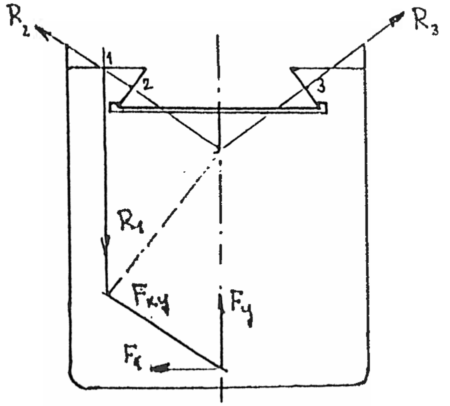
LOADING OF SLIDE WAYS

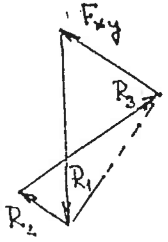




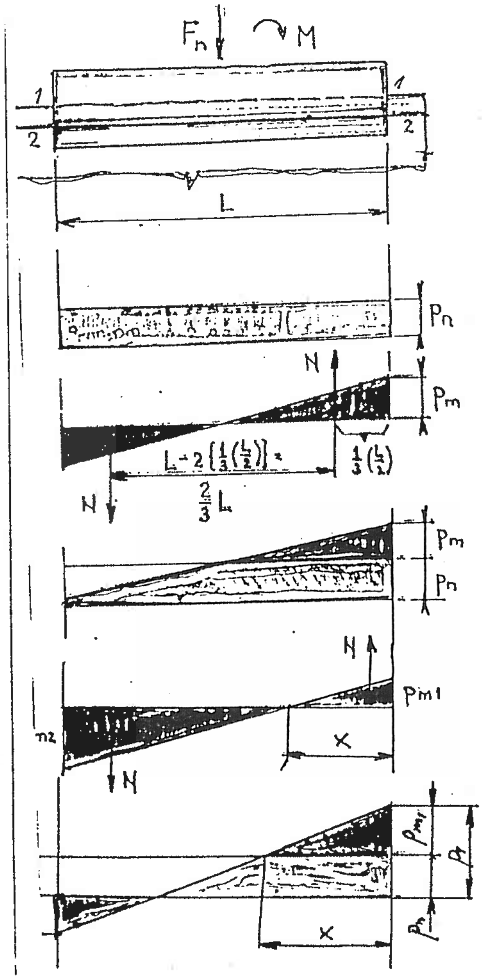


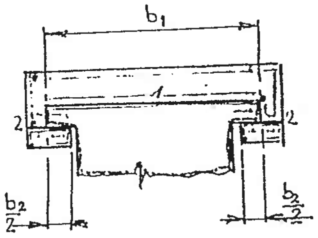






MAXIMUM PRESSURE - SLIDE LOAD DISTRIBUTION





I. FORCE act. (Fn)

pressure $p_n = \frac{F_n}{b_1 \cdot L}$

II. MOMENT act $M = N \cdot \frac{2}{3} L \Rightarrow N = \frac{3}{2} \frac{M}{L}$

THRUST $p_m = \frac{N}{b_1 \cdot \frac{L}{2}} = \frac{3}{2} \frac{M \cdot 2}{b_1 \cdot L^2} = \frac{3M}{b_1 \cdot L^2}$

$p_{max} = 2 p_m = \frac{6M}{b_1 \cdot L^2}$

III. COMBINE LOAD → Fn + M acting

under condition $p_{max} \leq p_n$ →

$$F_n \geq \frac{6M}{L}$$

IV. MOMENT M consider $b_2 \ll b_1$

$p_{m2} > p_{m1}$ $M = N\left\{\frac{2}{3}x + \frac{2}{3}(L-x)\right\} = \frac{2}{3} N \cdot L$

$N = \frac{1}{2} p_{m1} \cdot b_1 \cdot x = \frac{1}{2} p_{m2} \cdot b_2 (L-x) \to \frac{b_2 (L-x)}{b_1 \cdot x}$

