1 INTRODUCTION TO MACHINE TOOL DRIVES AND MECHANISMS—GENERAL PRINCIPLES OF MACHINE TOOL DESIGN

The machine tool is a machine that imparts the required shape to a workpiece with the desired accuracy by removing metal from the workpiece in the form of chips. In view of the extremely vast range of shapes that are in practise imparted to various industrial components, there exists a very large nomenclature of machine tools. Machine tools can be classified by different criteria as given below.

1. By the degree of automation into
   (i) machine tools with manual control,
   (ii) semi-automatic machine tools, and
   (iii) automatic machine tools.
2. By weight into
   (i) light-duty machine tools weighing up to 1 t,
   (ii) medium-duty machine tools weighing up to 10 t, and
   (iii) heavy-duty machine tools weighing greater than 10 t.
3. By the degree of specialisation into
   (i) general-purpose machine tools—which can perform various operations on workpieces of different shapes and sizes,
   (ii) single-purpose machine tools—which can perform a single operation on workpieces of a particular shape and different sizes, and
   (iii) special machine tools—which can perform a single operation on workpieces of a particular shape and size.

1.1 WORKING AND AUXILIARY MOTIONS IN MACHINE TOOLS

For obtaining the required shape on the workpiece, it is necessary that the cutting edge of the cutting tool should move in a particular manner with respect to the workpiece. The relative movement between the workpiece and cutting edge can be obtained either by the motion of the workpiece, the cutting tool, or by a combination of the motions of the workpiece and cutting tool. These motions which are essential to impart the required shape to the workpiece are known as *working motions*. Working motions are further classified into two categories:

1. Drive motion or primary cutting motion
2. Feed motion

Working motions in machine tools are generally of two types: rotary and translatory. Working motions of some important groups of machine tools are shown in Fig. 1.1.

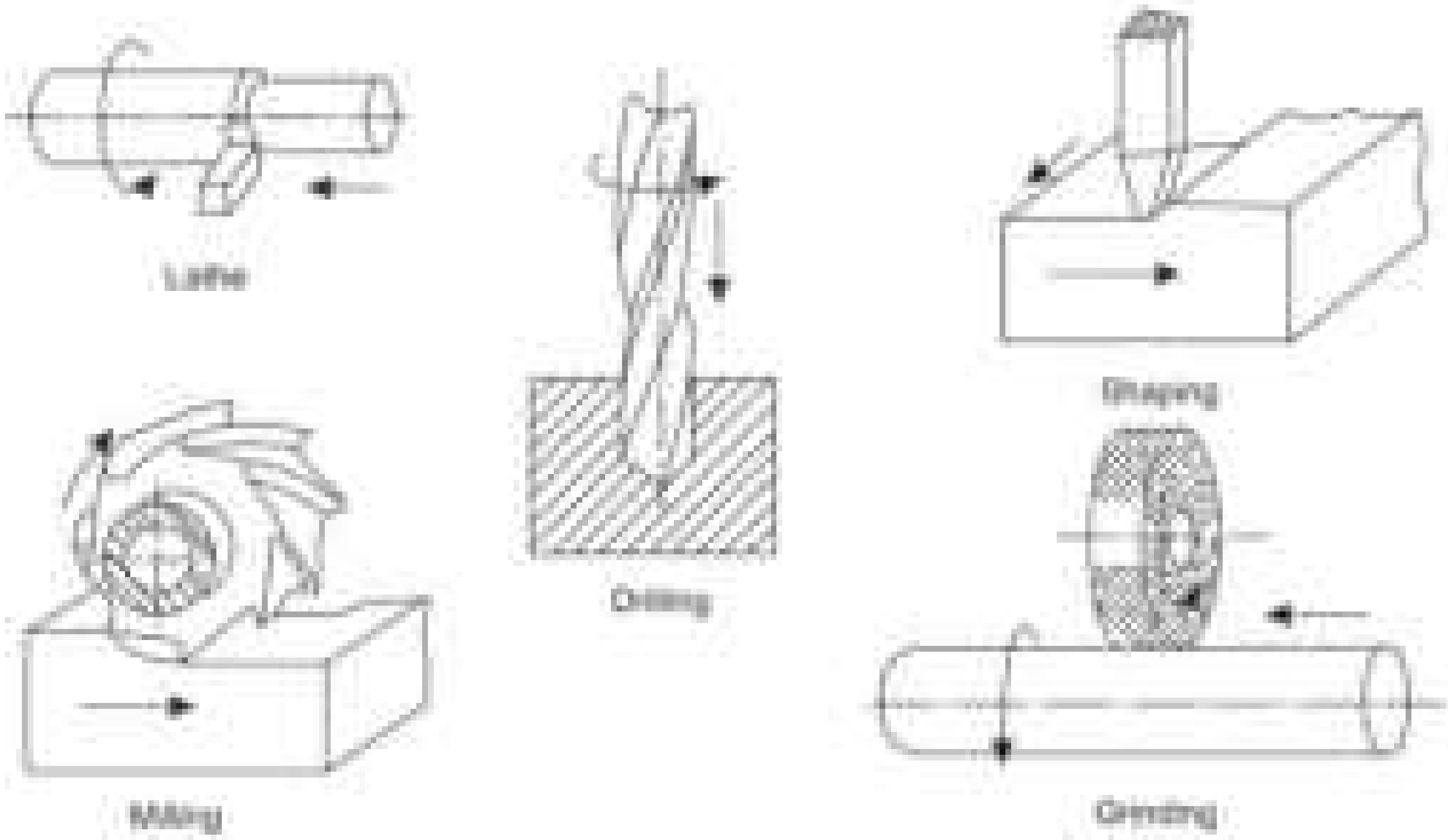


Fig. 1.1 *Working motions for some machine tools*

1. *For lathes and boring machines*
   drive motion—rotary motion of workpiece
   feed motion—translatory motion of cutting tool in the axial or radial direction
2. *For drilling machines*
   drive motion—rotary motion of drill
   feed motion—translatory motion of drill
3. *For milling machines*
   drive motion—rotary motion of the cutter
   feed motion—translatory motion of the workpiece
4. *For shaping, planing, and slotting machines*
   drive motion—reciprocating motion of cutting tool
   feed motion—intermittent translatory motion of workpiece
5. *For grinding machines*
   drive motion—rotary motion of the grinding wheel
   feed motion—rotary as well as translatory motion of the workpiece.

Besides the working motions, a machine tool also has provision for auxiliary motions. The auxiliary motions do not participate in the process of formation of the required surface but are nonetheless necessary to make the working motions fulfil their assigned function. Examples of auxiliary motions in machine tools are clamping and unclamping of the workpiece, idle travel of the cutting tool to the position from where cutting is to proceed, changing the speed of drive and feed motions, engaging and disengaging of working motions, etc.

In machine tools, the working motions are powered by an external source of energy (electrical or hydraulic motor). The auxiliary motions may be carried out manually or may also be power-operated depending upon the degree of automation of the machine tool. In general-purpose machine tools, most of the auxiliary motions are executed manually. On the other hand, in automatic machines, all auxiliary motions are automated and performed by the machine tool itself. In between these two extremes, there are machine tools in which the auxiliary motions are automated to various degrees, i.e., some auxiliary motions are automated while others are performed manually.

1.2 PARAMETERS DEFINING WORKING MOTIONS OF A MACHINE TOOL

The working motions of the machine tool are numerically defined by their velocity. The velocity of the primary cutting motion or drive motion is known as *cutting speed*, while the velocity of feed motion is known as *feed*.

The cutting speed is denoted by v and measured in the units m/min. Feed is denoted by s and measured in the following units:

1. mm/rev in machine tools with rotary-drive motion, e.g., lathes, boring machines, etc.,
2. mm/tooth in machine tools using multiple-tooth cutters, e.g., milling machines,
3. mm/stroke in machine tools with reciprocating-drive motion, e.g., shaping and planing machines, and
4. mm/min in machine tools which have a separate power source for feed motion, e.g., milling machines.

In machine tools with rotary primary cutting motion, the cutting speed is determined by the relationship,

$$v = \frac{\pi d n}{1000} \text{ m/min} \tag{1.1}$$

where d = diameter of workpiece (as in lathes) or cutter (as in milling machines), mm

n = revolutions per minute (rpm) of the workpiece or cutter

In machine tools with reciprocating primary cutting motion, the cutting speed is determined as

$$v = \frac{L}{1000 T_c} \text{ m/min} \tag{1.2}$$

where L = length of stroke, mm

T_c = time of cutting stroke, min

If the time of the idle stroke in minutes is denoted by T_i, the number of strokes per minute can be determined as

$$n = \frac{1}{T_c + T_i}$$

Generally, the time of idle stroke T_i is less than the time of cutting stroke; if the ratio T_c/T_i is denoted by K, the expression for number of strokes per minute may be rewritten as

$$n = \frac{1}{T_c(1 + T_i/T_c)} = \frac{K}{T_c(1 + K)} \tag{1.3}$$

Now, combining Eqs (1.2) and (1.3), the relationship between cutting speed and number of strokes per minute may be written as follows:

$$v = \frac{n \cdot L(K+1)}{1000K} \tag{1.4}$$

The feed per revolution and feed per stroke are related to the feed per minute by the relationship,

$$s_m = s \cdot n \tag{1.5}$$

where s_m = feed per minute
s = feed per revolution or feed per stroke
n = number of revolutions or strokes per minute

The feed per tooth in multiple-tooth cutters is related to the feed per revolution as follows:

$$s = s_z \cdot Z \tag{1.6}$$

where s = feed per revolution
s_z = feed per tooth of the cutter
Z = number of teeth on the cutter

The machining time of any operation can be determined from the following basic expression:

$$T_m = \frac{L}{s_m} \text{ min} \tag{1.7}$$

where T_m = machining time, min
L = length of machined surface, mm
s_m = feed per minute

1.2.1 Calculation of Machining Time

As mentioned above, the machining time of various operations is determined using Eq. (1.7), wherein s_m is found from Eq. (1.5) for single point tools and Eq. (1.6) for multiple tooth cutters. Further, for a given work-tool pair, an optimum cutting speed is specified for which the corresponding rpm or strokes/min is calculated using Eq. (1.1) and Eq. (1.4), respectively. It may further be noted that for a given length l of a workpiece, the actual tool travel is greater on account of the need to provide an approach of Δ1 for safe entry of tool (on commencement of machining) and over travel of Δ2 for safe exit of tool (on completion of the machining cut). Generally, Δ1 and Δ2 are taken equal to 2–3 mm. The difference in the formulae of machining time calculation for various operations arises from the individual process geometry, which is reflected in the corresponding tool travel. Hence, the calculation of tool travel for various operations is described below. In the figures of all the operations discussed below I indicates the tool position at the commencement of cut and II at the end of cut.

Operations on Lathe

(a) *Turning operation on workpiece held between centres (Fig. 1.2)*

length of tool travel $L = l + \Delta 1 + \Delta 2 + \Delta 3$

where l = length of workpiece
Δ1 = approach; generally equal to 2–3 mm
Δ2 = over travel; generally equal to 2–3 mm
Δ3 = $t \cot \varphi$, where t is depth of cut and φ is principal or side cutting edge angle; for straight edged tools $\varphi = 90°$, hence Δ3 = 0

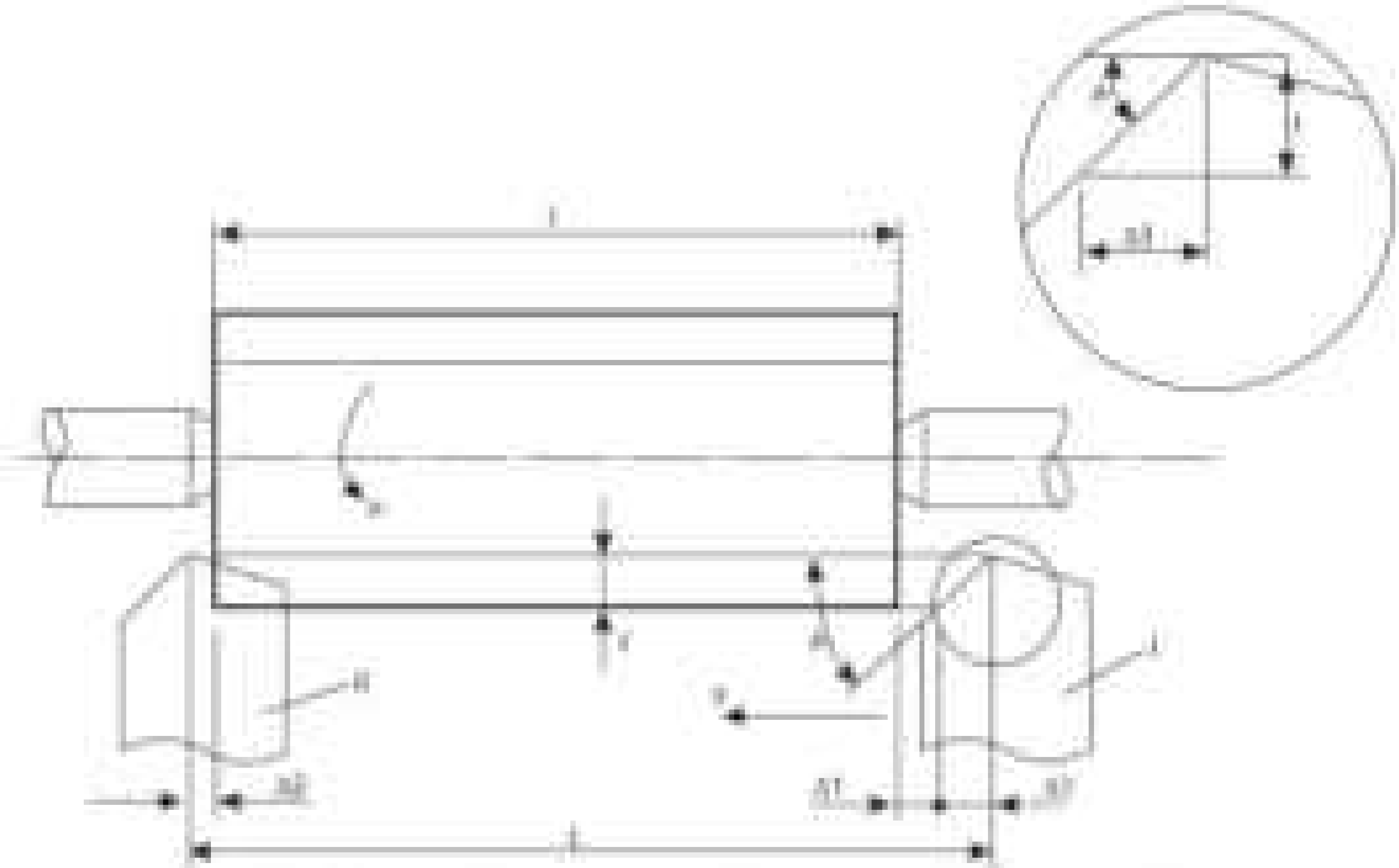


Fig. 1.2 *Turning operation on workpiece supported between centres*

(b) *Turning operation on workpiece clamped in chuck* (Fig. 1.3)

length of tool travel $L = l + \Delta 1 + \Delta 3$

where l = length of machined surface

$\Delta 1$ and $\Delta 3$ are the same as in turning of workpiece held between centres

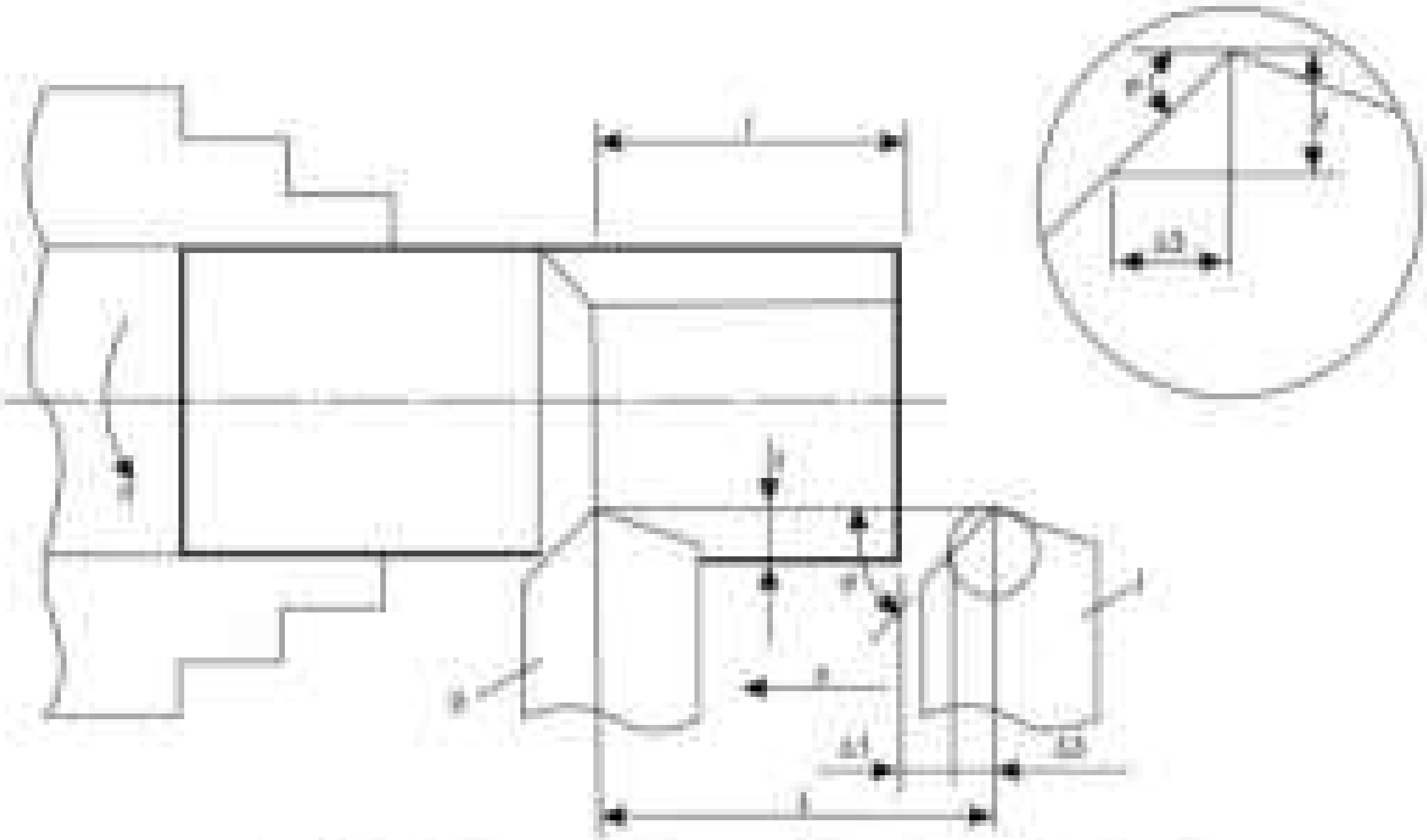


Fig. 1.3 *Turning operation on workpiece clamped in chuck*

(c) *Facing operation (Fig. 1.4)*

length of tool travel $L = D/2 + \Delta 1 + \Delta 2 + \Delta 3$

where D = diameter of workpiece

$\Delta 1$ = approach; generally equal to 2–3 mm

$\Delta 2$ = over travel; generally equal to 1–2 mm is essential to ensure that a protruding stem is not left attached to the face of the machined workpiece

$\Delta 3$ = $t \cot \phi$, where t is depth of cut and ϕ is principal or side cutting edge angle; for straight edged tools $\phi = 90°$, hence $\Delta 3 = 0$

The length of tool travel for parting and grooving operations is determined in a similar manner.