$= \frac{p_{m1}}{p_{m2}} = \frac{x}{L-x}$ $\Rightarrow x =$

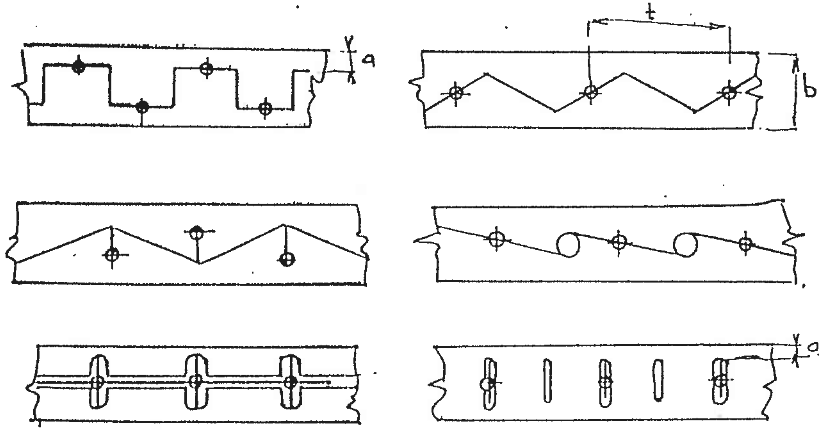
V. LOAD DISTRIBUTION Fn + M acting $F_n \geq \frac{6M}{L}$

$$\frac{x}{L-x} = \boxed{\sqrt{\frac{b_2}{b_1}}} = \sqrt{\frac{1}{\frac{b_1}{b_2}}} = \frac{1}{K}$$

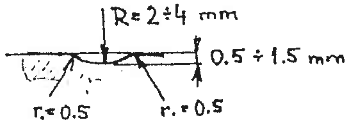
SLIDE WAYS - MATERIALS

SADDLE SLIDE (SHORT SURFACES)	SLIDEWAY BED (LONG SURF)	PRESSURE ALLOWANCE [MPa]	FRICTION COEFFICIE	
			IN MOTION	IN STEADY STATE
CAST IRON 180 ÷ 240 [HB]	CAST IRON 180 - 240 HB	0,5 - 3	0,15 - 0,17	0,25 - 0,27
	SURF. HARDENED 48 - 53 HRC	2,5 - 3,5	0,15 - 0,17	0,25 - 0,27
	STEEL BARS 55 - 65 HRC	2,5 - 3,5	0,05 - 0,15	0,2 - 0,25
PLASTICS	CAST IRON 180 - 240 HB	0,35 - 0,8	0,02 - 0,06	0,12 - 0,16
BRONZE	CAST IRON 180 - 240 HB	2 - 3	0,15 - 0,17	0,21 - 0,23