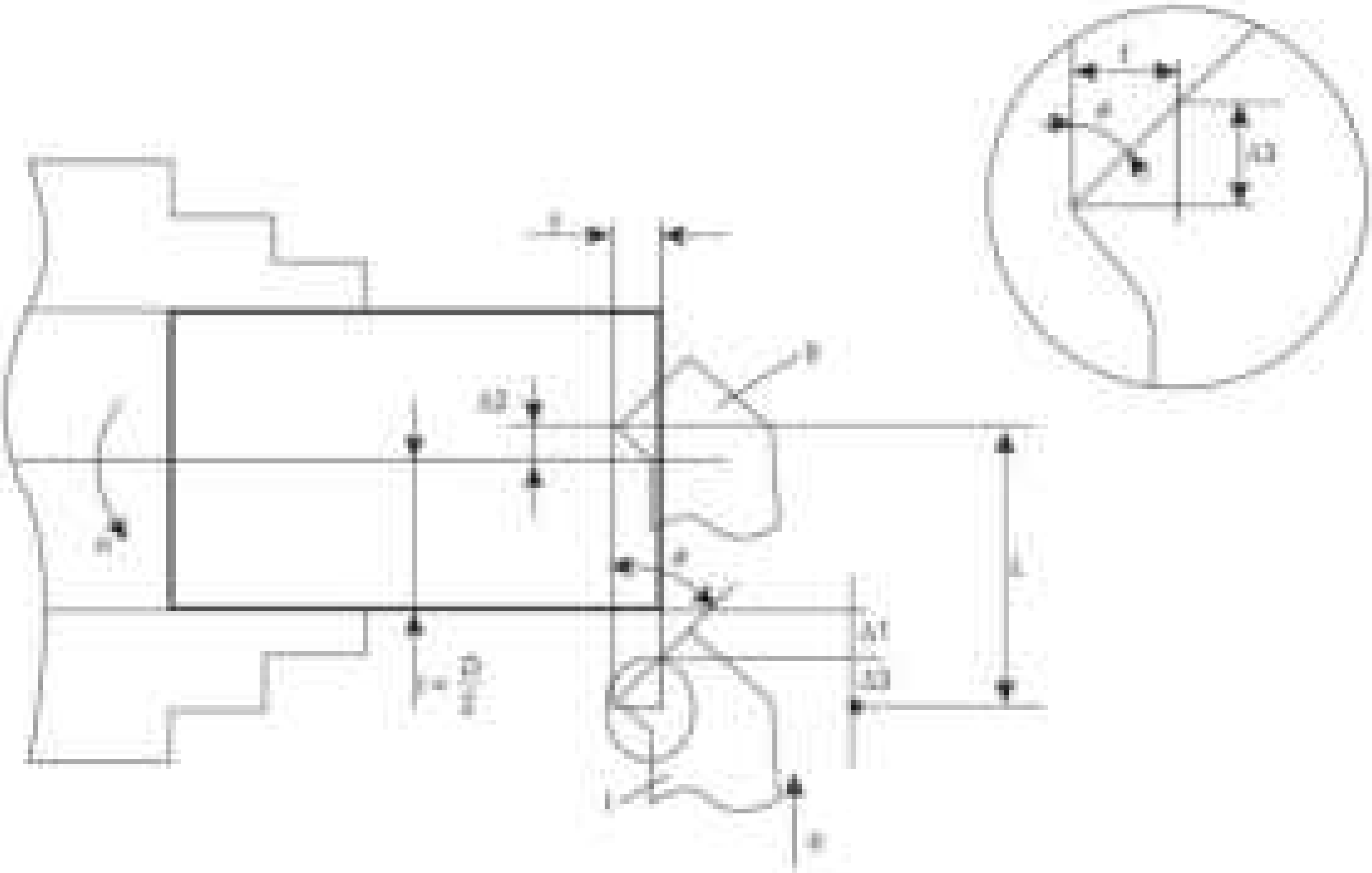


Fig. 1.4 *Facing operation*

(d) *Boring operation in partial length of workpiece; hole ϕd to be enlarged to ϕD (Fig. 1.5)*

length of tool travel $L = l + \Delta 1 + \Delta 3$

where l = length of bore

$\Delta 1$ = approach; generally equal to 2–3 mm

$\Delta 3$ = $t \cot \phi$, where t is depth of cut and ϕ is principal or side cutting edge angle; for straight edged tools $\phi = 90°$, hence $\Delta 3 = 0$

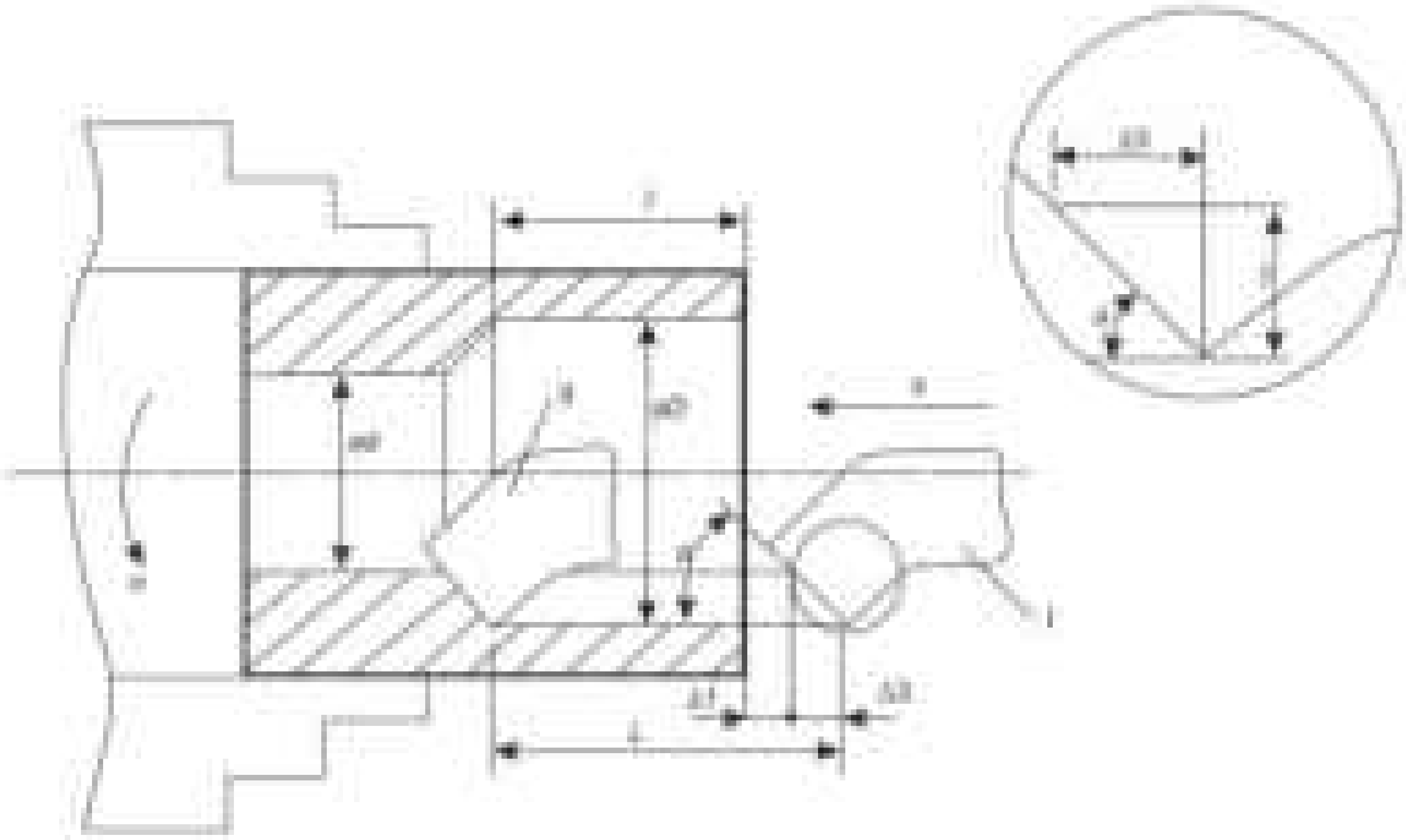


Fig. 1.5 *Boring operation in partial length of workpiece*

(d) *Boring operation in full length of workpiece; hole ϕd to be enlarged to ϕD (Fig. 1.6)*
length of tool travel $L = l + \Delta 1 + \Delta 2 + \Delta 3$

where l = length of bore

$\Delta 2$ = over travel; generally equal to 2–3 mm

$\Delta 1$ and $\Delta 3$ are the same as in boring operation in partial length of workpiece

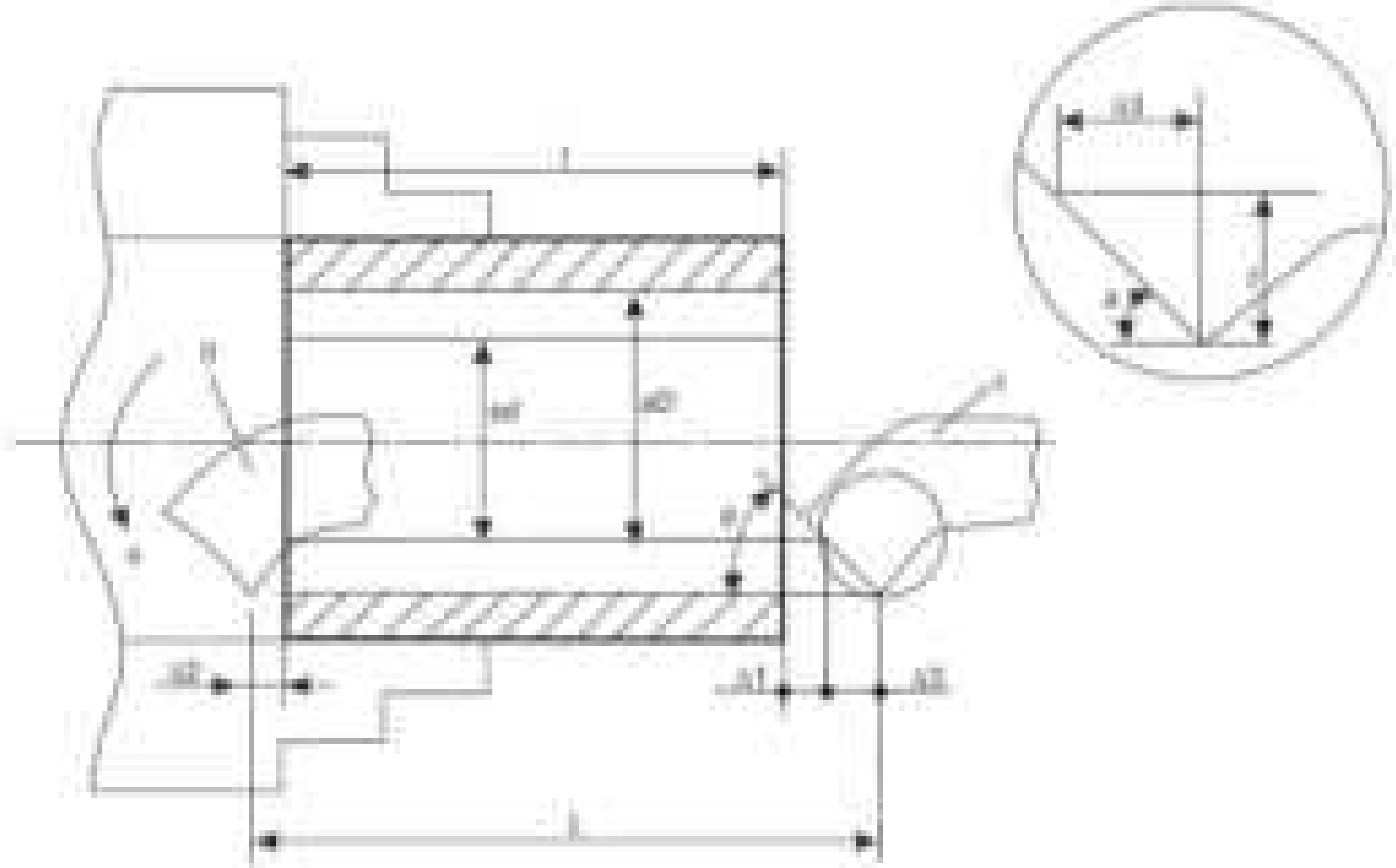


Fig. 1.6 *Boring operation in full length of workpiece*

Example 1.1

Determine the machining time for turning a shaft from $\phi 70$ mm to $\phi 64$ mm over a length of 200 mm at $n = 600$ rpm and $s = 0.4$ mm/rev. The turning tool has principal cutting edge angle $\phi = 45°$.

Depth of cut $= \dfrac{70 - 64}{2} = 3$ mm

Length of travel $L = 200 + t \cot \phi + \Delta 1 + \Delta 2$

Assuming $\Delta 1$ and $\Delta 2 = 2$ mm each

$$L = 200 + 3 \times 1 + 2 + 2 = 207 \text{ mm}$$

Machining Time $= \dfrac{207}{600 \times 0.4} = 0.8625$ min.

Example 1.2

A ring has to be cut out from a pipe of outside diameter $D = 100$ mm and inside diameter $d = 84$ mm at 250 rpm and feed 0.14 mm/rev. Calculate the machining time.

Length of travel in a pipe cutting operation is

$$L = \frac{D - d}{2} + \Delta 1 + \Delta 2$$

Assuming $\Delta 1 = \Delta 2 = 2$ mm

$$L = \frac{100 - 84}{2} + 2 + 2 = 12$$

Machining time $T_m = \dfrac{L}{n \cdot s_0} = \dfrac{12}{250 \times 0.14} = 0.342$ min.

Operations on Drilling Machine *Drilling operation* (Fig. 1.7)

length of tool travel $L = l + \Delta 1 + \Delta 2 + \Delta 3$

where l = height of the workpiece

$\Delta 1$ = approach; generally equal to 2–3 mm

$\Delta 2$ = over travel; generally equal to 2–3 mm

$\Delta 3 = (d/2) \cot \phi$, where d is drill diameter and 2ϕ is the lip angle of the drill

The length of tool travel for counter boring and reaming operations can be determined in a similar manner.

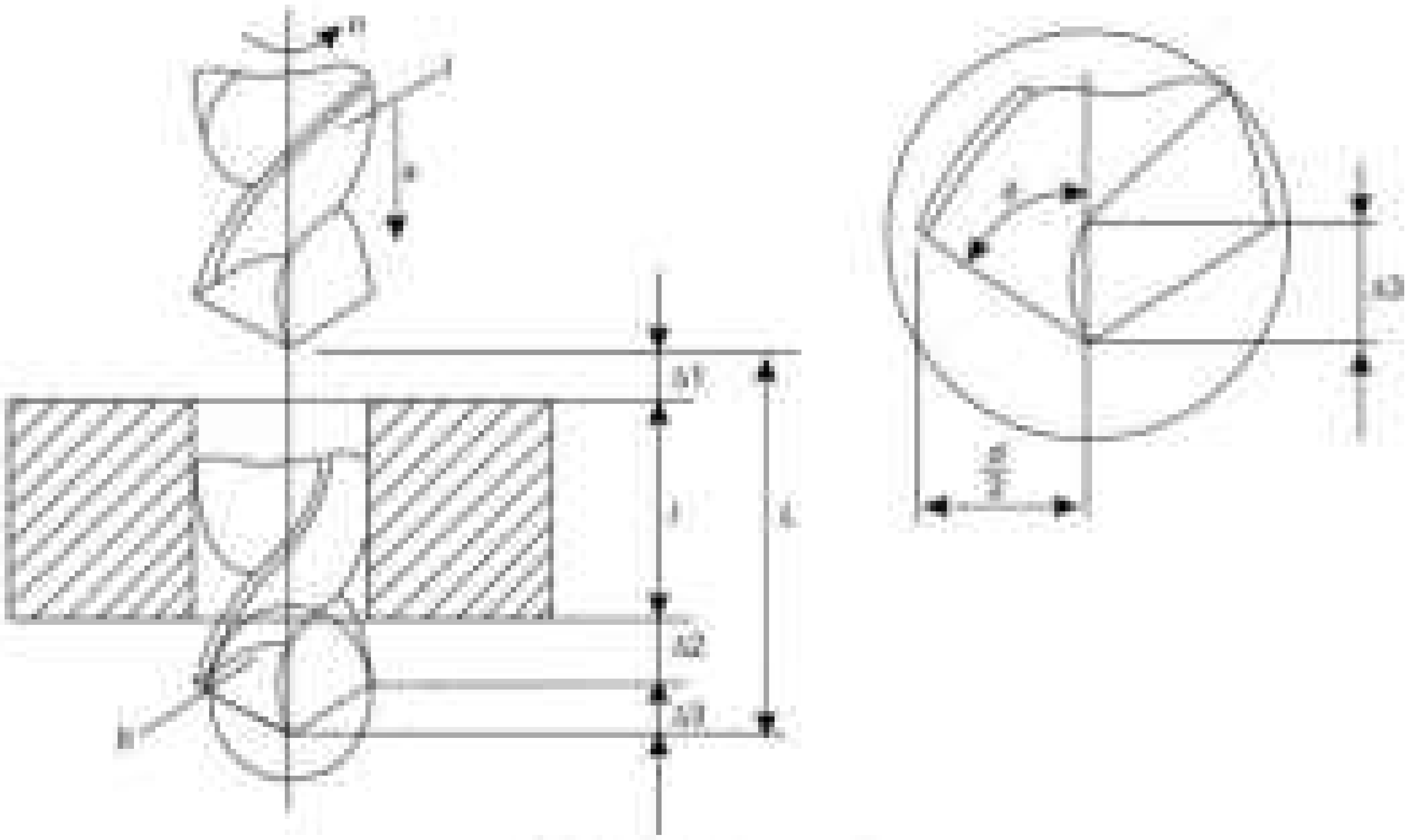


Fig. 1.7 Drilling operation

Example 1.3

Calculate the machining time for drilling a $\phi 30$ through hole in a 30 mm thick plate at a speed of 30 m/min and feed 0.1 mm/tooth.