SLIDE · LUBRICATION

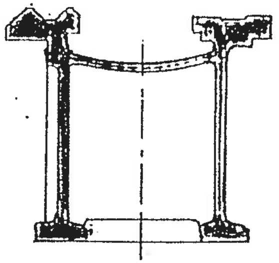


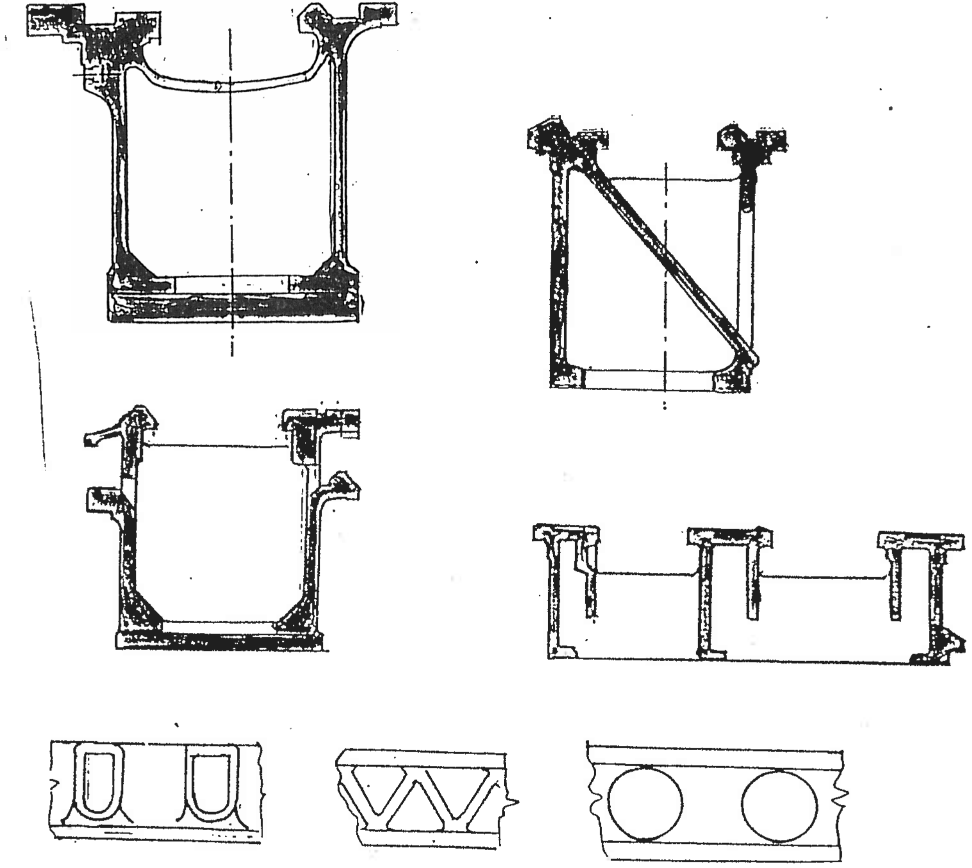
$t = (1 \div 2.5) \cdot b$; $a = (0.1 \div 0.15) \cdot b$



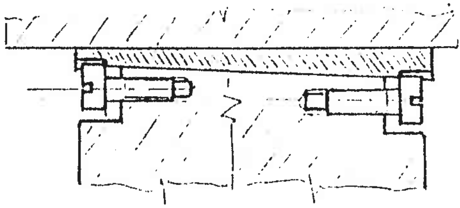
MACHINE TOOL BEDS - BASIC REQUIREMENTS:

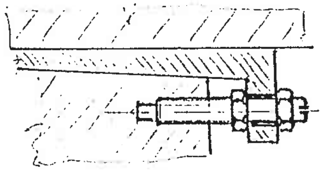
1. FORM STABILITY
2. STIFFNESS
3. DYNAMIC STIFFNESS
4. RESISTANCE TO WEAR OF SLIDEWAYS
5. EASY CHIP REMOVAL
6. SIMPLE MANUFACTURE