Length travel $L = 30 + \Delta 1 + \Delta 2 + \dfrac{d}{2} \cot \phi$

Assuming $\Delta 1 = \Delta 2 = 2$ mm and $\phi = 60°$ (half of lip angle)

$$L = 30 + 2 + 2 + \frac{30}{2} \cot 60° = 42.66 \text{ mm}$$

The rpm of the drill is

$$n = \frac{1000v}{\pi d} = \frac{1000 \times 30}{\pi \times 30} = \frac{1000}{\pi}$$

Feed per revolution of drill = 2 × feed per tooth because a drill has two cutting teeth.

Therefore, $s_o = 2 \times 0.1 = 0.2$ mm/rev

Hence, feed per minute $s_m = \dfrac{1000}{\pi} \times 0.2 = \dfrac{200}{\pi}$ mm/min

Machining time $T_m = \dfrac{L}{s_m} = \dfrac{42.66}{200/\pi} = 0.67$ min

Operations on Milling Machine In all the milling operations described below

$\Delta 1$ = approach; generally equal to 2–3 mm

$\Delta 2$ = over travel; generally equal to 2–3 mm

(a) *Horizontal milling machine: Plain milling operation* (Fig. 1.8)

length of cutter travel $L = l + \Delta 1 + \Delta 2 + \Delta 3$

where $\quad l$ = length of the workpiece

$$\Delta 3 = BC = \sqrt{OC^2 - OB^2} = \sqrt{R^2 - OB^2} = \sqrt{R^2 - (R-t)^2} = \sqrt{R^2 - (R^2 + t^2 - 2Rt)}$$

$$= \sqrt{2Rt - t^2} = \sqrt{t(D-t)}$$

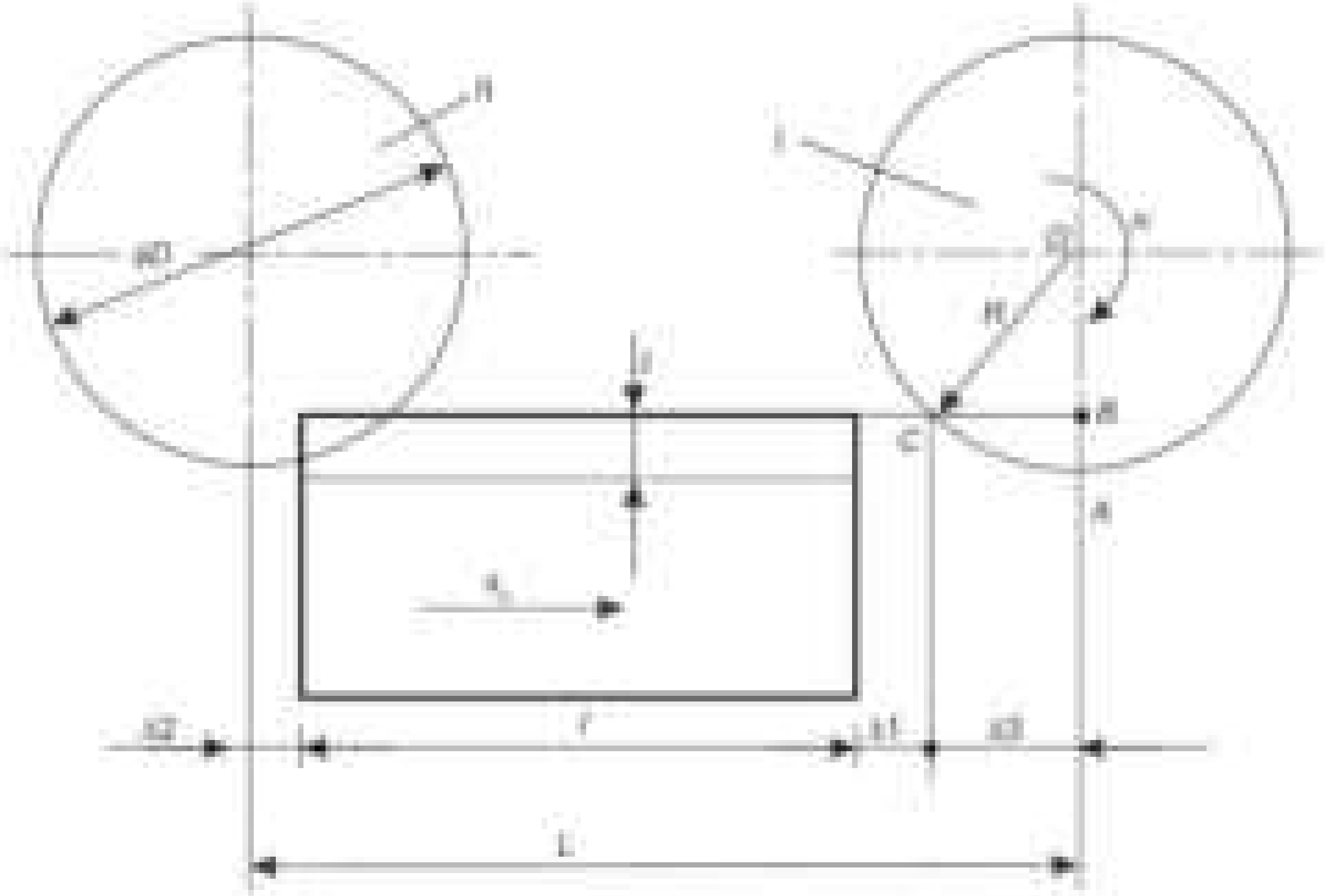


Fig. 1.8 *Plain milling operation*

(b) *Vertical milling machine: Symmetrical face milling operation* (Fig. 1.9)

length of cutter travel $L = l + \Delta 1 + \Delta 2 + \Delta 3$

where $\quad l$ = length of the workpiece

$$\Delta 3 = AB = OA - OB = R - \sqrt{OC^2 - BC^2} = R - \sqrt{R^2 - \left(\frac{B}{2}\right)^2} = 0.5\left(D - \sqrt{D^2 - B^2}\right)$$

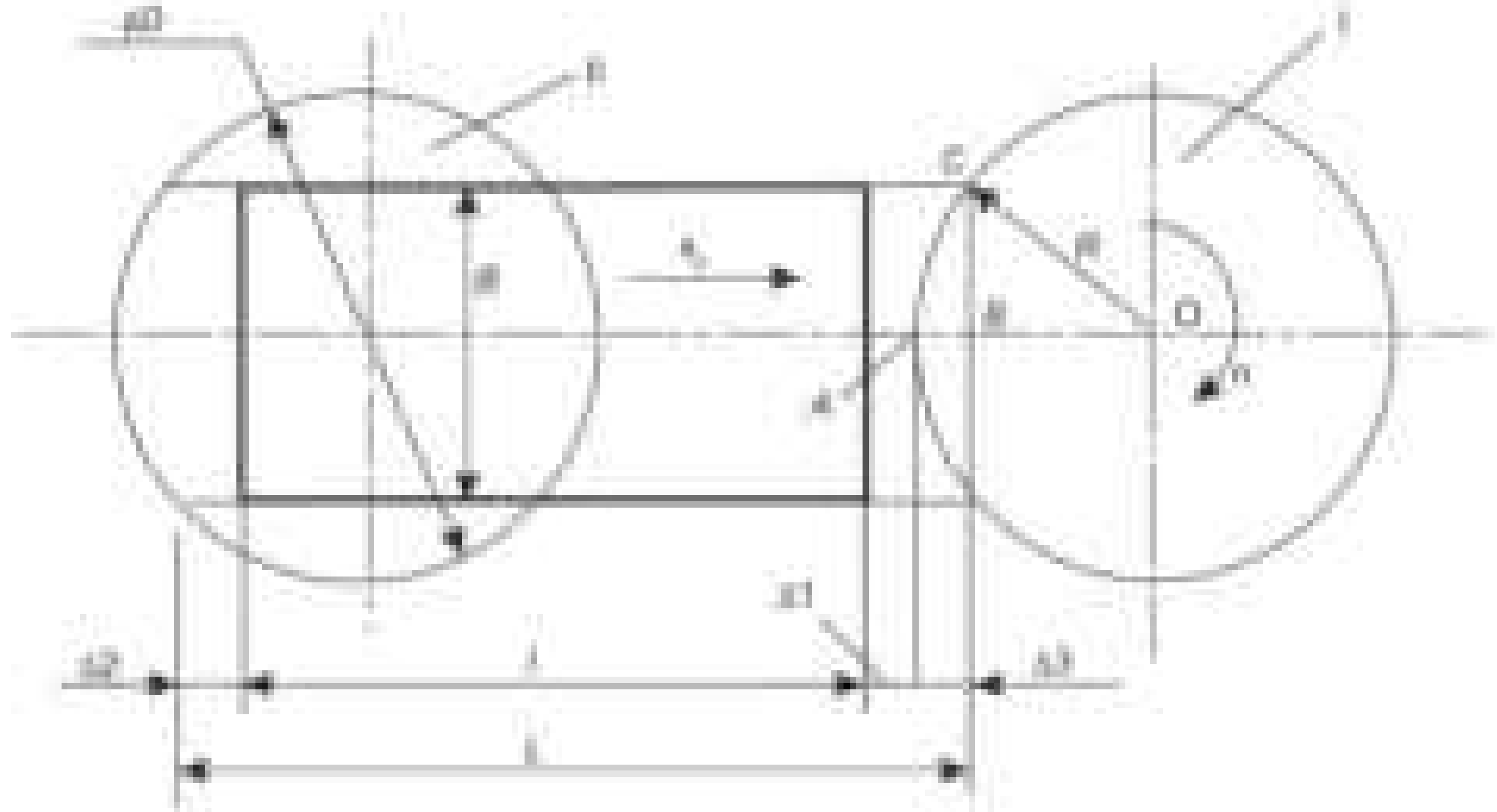


Fig. 1.9 Symmetrical face milling operation

(c) *Vertical milling machine: Asymmetrical face milling operation* $B > \frac{D}{2}$ (Fig. 1.10)
length of cutter travel $L = l + \Delta 1 + \Delta 2 + \Delta 3$
where l = length of the workpiece

$$\Delta 1 = AB = \sqrt{OA^2 - OB^2} = \sqrt{R^2 - (B - R)^2} = \sqrt{R^2 - R^2 - B^2 + 2BR} = \sqrt{B(D - B)}$$

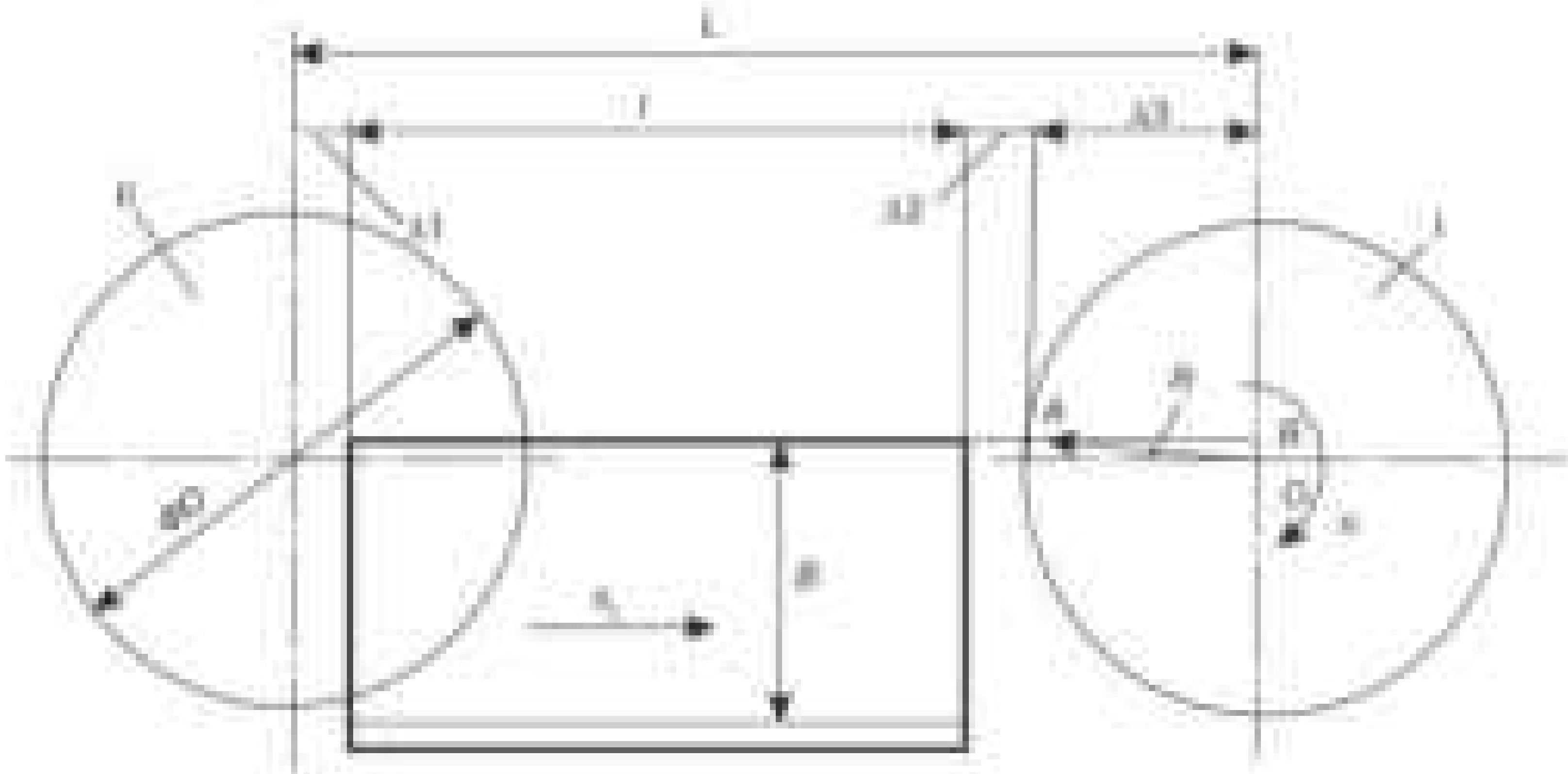


Fig. 1.10 Asymmetrical face milling operation, $B > \frac{D}{2}$

(d) *Vertical milling machine: Asymmetrical face milling operation* $B < \frac{D}{2}$ (Fig. 1.11)
length of cutter travel $L = l + \Delta 1 + \Delta 2 + \Delta 3$

where $\quad l$ = length of the workpiece

$$\Delta 1 = AB = \sqrt{OA^2 - OB^2} = \sqrt{R^2 - (R-B)^2} = \sqrt{R^2 - R^2 - B^2 + 2BR} = \sqrt{B(D-B)}$$

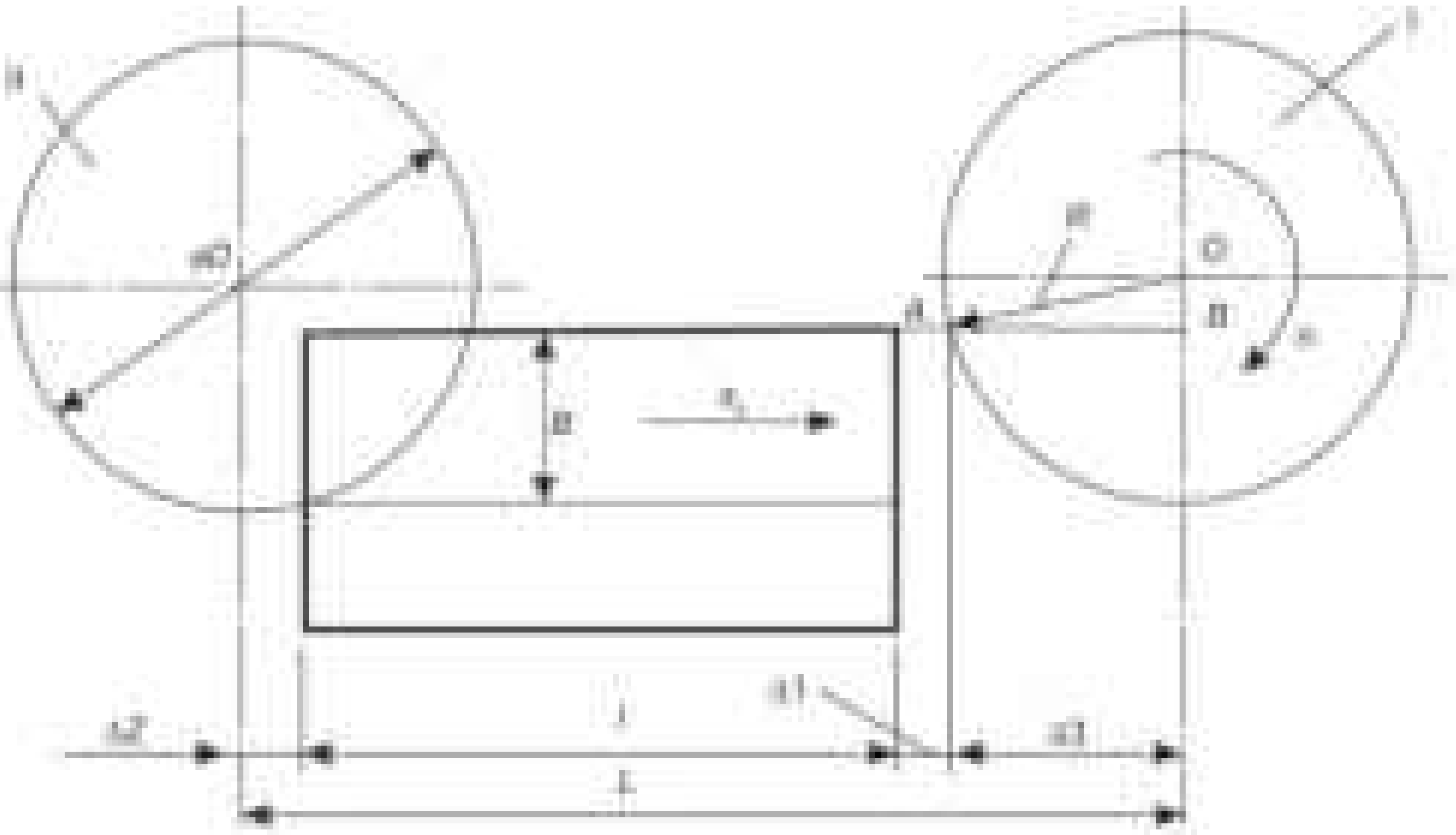


Fig. 1.11 Asymmetrical face milling operation, $B < \dfrac{D}{2}$

Example 1.4

A 200 mm long job is to be machined by a plain milling cutter of diameter $D = 40$ mm and 10 teeth. If the cutting speed is 30 m/min and feed is 0.08 mm/tooth, calculate the machining time for a depth of cut of 4 mm. Assume suitable approach and over travel.