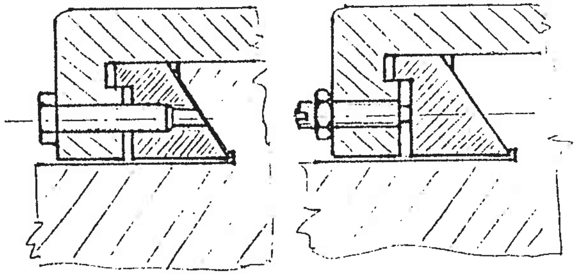


BACK-LASH ELIMINATION -

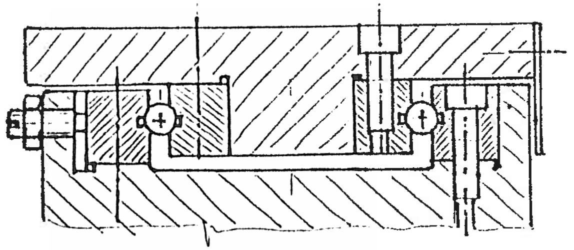




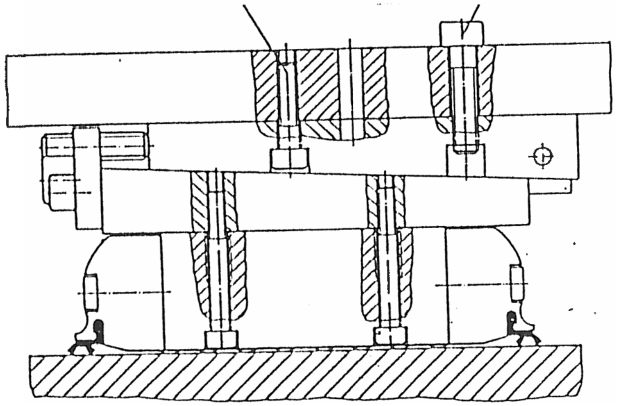
CONE BARS 1:100 ; 1:50



STRIGHT BARS

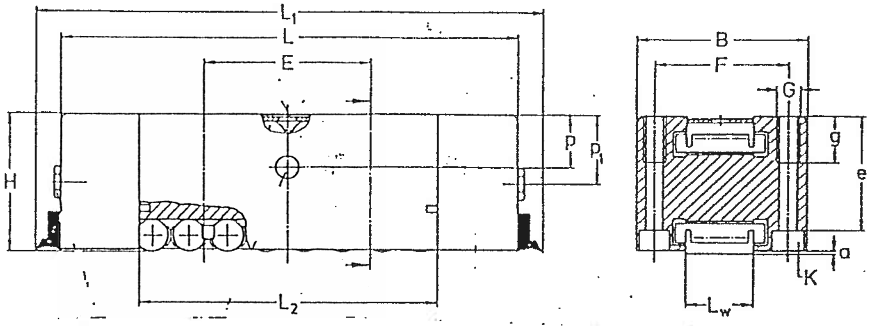


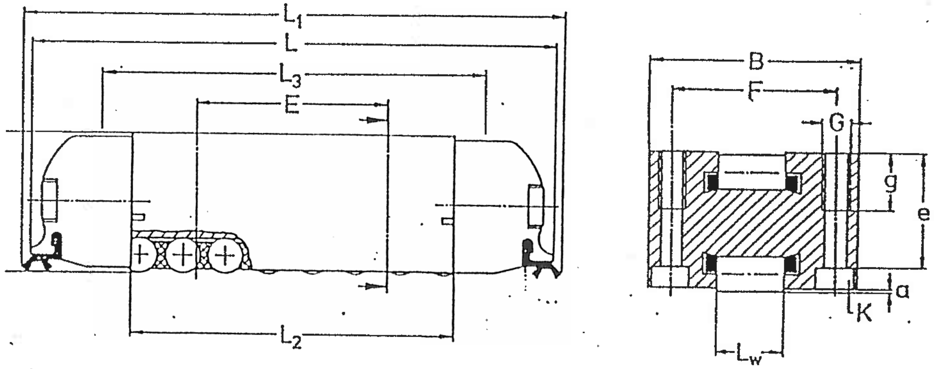
BALL SLIDES

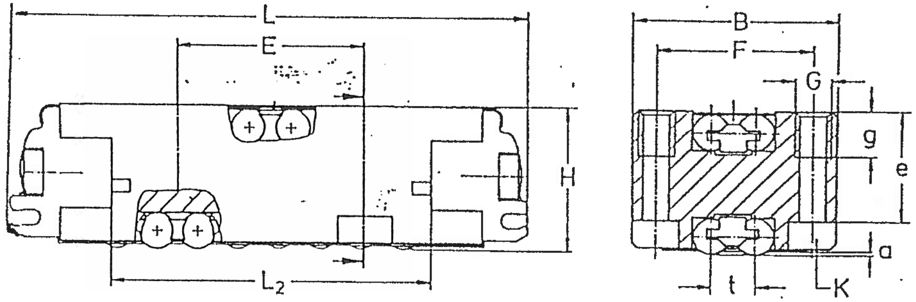


ROLLER NEST

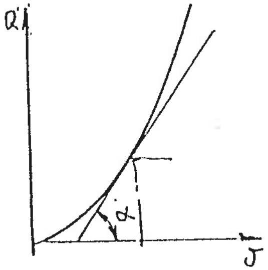
ROLLER NESTS - FOR SLIDE SUPPORT



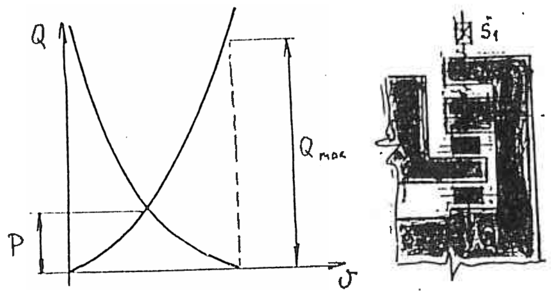




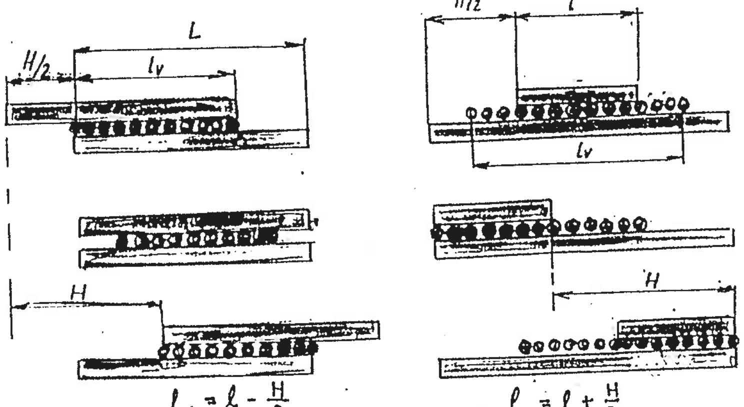
LOADING OF BEARING



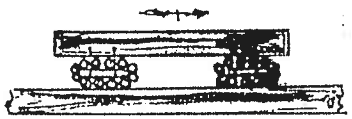
PRELOADING



ROLLING SLIDES



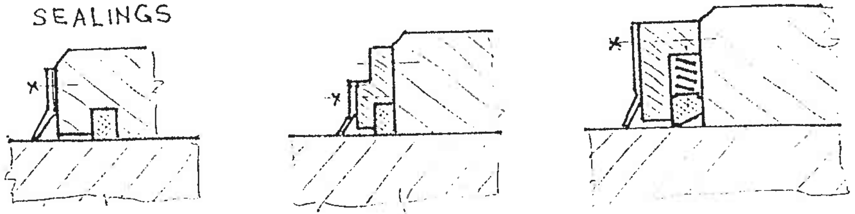
$l_v = l - \frac{H}{2}$ $\qquad$ $l_v = l + \frac{H}{2}$