Length of travel $L = 200 + \sqrt{t(D-t)} + \Delta 1 + \Delta 2$

Assuming $\Delta 1 = \Delta 2 = 2$ mm each

$$L = 200 + \sqrt{4(40-4)} + 2 + 2 = 216 \text{ mm}$$

The rpm of the milling cutter is

$$n = \frac{1000v}{\pi D} = \frac{1000 \times 30}{\pi \times 40}$$

Feed per minute $s_m = s_z \times z \times n$

$$= 0.08 \times 10 \times \frac{1000 \times 30}{\pi \times 40} = 191.0 \text{ mm/min}$$

Machining time $T_m = \dfrac{L}{s_m} = \dfrac{216}{191} = 1.13$ min.

Operations on Shaping Machine (Fig. 1.12)

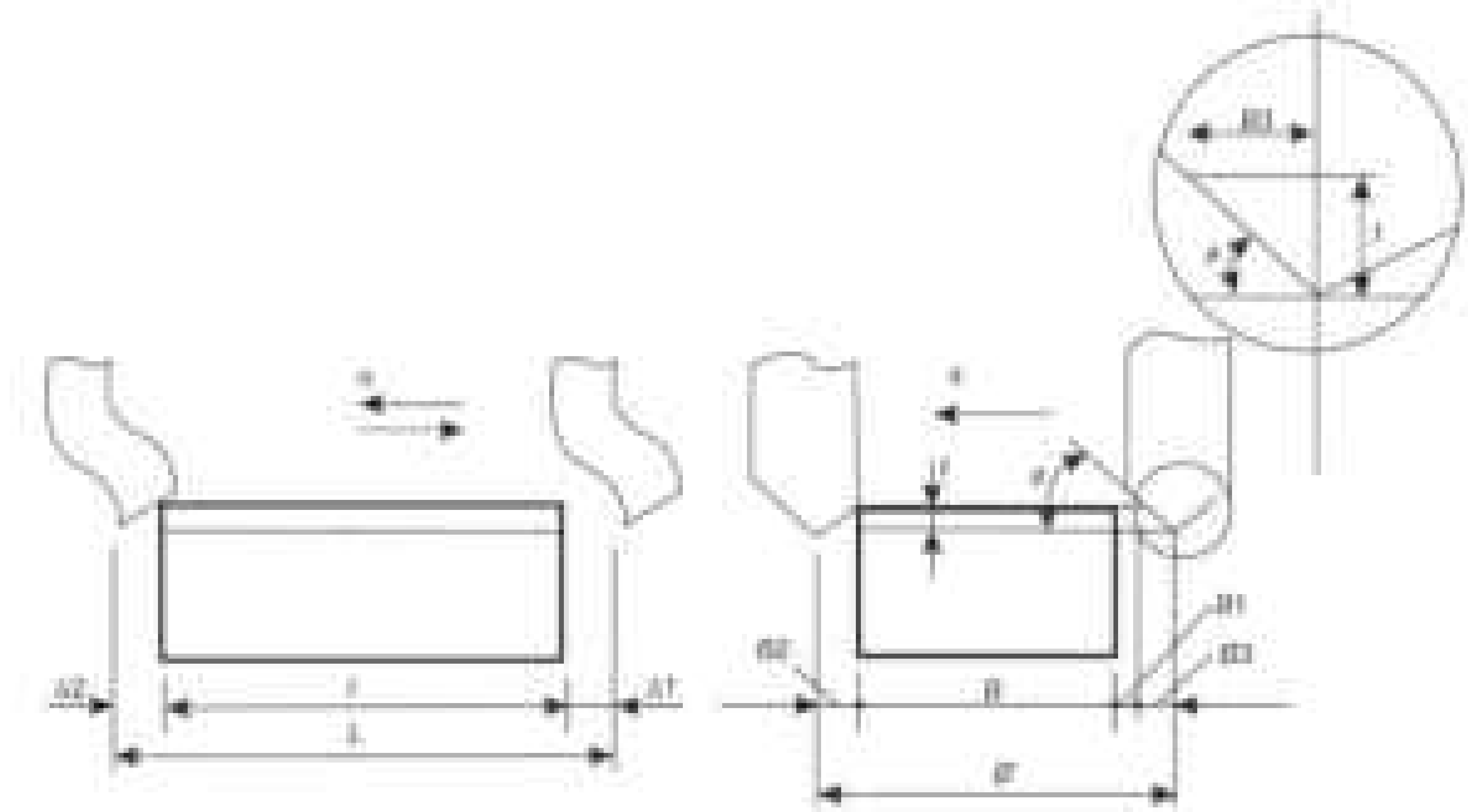


Fig. 1.12 Shaping operation

Machining time $= \dfrac{B'}{s \cdot n}$

where $B' = B + B1 + B2 + B3$

B = width of workpiece

$B1$ = approach, generally equal to 2–3 mm

$B2$ = over travel, generally equal to 2–3 mm

$B3$ = $t \cot \phi$, where t is depth of cut and ϕ is principal or side cutting edge angle; for straight edged tools $\phi = 90°$, hence $\Delta 3 = 0$

s = feed per stroke

n = strokes/min which is found from Eq. (1.4)

The machining time of planing and slotting operations can be determined in a similar manner.

Example 1.5

A 100 mm wide and 200 mm long surface is to be machined on a shaper, using feed per stroke of 0.5 mm. If the cutting speed is 20 m/min and the ratio of return time to cutting time is 1 : 1.25, calculate the time required to machine the job. Assume suitable approach and over travel.

Strokes per minute of the shaper is

$$n = \frac{1000vK}{L(K+1)}$$

For a job of length 200 mm, the typical stroke length will be approximately 20% greater. Hence,

$$L = 1.2 \times 200 = 240 \text{ mm}$$

Therefore, $n = \dfrac{1000 \times 20 \times 1.25}{240(1.25 + 1)} = 46.29$ strokes/min.

Correcting this value to the nearest available value available on the shaper, say 50 strokes/min and assuming that the operation is carried out with a straight edged tool and that $B1 = B2 = 2$ mm each

Machining time $T_m = \dfrac{B + B1 + B2}{} = \dfrac{100 + 2 + 2}{0.3 \times 50} = 6.934$ min.

Operations on Grinding Machine

(a) *Cylindrical Grinding: External-Traverse cut* (Fig. 1.13)

Grinding time $T = \dfrac{Lh}{n_{wp} kBt} K$, min

where L = length of workpiece

$s_l = kB$ mm/rev of workpiece is the longitudinal feed of the reciprocating motion of the workpiece; $k = 0.3 - 0.5$ for rough grinding and $D_{wp} < 20$ mm, $k = 0.7 - 0.85$ for rough grinding and $D_{wp} \geq 20$ mm; $k = 0.2 - 0.4$ for finish grinding

h = allowance, mm;

$t = s_r$ = radial feed/stroke, mm is akin to depth of cut and is given intermittently at the end of stroke, *i.e.*, on traversing the length of the workpiece; typically $t = 0.01–0.025$ mm

K = 1.2 for rough grinding and 1.4 for finish grinding

n_{wp} = rpm of the workpiece

B = width of the grinding wheel

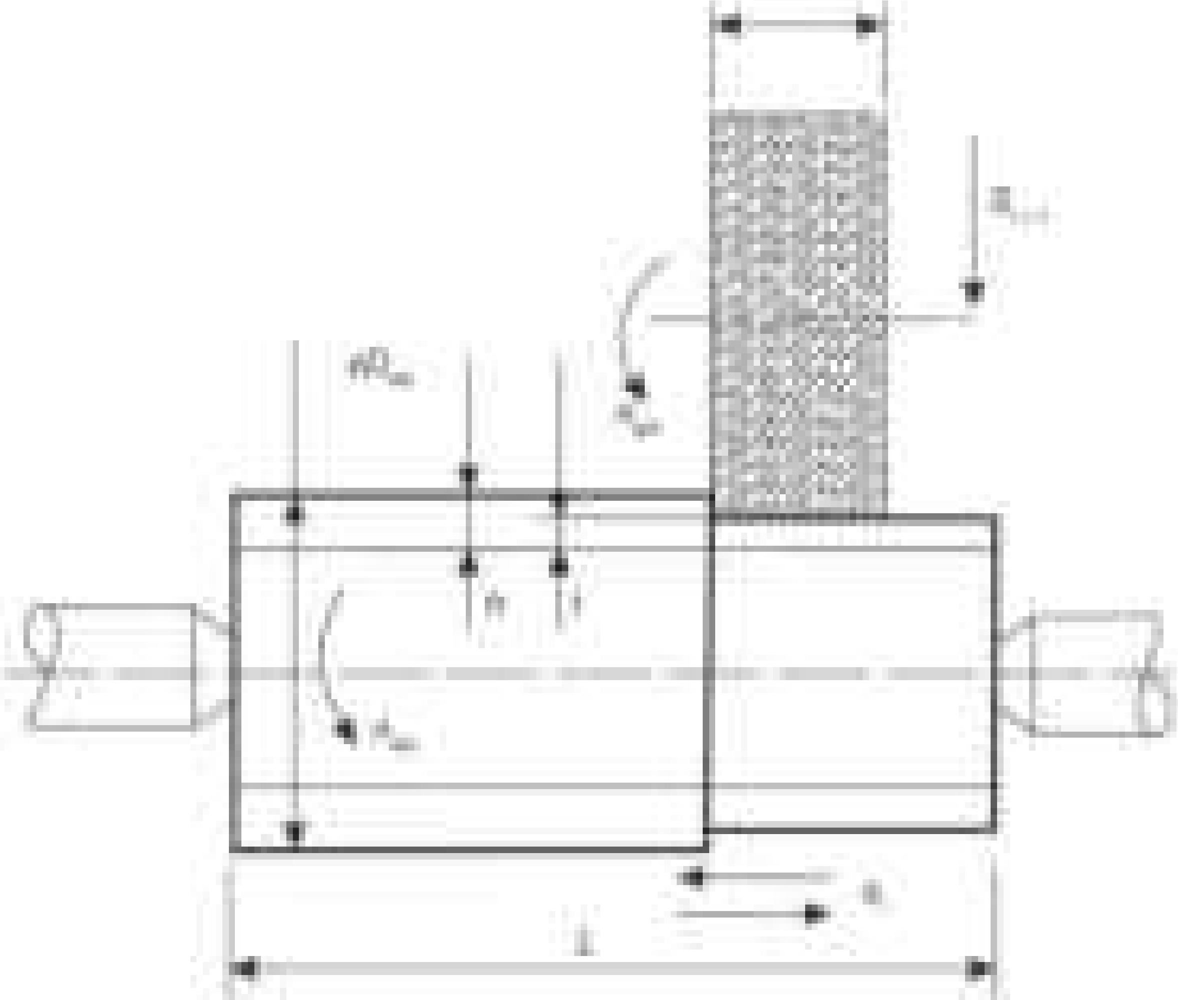


Fig. 1.13 *External cylindrical grinding—traverse cut*

(b) *Cylindrical grinding: external: Plunge cut* (Fig. 1.14)

Grinding time $T = \dfrac{h}{s_t n_{wp}} K,$

where $s_t = 0.0025 - 0.20$ mm per revolution of workpiece is the transverse feed

h, n_{wp} and K are the same as in traverse cut external grinding.

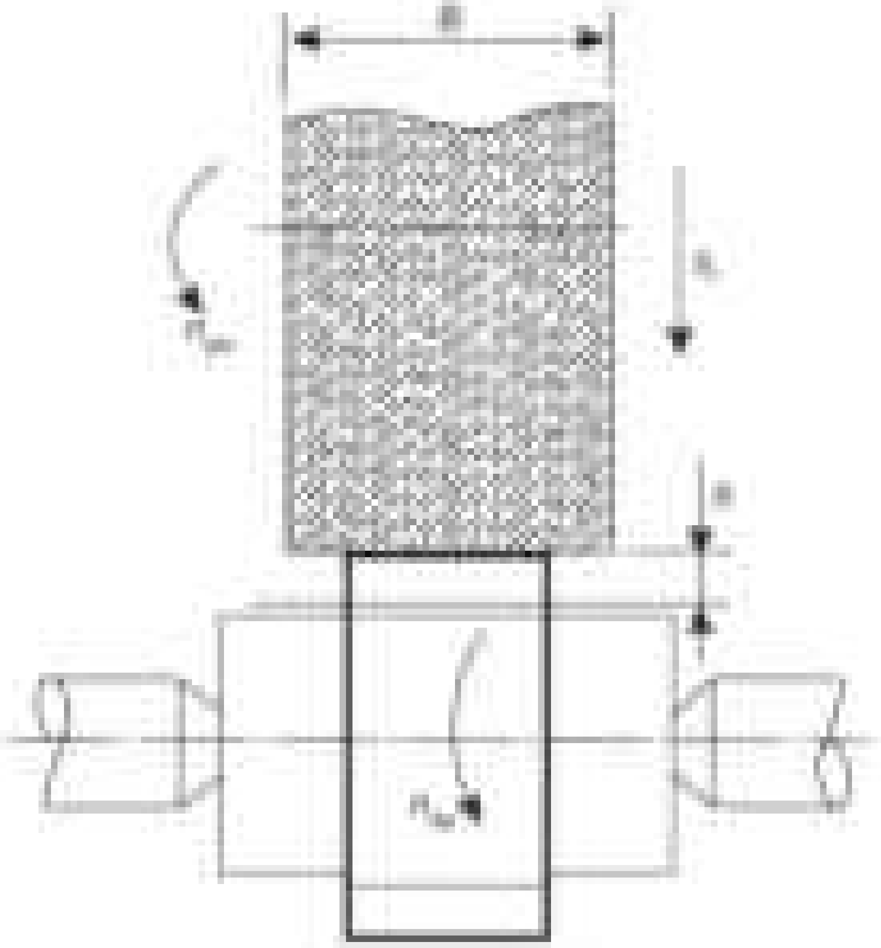


Fig. 1.14 *External cylindrical grinding—plunge cut*

(c) *Cylindrical Grinding: Internal* (Fig. 1.15)

Internal cylindrical grinding is carried out in two ways; with a rotating workpiece or a stationary workpiece. In the latter case, the grinding wheel not only rotates about its own axis, but also executes a planetary motion such that its centre moves along the planetary motion circle (PMC). This method is employed for large workpieces.

Grinding time $T = \dfrac{2Lh}{n_{wp} \lambda B t} K$, min for internal grinding with rotating workpiece

Grinding time $T = \dfrac{2Lh}{n_{PMC} \lambda B t} K$, min for internal grinding with stationary workpiece

where L = length of workpiece

$s_l = \lambda B$ mm/rev of workpiece is longitudinal feed, $\lambda = 0.4$–0.8 for rough grinding and 0.25–0.45 for finish grinding

$s_r = t$ = radial feed/double stroke, mm; typically $t = 0.005$–0.03 mm for rough grinding and 0.002–0.1 for finish grinding. It is given at the end of one complete to-and-fro stroke (double stroke), which explains the presence of $2L$ in the formula of machining time calculation

n_{wp} = rpm of the workpiece
n_{PMC} = rpm of planetary motion of the grinding wheel
B = width of the grinding wheel
h = allowance, mm;
K = 1.3 for rough grinding and 1.6 for finish grinding

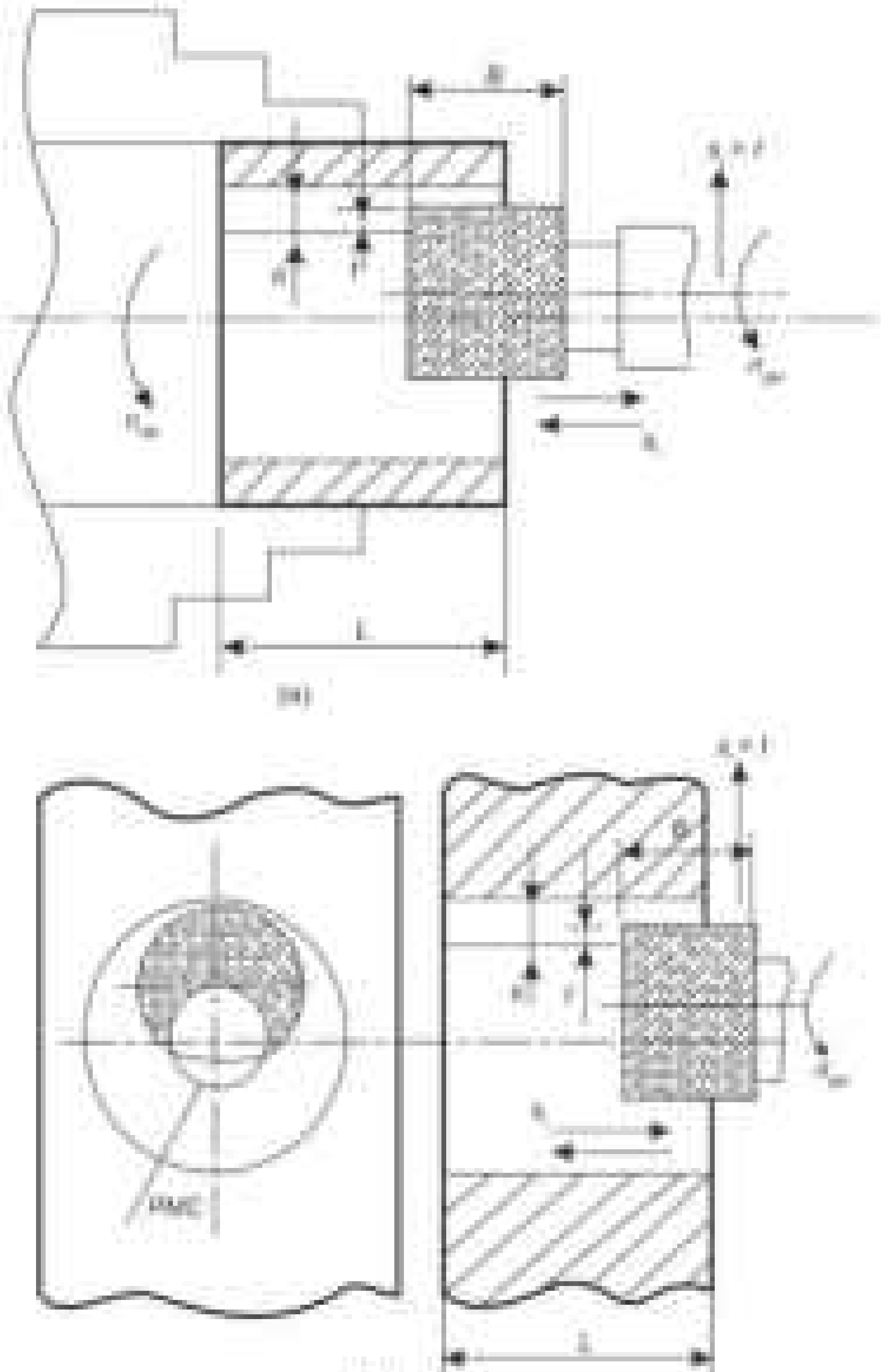


Fig. 1.15 *Internal cylindrical grinding*
(a) with rotating workpiece
(b) with stationary workpiece

(d) *Surface grinding: Peripheral–Planer feed* (Fig. 1.16)

Grinding time $T = \dfrac{LhH}{s_w k B t} K$, min;

where L = length of stroke; $L = l + 10$ mm, where l is length of workpiece

$s_t = kB$, mm/stroke is the transverse feed which is given at the end of stroke, i.e., on traversing the length of the workpiece; k = 0.4–0.7 for rough grinding and 0.25–0.35 for finish grinding

t = 0.015–0.15 mm for rough grinding and 0.005–0.015 for finish grinding. It is akin to depth of cut and is given intermittently at the end of stroke, i.e., on traversing the length of the workpiece

$H = B_{wp} + B + 5$ mm;

s_w = feed of table, mm/min

h = allowance, mm

K = 1.25 for rough grinding and 1.4 for finish grinding

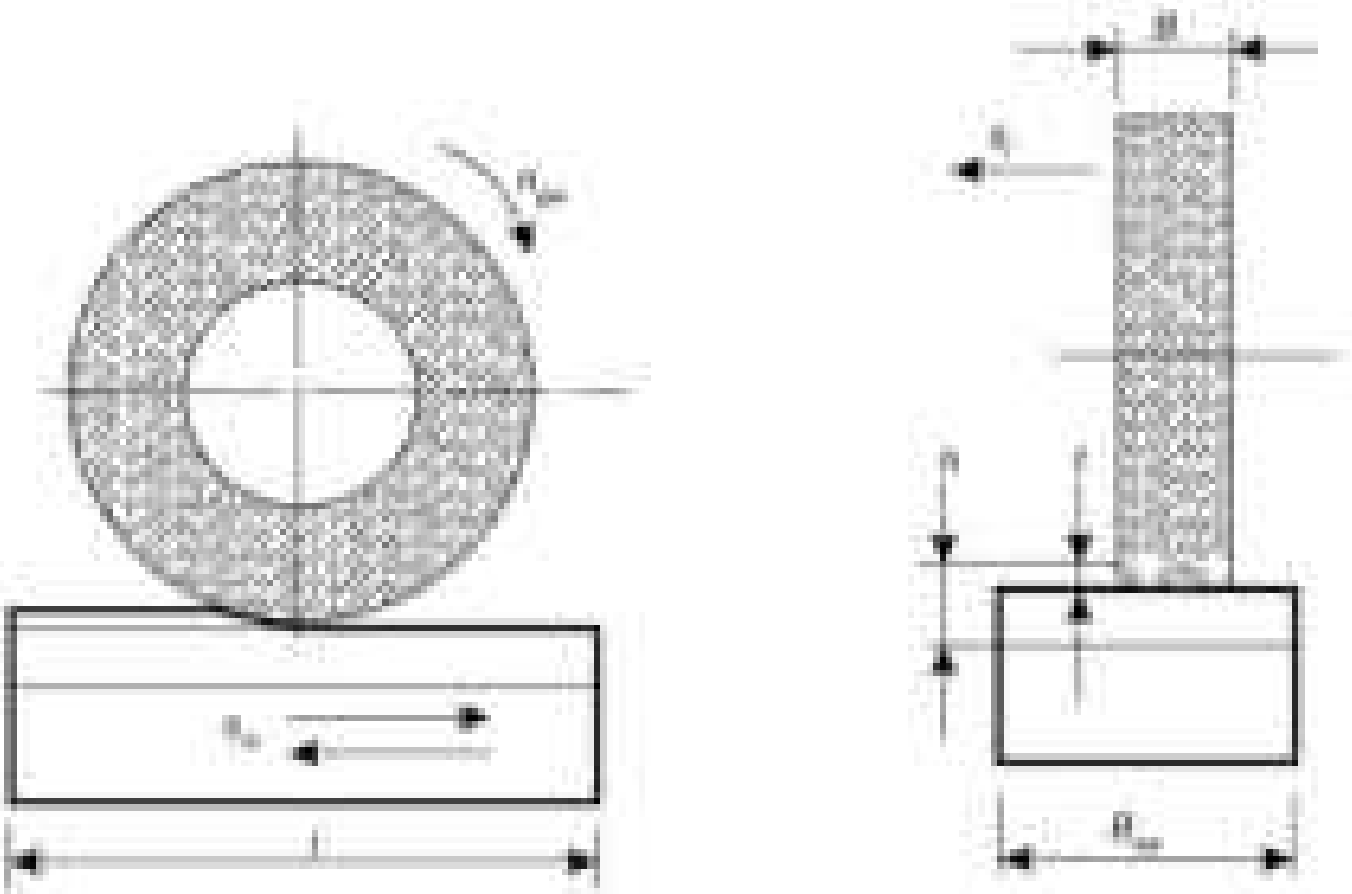


Fig. 1.16 *Peripheral surface grinding*

(e) *Surface grinding: Face–Planer feed* (Fig. 1.17)

Surface grinding with the face of grinding wheel is generally carried out with grinding wheels having diameter D greater than the width of the workpiece B. Therefore, the transverse feed s_t is not required (*see* Fig. 1.17a).

If the feed in depth is given at the end of stroke, i.e., on traversing the length of the workpiece, then grinding time is determined from the expression,

$$T = \frac{Lh}{s_w t} K, \text{ min}$$

where $L = l + \Delta 1 + \Delta 2 + D$ (*see* Fig. 1.17c)

If the feed in depth is given at the end of one complete to-and-fro stroke (double stroke), then grinding time is determined from the expression,

$$T = \frac{2Lh}{v_{sp}t}K; \text{ min}$$

where $L = l + \Delta 1 + \Delta 2 + \Delta 3$;
l = length of workpiece
$\Delta 1$ = approach; generally equal to 2–3 mm
$\Delta 2$ = over travel; generally equal to 2–3 mm
$\Delta 3 = 0.5(D - \sqrt{D^2 - B^2})$ as in symmetrical face milling (see Fig. 1.17b)
v_{sp}, t and h are the same as in peripheral surface grinding

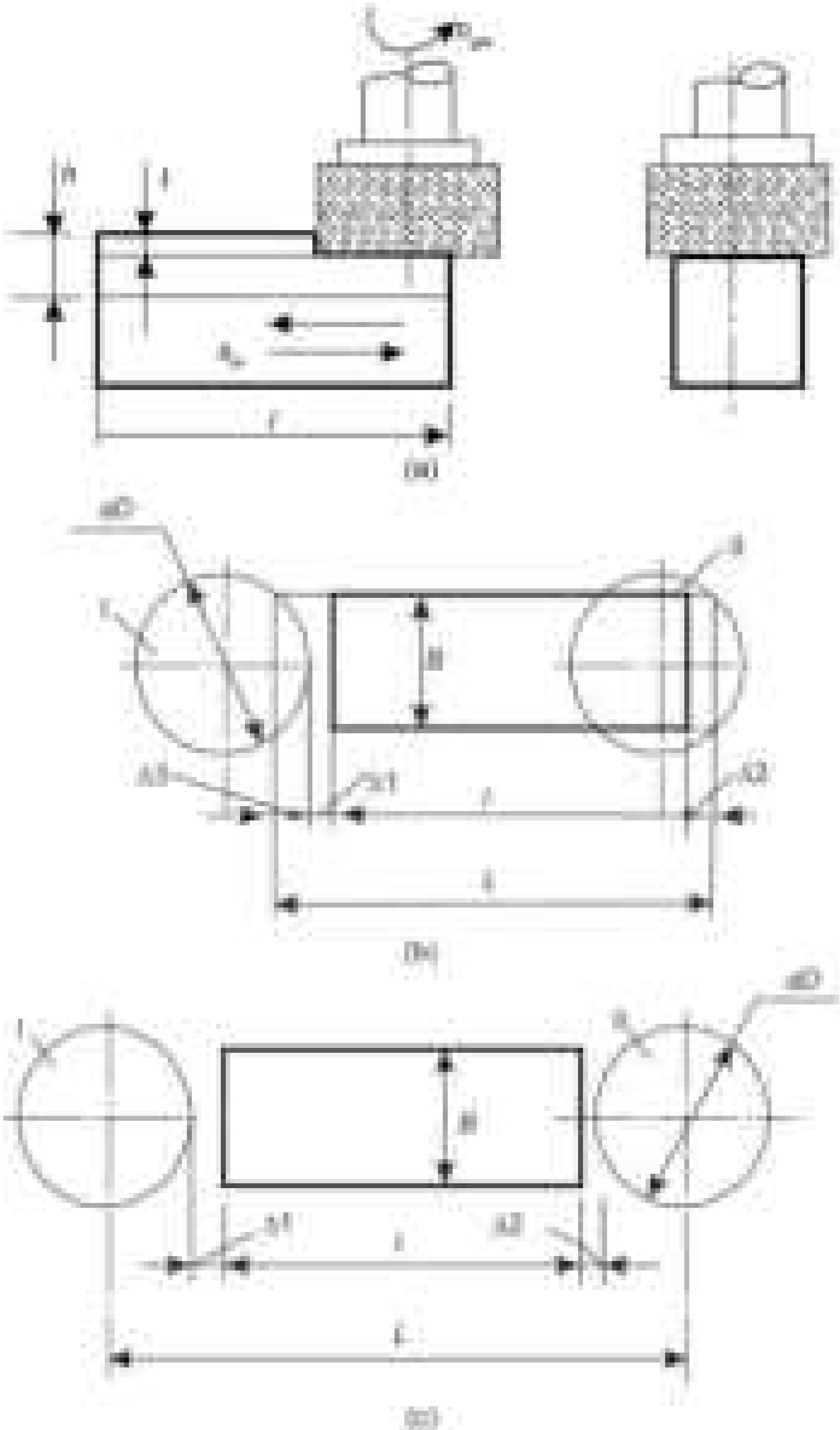


Fig. 1.17 *Face surface grinding*

Example 1.6

A ϕ40 and 210 mm long step is to be machined on a cylindrical grinding machine. Grinding wheel diameter is 600 mm and its width 63 mm. Allowance is 0.2 mm and radial feed 0.005 mm per stroke. Transverse feed (mm per revolution of work) $s_l = kB$, where $k = 0.3$. If peripheral speed of the grinding wheel and workpiece is 30 m/s and 35 m/min, respectively, determine the machining time.

Length of stroke of the table = 210 mm

$$\text{rpm of the workpiece } n_{wp} = \frac{1000 v_{wp}}{\pi D_{wp}} = \frac{1000 \times 35}{\pi \times 40} = \frac{875}{\pi}$$

Allowance $h = 0.2$ mm

Longitudinal feed of reciprocating motion of workpiece $s_l = k \cdot B = 0.3 \times 63 = 18.9$ mm/rev

Radial feed $t = 0.005$ mm/stroke

Assuming it to be case of finish grinding we take $K = 1.4$

$$\text{Machining time } T_m = \frac{Lh}{n_{wp} s_l t} K = \frac{210 \times 0.2}{\dfrac{875}{\pi} \times 18.9 \times 0.005} \times 1.4 = 2.22 \text{ min.}$$

1.3 MACHINE TOOL DRIVES

The machine tool drive is an aggregate of mechanisms that transmits motion from an external source to the operative elements of the machine tool.

The external source of energy is generally a three-phase ac motor which has a rotary motion at its output shaft. The rotary motion of the output shaft of the motor is transmitted to the operative element to provide an appropriate working or auxiliary motion. When the required motion is rotary, the transmission takes place through mechanisms that transfer rotary motion from one shaft to another. However, if a translatory motion is required, the transmission invariably includes a mechanism for transforming rotary motion into translatory.

It is a general requirement for machine tool drives that they should have provision for regulating the speed of travel of the operative elements. The regulation may be available in discrete steps or it may be stepless, i.e., continuous. The former are known as *stepped drives* and the latter *stepless*.

Transmission of motion from the external source to the operative element can take place through mechanical elements, such as gears, chains, belts, etc., or by means of hydraulic and electrical circuits. The drives are correspondingly known as *mechanical*, *hydraulic* and *electrical*. Mechanical drives may be of stepped or stepless type, but hydraulic and electrical drives are invariably stepless in nature.

It may be thus seen that a machine tool drive consists basically of

1. an electric motor, and
2. a transmission arrangement.

The procedure of selecting the electric motor will now be explained followed by a brief description of the elements that constitute the transmission arrangement in mechanical and hydraulic drives. The detailed design of the transmission arrangement will be discussed in Chap. 2.

1.3.1 Selection of Electrical Motor

As stated above, three-phase asynchronous ac motors (also known as induction motors) are generally used as the source of power in machine tools. The power rating of the electric motor in general-purpose machine tools is calculated by the formula

$$N_m = \frac{N_c}{\eta}\ \text{kW} \tag{1.8}$$

where N_m = power rating of the electric motor, kW

N_c = total power required for removing metal, kW

η = coefficient of efficiency of the drive

The power spent on a cutting operation consists of the power required to overcome each component of the cutting force. In general, the cutting force can be resolved into three mutually perpendicular components P_x, P_y, and P_z. In a simple turning operation let P_z be the component of the cutting force coinciding with the velocity vector, P_x—the component coinciding with the direction of axial feed and P_y—the component coinciding with the direction of radial feed. Let the corresponding velocities be v, s_a and s_r, where v is the cutting speed, s_a the feed in the axial direction and s_r the feed in the radial direction. The power required for the cutting operation will be

$$N_c = \frac{P_z \cdot v}{60 \times 75 \times 1.36} + \frac{P_y \cdot s_r \cdot n}{60 \times 75 \times 1.36 \times 1000} + \frac{P_x \cdot s_a \cdot n}{60 \times 75 \times 1.36 \times 1000}\ \text{kW}$$

The first factor on the right-hand side represents the power required for removing metal, while the second and third factors represent the power required for the feed motion in radial and axial directions, respectively. In a cylindrical turning operation $s_r = 0$, hence the second factor becomes zero. Also, the third factor is generally negligibly small as compared to the first, and therefore, the simplified expression for the motor power rating can be written as

$$N_c = \frac{P_z \cdot v}{6120}\ \text{kW} \tag{1.9}$$

The value of N_c calculated from Eq. (1.9) should be increased by about 5% to accommodate the power requirements of the feed motion.

The value of η may be expressed as

$$\eta = \eta_1 \cdot \eta_2 \cdot \eta_3 \cdot \ldots \cdot \eta_n$$

where η_1, η_2, η_3, ... η_n are the coefficients of efficiency of the individual transmissions involved in transmitting motion from the motor to the operative element. These values for different transmissions and supports are given in Table 1.1.

Table 1.1 *Values of coefficient of efficiency for various transmission and supports*[1]

Type of Transmission or Support	*Coefficient of Efficiency*
Belt drive with flat belt	0.98
Belt drive with V-belt	0.96
Spur gear drive	0.98
Helical gear drive	0.97
Bevel gear drive	0.96
Ball or roller bearing	0.995
Crank and slider mechanism	0.90
Jaw clutch	0.95
Multiple-disc friction clutch operating in oil	0.90

The overall transmission efficiency generally lies between $\eta = 0.8$–0.85 for machine tools with rotary primary cutting motion and $\eta = 0.6$–0.7 for machine tools with reciprocating primary cutting motion.

The value of N_c for various cutting operations (for different cutting tool and workpiece materials) can be determined from empirical formulae that are available in textbooks on the theory of metal cutting. The power required for cutting should be calculated for the following conditions: rough machining of a soft material with cemented carbide tool using maximum workpiece diameter (in lathes and boring machines), maximum cutter diameter (in milling and drilling machines) and maximum stroke length (in shaping and planing machines).

On the basis of the calculated N_{em} value, a standard induction motor is selected having the nearest available power rating that is equal to or slightly greater than the calculated value.