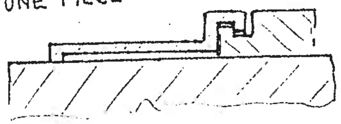
ROLLING NEST } = INFINITIVE LENGTH l_v

SLIDE COVERS

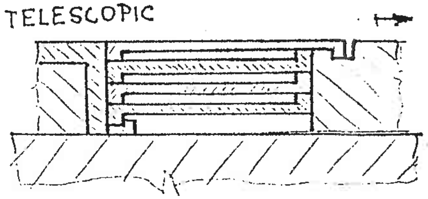
SEALINGS

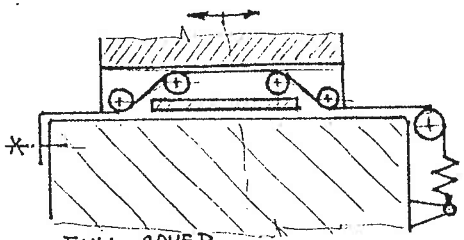


ONE PIECE

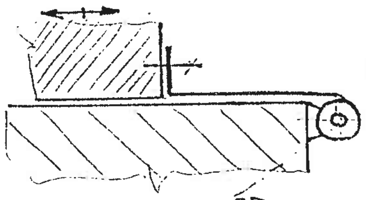


TELESCOPIC





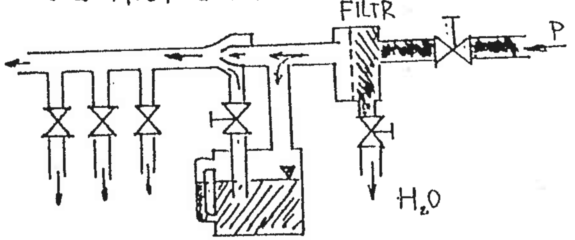
FULL COVER



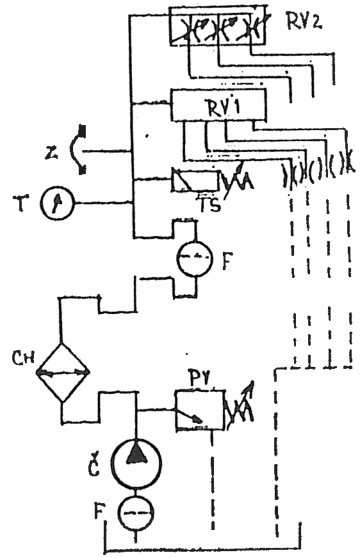
WINDER

LUBRICATING SYSTEMS

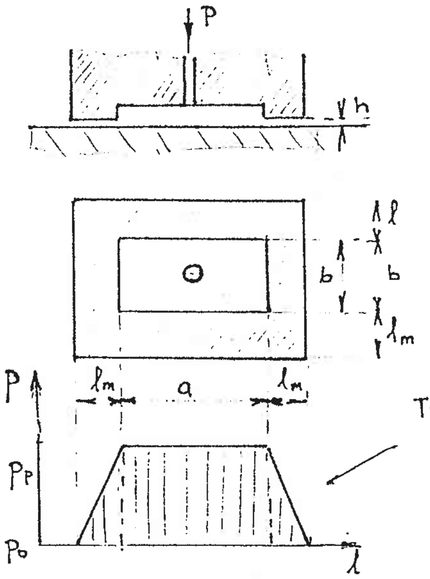
OIL MIST LUBRICATION

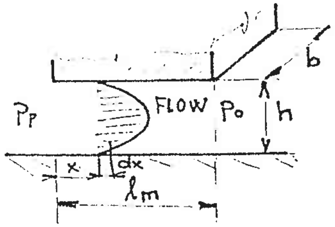


CLOSED CIRCUIT - PRESSURE LUBRICATION



HYDROSTATIC SUPPORT





FLOW

$$Q_m = \int_0^{l_m} \frac{b \cdot h^3}{12\,\eta} \cdot \frac{dp}{dx} \quad [dm^3 s^{-1}]$$

TOTAL SUPPORT

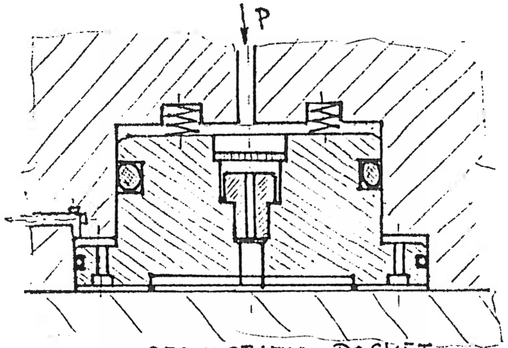
$F = \sum p_i \cdot S_i$;

$S_1 = a \cdot b$ $[mm^2]$;

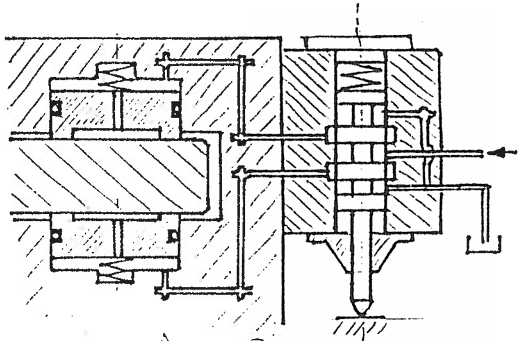
$S_2 = 2 \cdot (a+b) \cdot l_m$ $[mm^2]$

p_P = pocket pressure [MPa]

LOAD $F = p_P \cdot (S_1 + \frac{1}{2} S_2) = p_P \cdot [a \cdot b + (a+b) \cdot l_m]$ [N]



SERVOSTATIC POCKET



SERVOSTATIC SUPPORT