The selection of electric motors that work under conditions of variable loading is done by a different procedure[1] which is explained below. This procedure is expedient for single purpose and special machine tools that machine identical parts according to a set sequence of operations which is continuously repeated. An example of such a loading is shown in Fig. 1.18.

The power rating of the motor is determined from considerations of permissible overloading and heating, and the higher of the two values is taken for selecting the motor.

From Consideration of Overloading

$$N_{em} = \frac{N_{max}}{\eta \cdot \lambda} \tag{1.10}$$

where N_{max} = maximum power required in the whole cycle (N_3 in the example of Fig. 1.18)

λ = permissible overloading coefficient for the given type of motor

η = coefficient of efficiency of the drive

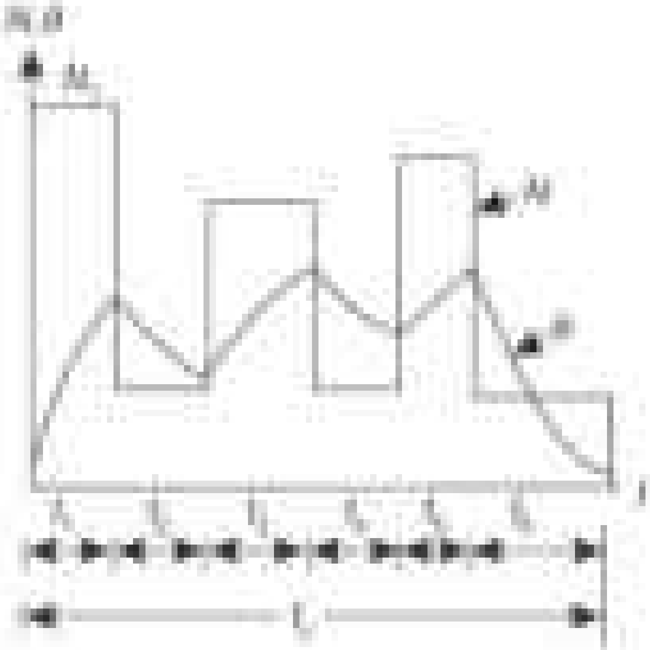


Fig. 1.18 *Variable loading cycle in which the motor temperature comes down to the ambient temperature*

From Consideration of Heating The calculation of power rating from the consideration of heating consists in determining an equivalent average load of constant value such that heating of the motor due to this load is equal to the sum of heat due to individual load components of the variable loading cycle. The power rating is determined by the expression

$$N_{eq} = \frac{1}{\eta}\sqrt{\sum_{i=1}^{n} \frac{N_i^2 t_i}{t_c}} \tag{1.11}$$

where N_{eq} = equivalent power rating
N_i = power required for ith sequence of the variable loading cycle
t_i = duration of the ith sequence of the variable loading cycle
t_c = cycle time
n = total number of sequences in the cycle
η = coefficient of efficiency of the drive

In the variable loading cycle of Fig. 1.18, the time ratio of cutting and idle sequences of the cycle was such that at the end of the cycle, the motor temperature came down to the temperature of the surrounding atmosphere. However, the variable loading cycle may be such that the motor temperature tends to acquire a more or less stable value higher than that of the surrounding atmosphere (Fig. 1.19). In such cases, it is possible to select a motor used for continuous loading based upon the equivalent power rating calculated from Eq. (1.11). However, special motors can also be employed. These special motors are alternatively switched on and off, and are characterised by the ratio,

$$\varepsilon = \frac{t_{on}}{t_{on} + t_{off}} \times 100$$

where t_{on} = time during which motor remains switched on
t_{off} = time during which motor remains switched off

Generally, standard motors are manufactured for ε values of 15, 25, 40, and 60%.

The value of ε according to the time of cutting and idle sequences in the variable loading cycle is determined as

$$\varepsilon_a = \frac{t_c}{t_c + t_i} \times 100$$

In general, the ε_a value for a cycle differs from the standard value provided on the motor. Therefore, the nearest standard value of ε_{st} is selected and the power rating is determined from the expression,

$$N_m = \frac{1}{\eta} N_a \sqrt{\frac{\varepsilon_a}{\varepsilon_{st}}} \tag{1.12}$$

where $N_a = N_{eq} / \sqrt{\varepsilon_a}$; N_{eq} being the equivalent power rating calculated for the given cycle from the consideration of heating as discussed above.

In machine tools, electric motors are selected by this method for $\varepsilon_a < 60\%$. For higher values of ε_a, the motor is selected on the basis of the equivalent power rating from the consideration of heating.

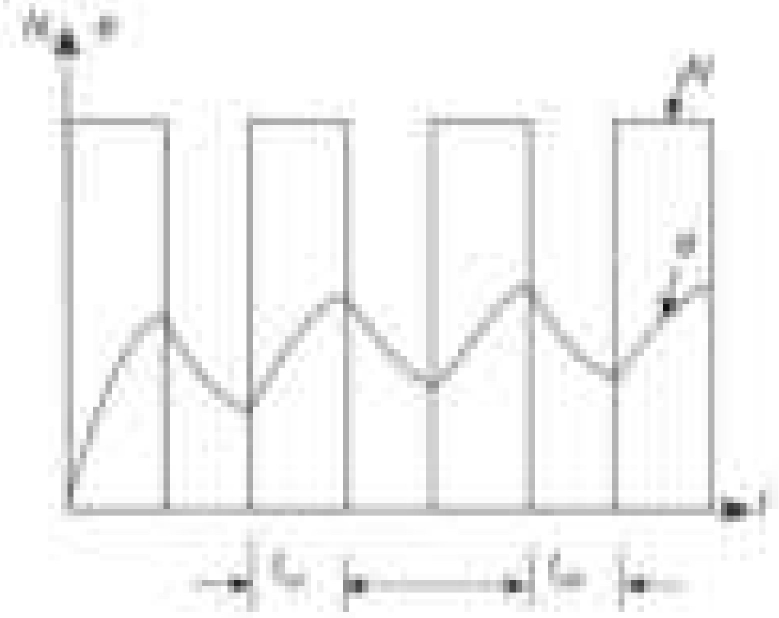


Fig. 1.19 *Variable loading cycle in which the motor temperature acquires a stable value greater than the ambient temperature*

1.4 HYDRAULIC TRANSMISSION AND ITS ELEMENTS

Hydraulic transmission is used in machine tools for providing rotary as well as translatory motion, although the latter application is more common. Hydraulic transmission, as a rule, provides stepless regulation of the speed and feed rate.

The functioning of a rotary hydraulic drive can be explained with the help of Fig. 1.20. The electric motor rotates the rotor of vane pump through gear pair Z_1/Z_2. During rotation, the pump sucks in oil from the reservoir and delivers it under pressure to the hydraulic motor. The hydraulic motor is, in principle, another vane pump mounted in the reverse manner, so that oil delivered under pressure rotates its vanes and hence the rotor. From the output shaft of the hydraulic motor, rotary motion is transmitted to the machine-tool spindle through a belt drive. A pressure valve in the delivery line limits the maximum pressure at which oil is delivered to the hydraulic motor. The actual pressure can be read on the pressure gauge.

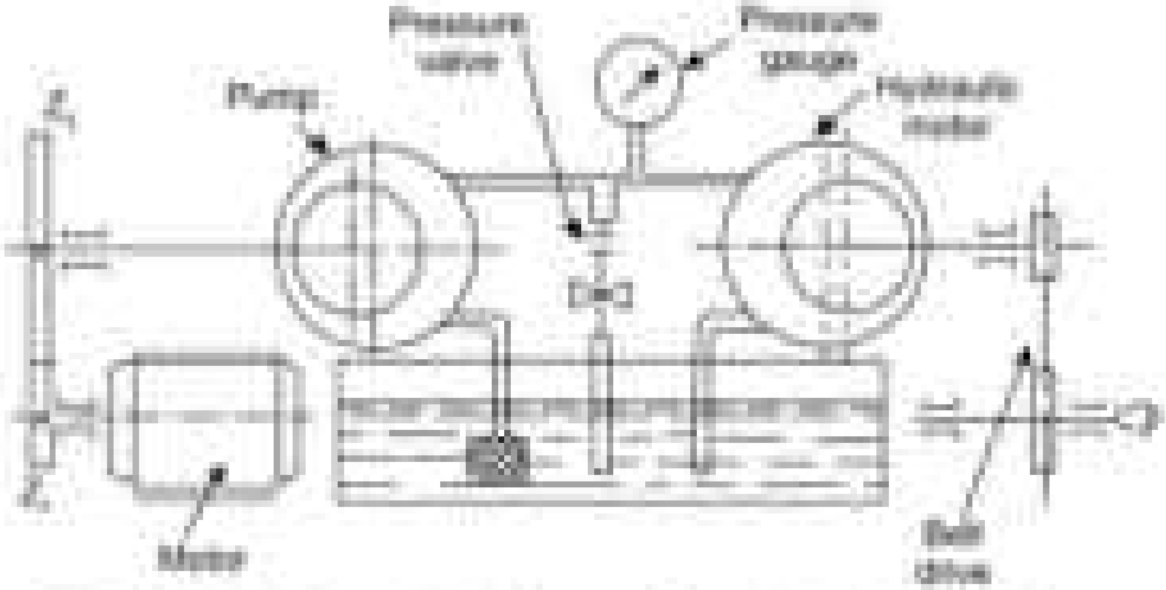


Fig. 1.20 *Schematic diagram of a rotary hydraulic drive*

The principle of operation of a translatory hydraulic drive is discussed below. The drive (Fig. 1.21) consists of a gear pump which sucks oil from the reservoir and delivers it to the direction control value through a throttle. The function of the throttle is to enable regulation of the speed of travel of the operative element. In the position of the control valve drawn by firm lines, oil is delivered into the right-hand chamber of the hydraulic cylinder, moving the piston towards the left. The machine-tool table which is rigidly attached to the piston is also moved leftwards. Oil from the left-hand chamber of the hydraulic cylinder returns to the reservoir through the direction-control valve. It can be seen from Fig. 1.21 that the control-valve piston is coupled to the operative element by means of a rocking lever. Therefore, the leftward movement of the machine-tool table is accompanied by a movement of the control-valve piston in the same direction. The leftward movement of the table stops when the control-valve piston comes to occupy the position shown by dotted lines. In this position, oil begins to flow in the left-hand chamber of the hydraulic cylinder, pushing the piston rightwards, thus reversing the direction of translatory motion of the table. The hydraulic circuit has a pressure valve to drain off excessive oil which does not pass through the throttle aperture.

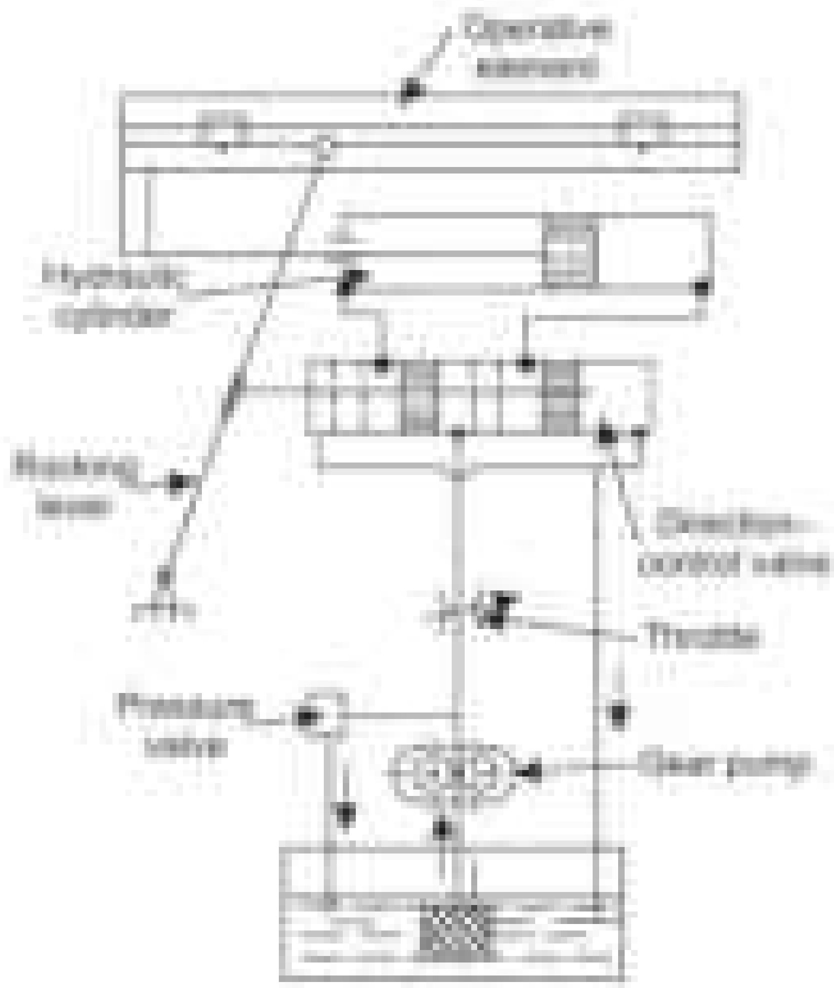


Fig. 1.21 *Schematic diagram of a translatory hydraulic drive*

From the description of simple rotary and translatory motion hydraulic drives, it may be concluded that these drives are made up of individual elements and units which are appropriately joined into a circuit by means of pipe lines. The important elements of a hydraulic transmission are:

1. Pumps
2. Hydraulic cylinders
3. Direction-control valves
4. Pressure valves
5. Throttles

These elements will now be dealt with to the extent necessary for a proper appreciation of their application in machine tools. Besides these elements, the hydraulic circuits of machine tools include auxiliary elements, such as filters, accumulators, seals and packings, relays, etc. Students are advised to consult a basic text on hydraulics and hydraulic machines for a detailed insight into the functioning of the hydraulic equipment.

1. Pumps The pumps primarily serve the purpose of sucking oil and delivering it under pressure to various hydraulic devices. On the basis of the operating principle, pumps can be classified as constant delivery pumps and variable delivery pumps.

The constant delivery pumps generally employed in machine tools are gear pumps and vane pumps. The working principle of a gear pump can be explained with the help of Fig. 1.22. The pump consists of a pair of meshing gears of which the driving gear is directly coupled to an electric motor. The oil is sucked into the gap between the meshing teeth on the suction side and squeezed out under pressure on the delivery side.

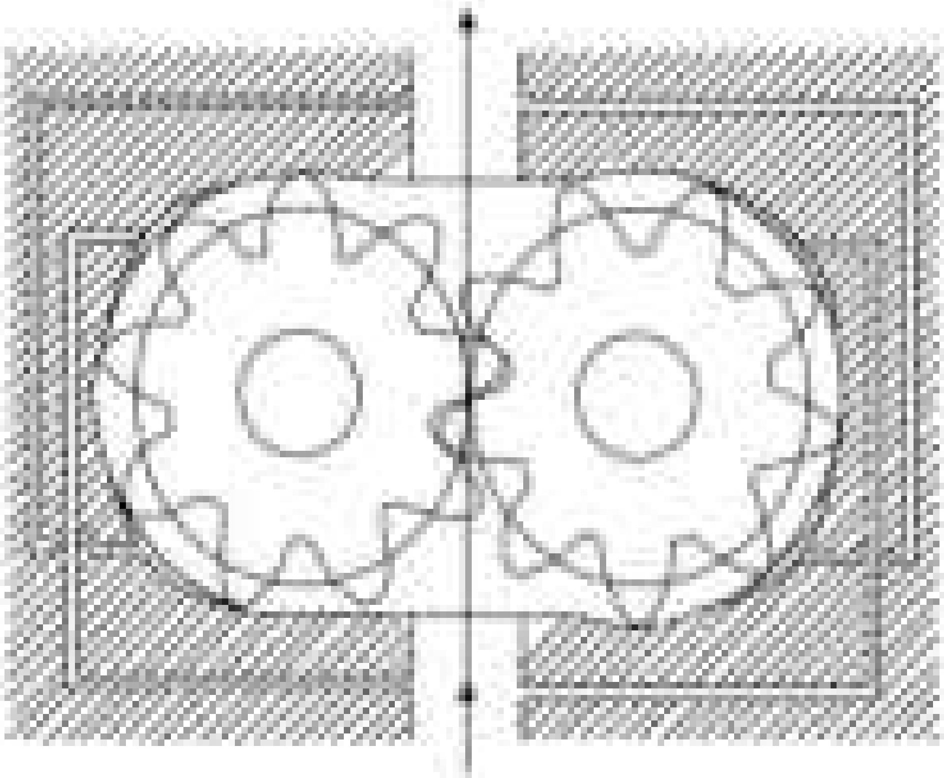


Fig. 1.22 *Schematic diagram of a gear pump*

The volume of oil delivered by a gear pump is given by the expression,

$$Q = \frac{\pi d_0 (d_e - d_0)}{10^9} \cdot b \cdot n \text{ m}^3/\text{min} \tag{1.13}$$

where d_0 = pitch circle diameter of the gears, mm

d_e = addendum circle diameter of the gears, mm

b = width of gears, mm
n = rpm of the driving gear

The power rating of the motor required to run a pump is determined from the expression,

$$N = \frac{p \cdot Q}{6 \times 10^4 \eta_m \cdot \eta_v} \text{ kW} \tag{1.14}$$

where p = pressure developed by the pump, N/m^2
Q = volume of oil delivered by the pump, m^3/min
η_m = coefficient of mechanical efficiency of the pump; generally η_m = 0.7–0.9
η_v = coefficient of volumetric efficiency of the pump (leakage losses), generally η_v = 0.7–0.8

The schematic diagram of a constant delivery vane pump is shown in Fig. 1.23. The rotor mounted on a splined shaft rotates inside the stator, whose profile is shown in Fig. 1.24. As the rotor rotates, the vanes reciprocate radially and complete two complete cycles of suction and delivery in one revolution of the rotor. Pockets 1 and 2 serve for suction, and 3 and 4 for delivery.

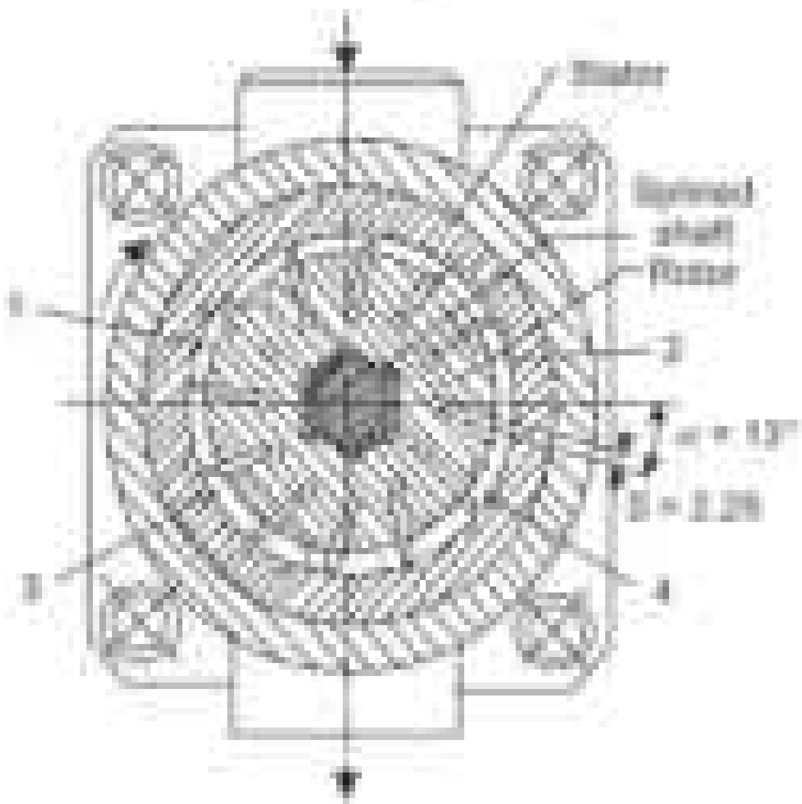


Fig. 1.23 Schematic diagram of a constant-delivery vane pump

The volume of oil delivered by a constant delivery vane pump is given by the expression,

$$Q = \frac{2Bn}{10^6}\left[\pi(r_2^2 - r_1^2) - \frac{(r_2 - r_1)b \cdot z}{\cos\alpha}\right] \text{m}^3/\text{min} \tag{1.15}$$

where B = width of rotor, mm
n = rpm of rotor
r_2 = major semi-axis of the stator profile, mm
r_1 = minor semi-axis of the stator profile, mm

t = thickness of vanes, mm
z = number of vanes
α = angle which the vane makes with the radius, generally $\alpha = 13°$

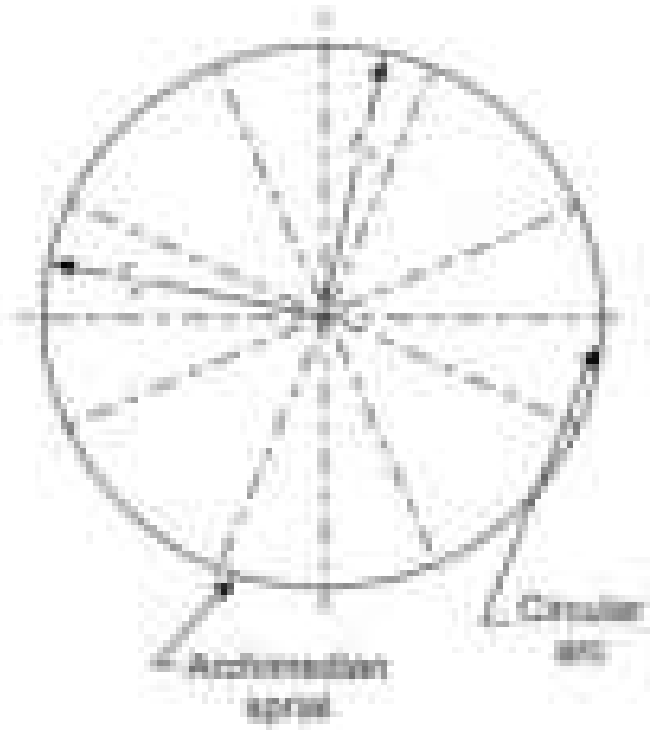


Fig. 1.24 *Profile of the stator of a constant-delivery vane pump*

The variable delivery pumps commonly used in machine-tool hydraulic drives are vane pumps and radial piston pumps.

The schematic diagram of a variable delivery vane pump is shown in Fig. 1.25. The vanes reciprocate in radial slots of the rotor which is eccentrically mounted with respect to the stator. The rotor axis is generally fixed but the stator can be displaced to vary eccentricity and hence pump delivery. The stator in this case has a circular profile, and therefore, no delivery takes place if the rotor and stator axes become concentric. The radial reciprocation of vanes is controlled by means of rollers, attached to the vanes, that move in an annular guiding ring concentric with the stator. The volume of oil delivered by a variable delivery vane pump is given by the expression,

$$Q = \frac{2en}{10^6}[B(\pi D - tz) + 4\pi bd] \text{ m}^3/\text{min} \tag{1.16}$$

where B = width of rotor, mm
n = rpm of rotor
e = eccentricity, mm
D = stator bore, mm
d = diameter of rollers, mm
b = width of annular guiding ring, mm
t = thickness of vanes, mm
z = number of vanes

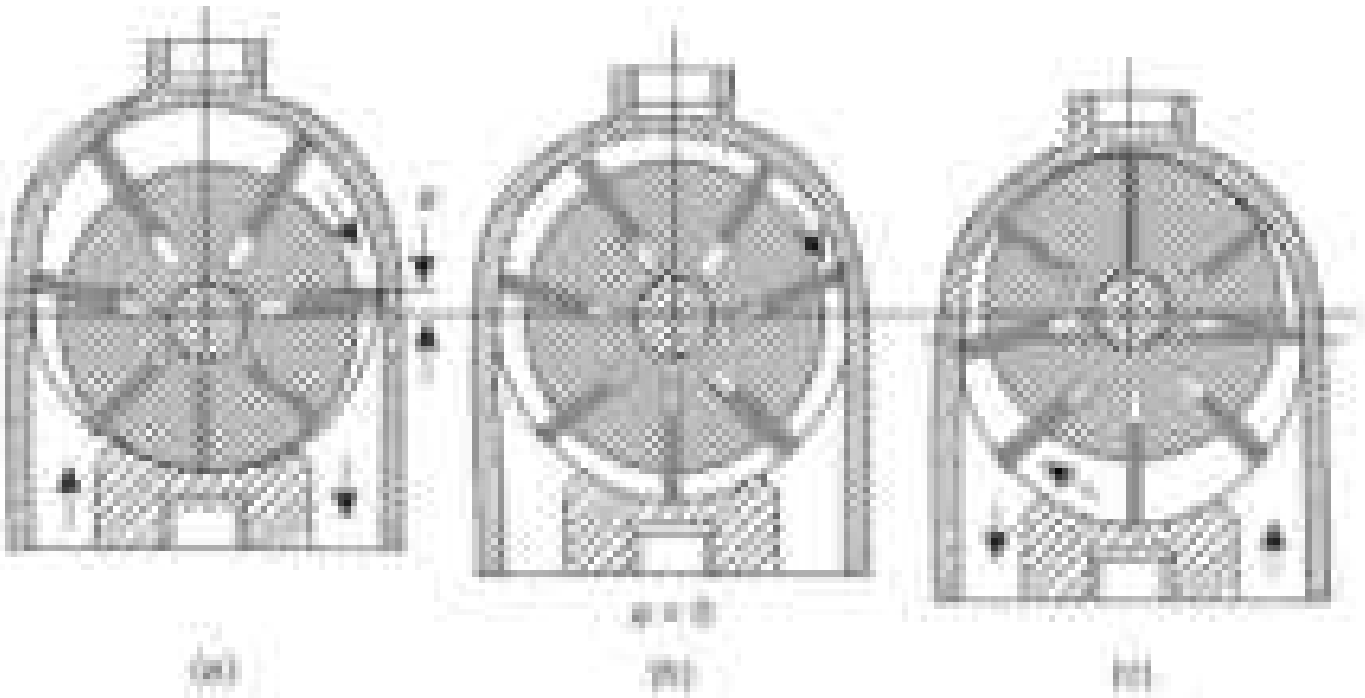


Fig. 1.25 Schematic diagram of a variable-delivery vane pump

The working principle of the radial piston pump is similar to that of the variable delivery vane pump. The only difference is that the vanes are replaced by mini pistons, each of which reciprocates in its cylinder. The manufacture of cylindrical sliding surfaces of pistons and cylinders is easier than that of rectangular vanes. Therefore, piston pumps can be manufactured with tighter fits and are distinguished by lower leakage losses. The volume of oil delivered by a radial piston pump may be determined by the expression,

$$Q = \frac{\pi d^2 ezn}{2 \times 10^3}\ \mathrm{m^3/min} \tag{1.17}$$

where d = diameter of pistons, mm
e = eccentricity, mm
z = number of pistons
n = rpm of rotor

Gear pumps are used for pressures up to 100 kgf/cm^2, vane pumps for pressures up to 25 kgf/cm^2 and piston pumps for pressures up to 140 kgf/cm^2.

All pumps described above can in principle be used as hydraulic motors by reversing their operation. However, in practice, only variable delivery vane pumps and radial piston pumps are used because they ensure a wider range of speed regulation and also have higher efficiency than gear pumps, especially at low speeds.

2. Hydraulic Cylinders Hydraulic cylinders are used in hydraulic drives where translatory motion of the operative element (generally of the machine-tool table) is required. A simple cylinder with the piston rod only on one side (Fig. 1.26a) provides different piston velocities in two directions, while a double-end rod cylinder (Fig. 1.26b) provides identical piston velocity in both directions.

The piston speed and flow rate of oil to the cylinder are related as follows:

$$Q = A \cdot v \tag{1.18}$$

where Q = amount of oil fed to the cylinder per unit time, m^3/min

A = effective area of cross section of the piston, m^2

v = velocity of piston, m/min

The minimum pressure required to move the piston can be determined from the expression,

$$p = \frac{P}{A} \text{ kgf/cm}^2$$

where P = resisting force, kgf

A = effective area of cross section of the piston, cm^2

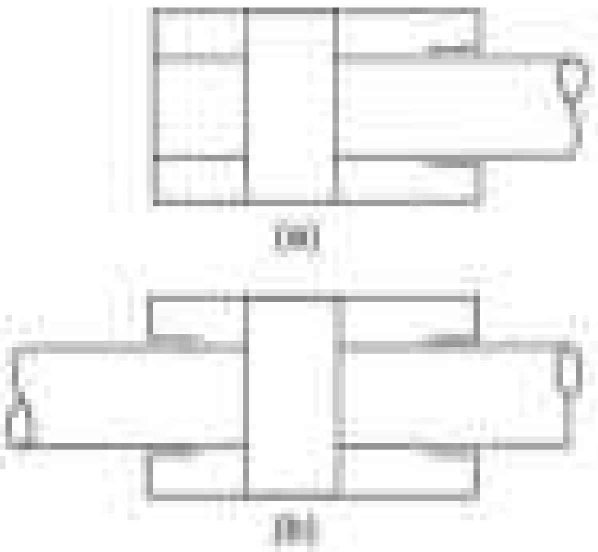


Fig. 1.26 *Cylinders: (a) Single-piston rod type (b) Double-end rod type*

3. Direction-control Valves The function of these valves is to change the direction of fluid flow. Direction-control valves are generally available in two design versions—with a rotary spool and with a sliding piston.

The working of a rotary, spool-type direction-control valve can be explained with the help of its schematic diagram shown in Fig. 1.27. The valve is divided into two halves by a partition. The valve has four ports 1, 2,

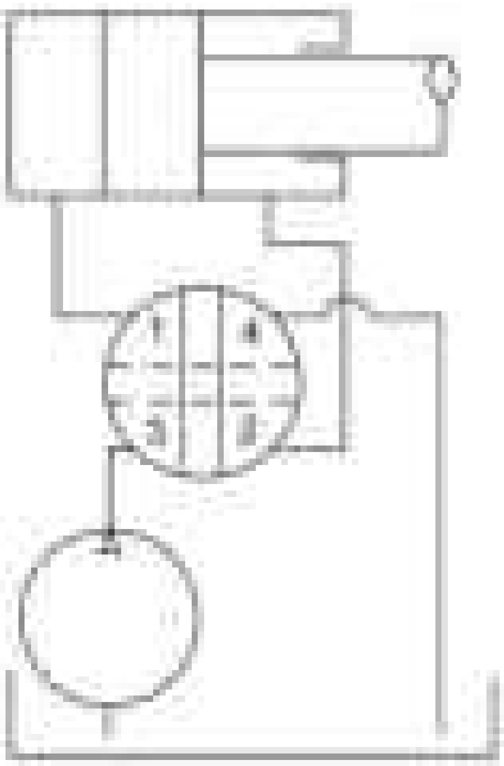


Fig. 1.27 Schematic diagram of rotary, spool-type direction-control valve

3, 4, of which ports 1 and 2 are connected to the two chambers of the hydraulic cylinder, while ports 3 and 4 are connected to the pump line and reservoir, respectively. The direction of oil flow is reversed by rotation of the partition inside the valve body. When the partition occupies the position shown in Fig. 1.27 by firm lines, port 1 is connected to the pump and oil is delivered to the left-hand chamber of the cylinder; at the same time the oil in the right-hand chamber of the cylinder is discharged into the reservoir through ports 2 and 4. When the partition occupies the position, depicted in Fig. 1.27 by dotted lines, the port connections get reversed, i.e., the pump gets connected to the right-hand chamber of the cylinder through port 2, while the oil in the left-hand chamber is discharged into the reservoir through ports 1 and 4. The direction of travel of the piston is thus reversed by shifting the partition from one position to the other.

The working of a four-way, two-position, piston-type direction-control valve was explained while discussing the translatory motion hydraulic drive of Fig. 1.21. This valve (Fig. 1.28) has five ports. Ports 1 and 2 are connected to the left- and right-hand chambers of the hydraulic cylinder, respectively. Port 3 is connected to the pump line, while ports 4 and 5 are interconnected and serve for draining oil into the reservoir. In the position of the piston shown by firm lines, oil is fed into the left-hand chamber of the cylinder through port 1 and the oil from the right-hand chamber is drained into the reservoir through ports 2 and 5. When the piston occupies the position shown by dotted lines, port 2 gets connected to the pump line, thus delivering oil to the right-hand chamber of the cylinder, while the oil in the left-hand chamber is drained back to the reservoir through ports 1 and 4.

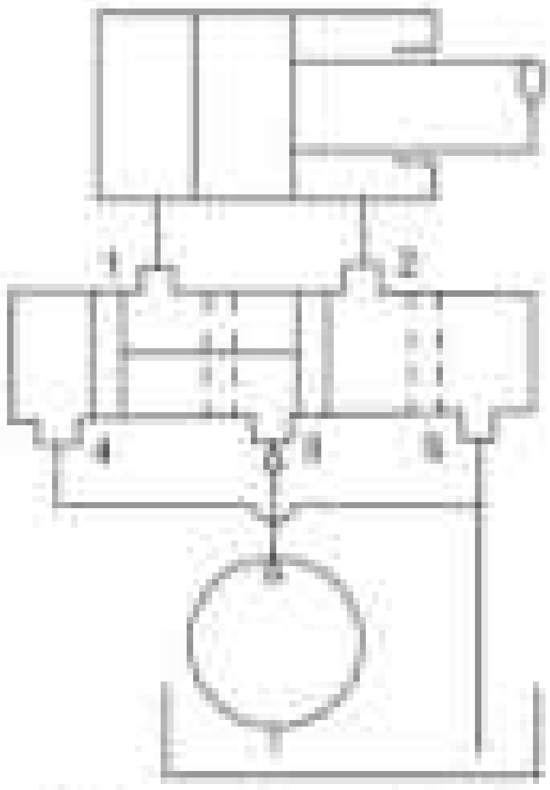


Fig. 1.28 *Schematic diagram of a four-way, two-position, piston-type direction-control valve*

A four-way, three-position, piston-type direction-control valve is schematically shown in Fig. 1.29. This valve also has five ports which are connected in the same manner as the ports of the four-way, two-position valve. When the valve piston is in the central position, all the ports are connected to each other and the oil which is pumped into the valve returns to the reservoir without affecting any change in the position of the hydraulic cylinder. When the valve piston occupies the extreme left position, oil is fed into the right-hand chamber of the cylinder through port 2, as the draining port 5 is closed. Oil in the left-hand chamber is drained back to the reservoir through ports 1 and 4. When the valve piston is shifted to the extreme right position, draining port 4 gets closed and oil is delivered to the left-hand chamber of the cylinder through port 1; in this position oil from the right-hand chamber is drained through ports 2 and 5. In machine-tool hydraulic systems, the sliding piston direction-control valves are used more extensively than rotary spool valves. Two-position valves are used in machine tools in which machining operation is done in several passes, *e.g.*, grinding machines. The three-position valves are used in single-pass machine tools, such as drilling and milling machines. Multiple-position valves find application in automatic drilling, milling and other machines, in which machining of the workpiece is completed in one pass.

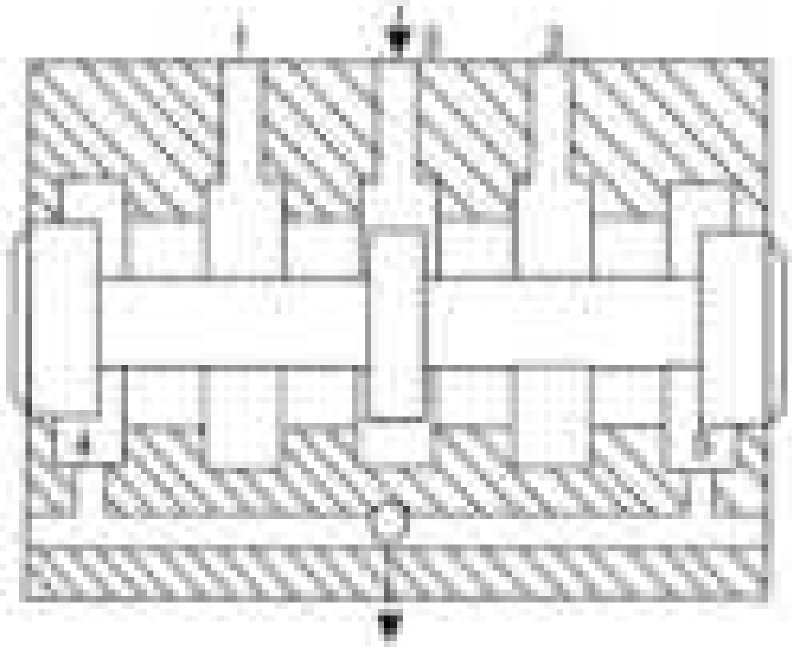


Fig. 1.29 *Schematic diagram of a four-way, three-position, piston-type direction-control valve*

4. Pressure Valves The function of pressure valves is to limit the pressure in a particular line of the hydraulic circuit. Pressure valves are used as safety valves (as in Fig. 1.20) to protect the system against excessive pressure and as bypass valves (as in Fig. 1.21) to drain off the excessive amount of oil. The basic design of safety and bypass pressure valves is identical; however, design details differ on account of different functional requirements of the two. Safety valves are not operated frequently, and therefore, they are designed to be oil-tight when closed. On the other hand, bypass valves operate almost continuously, and therefore, the design requirement for these valves is not oil tightness of joints but higher wear resistance of seals and packings.

The simplest type of pressure valve is the ball or poppet valve which is shown in Fig. 1.30. The ball (or poppet) is pressed against the opening by a spring, whose force can be regulated by means of a threaded sleeve. When the pressure of oil coming through port 1 exceeds the spring pressure, the ball is raised and the oil is drained back into the reservoir through ports 2 and 3. The ball or poppet valve is generally used only as a safety valve. Its application as a bypass valve is not recommended as it suffers from serious drawbacks, such as pressure pulsations and vibrations.

A spool-type pressure valve which has better performance characteristics is shown in Fig. 1.31. Ports 1 and 2 of the valve are connected to the pressure line, the former directly and the latter through a constricted passage. Port 3 is connected to the reservoir. In the condition of equilibrium,

$$P + F = P_s + W$$

where P = force acting at the head end of the valve

F = friction force

P_s = spring force

W = weight of the spool

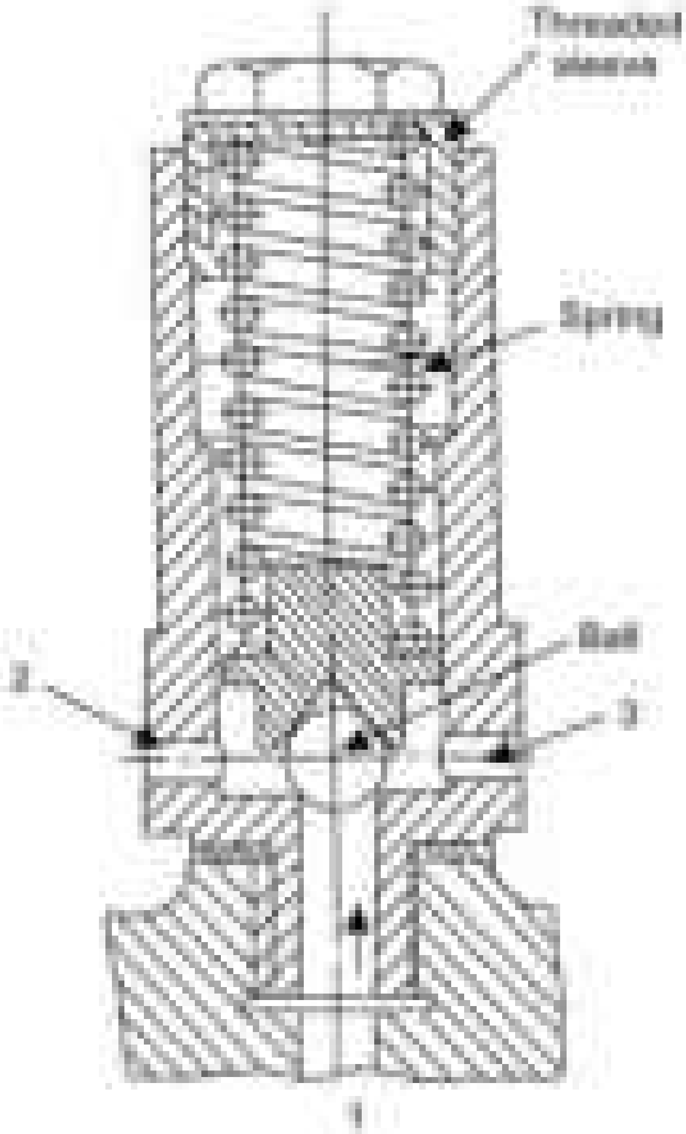


Fig. 1.30 *Schematic diagram of a ball-type pressure valve*

When due to increase in the pressure, force $P \times F$ exceeds $P_s + W$, the spool gets displaced upwards and port 1 gets directly connected to port 3, thus allowing excessive oil to be drained back to the reservoir and resulting in a fall of pressure.

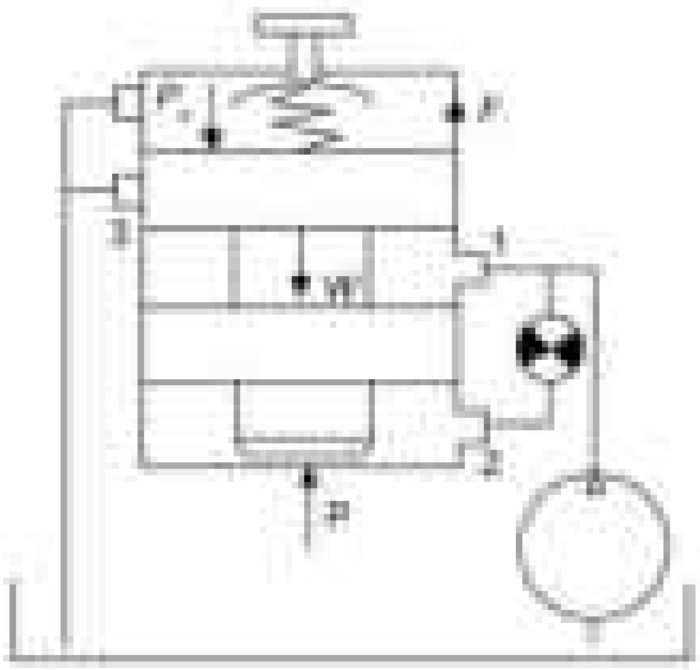


Fig. 1.31 *Schematic diagram of a spool-type pressure valve*

A still better design of pressure valves is shown in Fig. 1.32. This valve is known as a *piston-type pressure valve* or a *compound relief valve*. The pressure line is connected directly to the pilot end and the lower face of piston by ports 1 and 2, respectively; it is also connected to the piston end by port 3 through a constricted passage. Port 5 is connected to the reservoir. In the condition of equilibrium,

$$P_0 + P_1 + F = P_2 + P_{s1} + W$$

where P_0 = force acting at the pilot-valve end
P_1 = force acting on the lower face of the piston
F = friction force
P_2 = force acting on the piston end
P_{s1} = force of spring 1
W = weight of the piston and pilot

When the pressure in the line increases, the equilibrium gets disturbed and a resultant force begins to act in the upward direction. As long as this resultant force P_r, is less than spring force P_{s2} of the ball valve, the piston remains stationary. However, when $P_r > P_{s2}$, the ball valve opens, pressure at the piston end drops, the piston along with the pilot moves upwards and gets directly connected to the draining port 5. Excess of oil is drained back to the reservoir and the line pressure drops.

Spool and piston-type pressure valves are used mostly as bypass valves. The piston-type pressure valve has the ability to absorb minor pressure variations and is, therefore, the best from the point of view of pressure pulsations and dynamic behaviour.

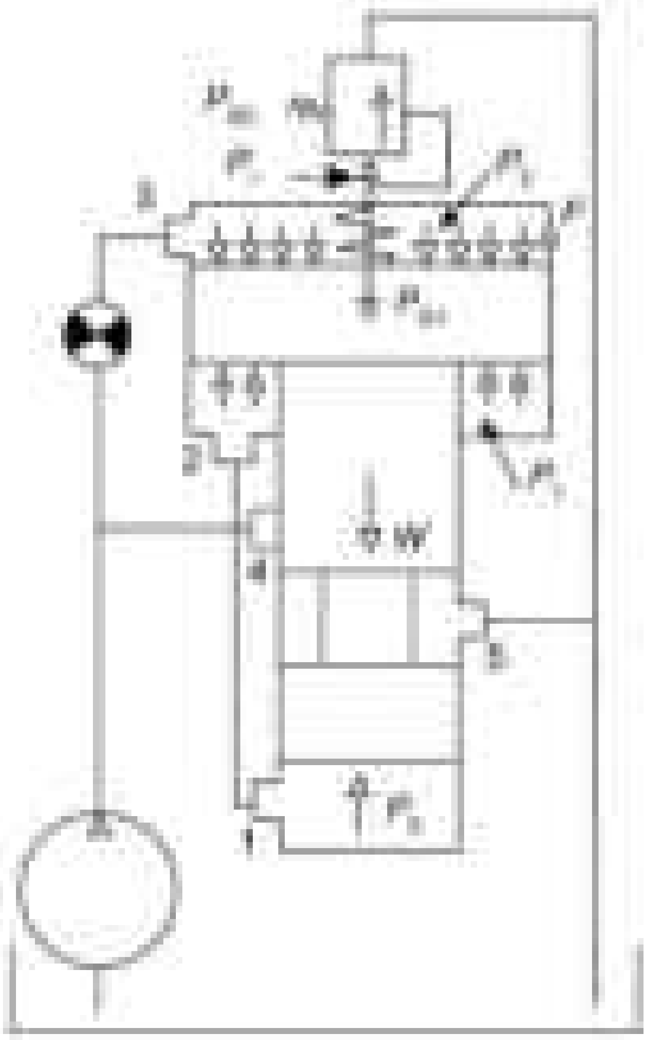


Fig. 1.32 *Schematic diagram of a compound-relief valve*

5. Throttles Flow control valves with a fixed orifice are used in machine tools to minimise vibrations and smooth out transient flow (e.g., constricted passages used in the hydraulic circuits of Figs 1.31 and 1.32). Flow-control valves or throttles which have provision for changing the area of the constricted passage are used to regulate the oil flow in machine-tool hydraulic systems (e.g., the hydraulic circuit of Fig 1.21). The schematic diagrams of a few of the simplest throttle valves are given in Fig. 1.33.

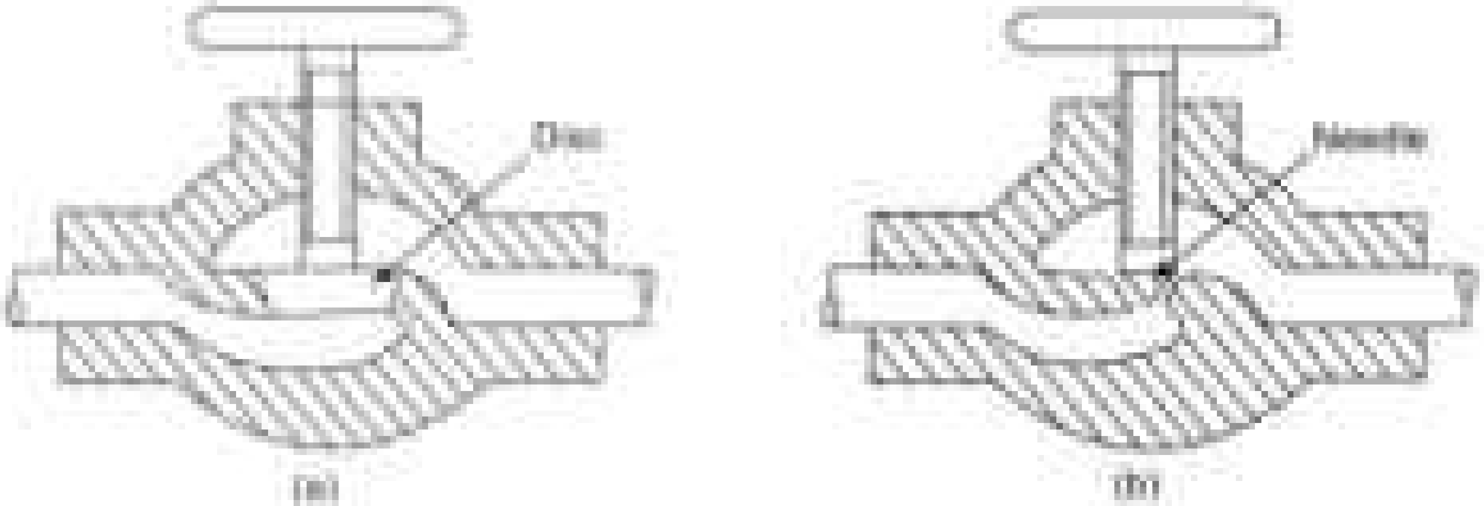


Fig. 1.33 *Throttle valves: (a) Globe valve (b) Needle valve*

In all these valves, the area of the constricted passage is varied by displacing a movable member, for instance, the moving member in the globe valve (Fig. 1.33a) is a disc, and in the needle valve (Fig. 1.33b) a needle. In simple valves of this type, changes in oil temperature and pressure go uncompensated. Therefore, if the pressure or temperature of the oil changes, the flow through the valve can change even at a fixed setting of the constricted passage. The chief aim of compensating for variations of oil pressure and temperature is to provide uniform travel of the machine-tool operative element. This aspect has been dealt with in Sec. 2.9.1 in which stabilisation of the motion velocity with the help of reducing valves has been discussed.

1.5 MECHANICAL TRANSMISSION AND ITS ELEMENTS

Mechanical transmission is employed for transmitting rotary as well as translatory motion to the operative element. This transmission can provide both stepped and stepless regulation of speed and feed rates. Stepless regulation is achieved through special devices called variators, which will be discussed in Sec. 2.9.3. A mechanical transmission that provides for stepped regulation of speed and feed rates is made up of elementary drives and mechanisms. For ease in presentation, the elements of mechanical transmission can be classified into the following groups:

1. Elementary transmissions that transfer rotation
2. Elementary transmissions that transform rotary motion into translatory motion
3. Devices for intermittent motion
4. Reversing and differential mechanisms
5. Special mechanisms and devices
6. Couplings and clutches

1.5.1 Elementary Transmissions for Transmitting Rotary Motion

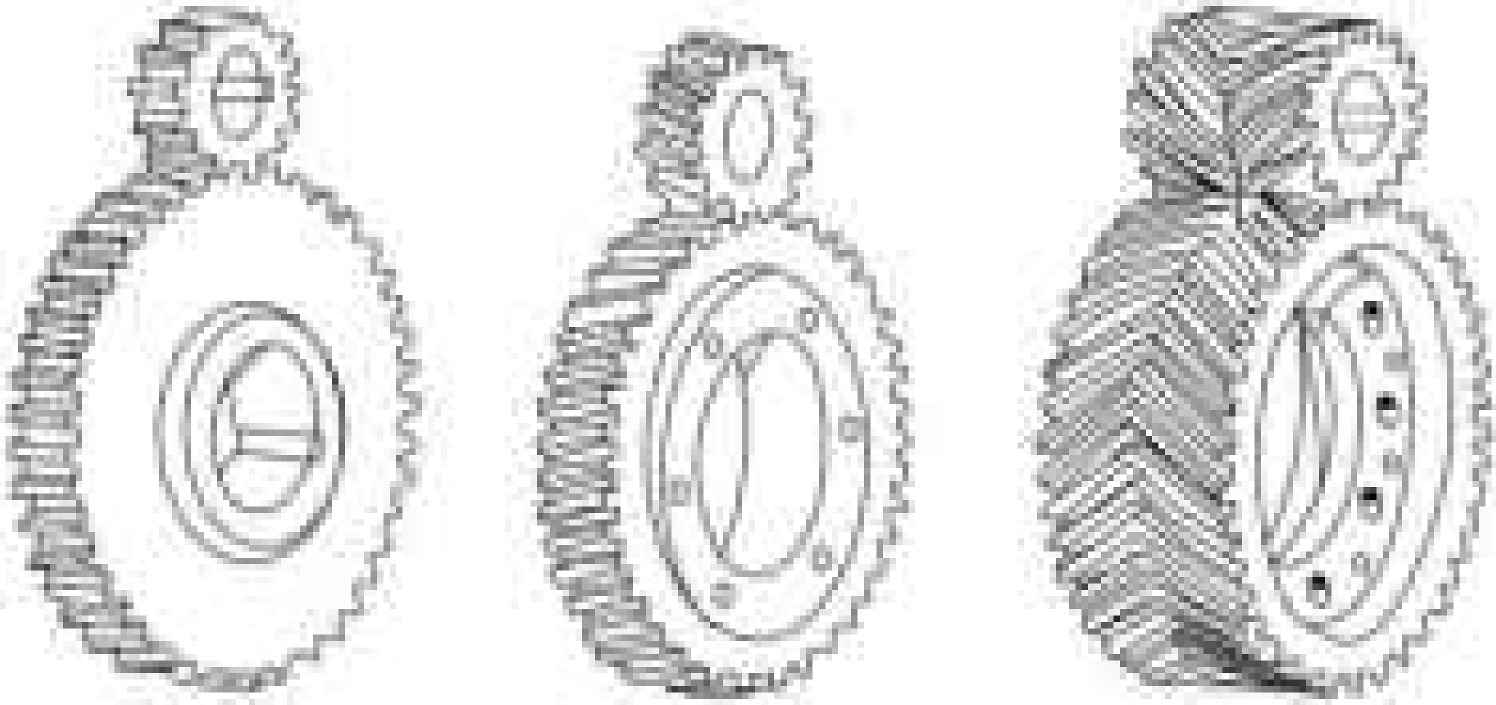
The important elementary transmissions which are used for transmitting rotary motion from one shaft to another are described below.

Gear Transmission In a gear transmission, the rpm of the driven shaft is determined as

$$n_2 = n_1 \cdot \frac{Z_1}{Z_2}$$

where n_2 = rpm of the driven shaft
n_1 = rpm of the driving shaft
Z_1 = number of teeth of the driving gear
Z_2 = number of teeth of the driven gear

The ratio Z_1/Z_2 is known as the transmission ratio of the gear drive and is constant for a particular gear pair.



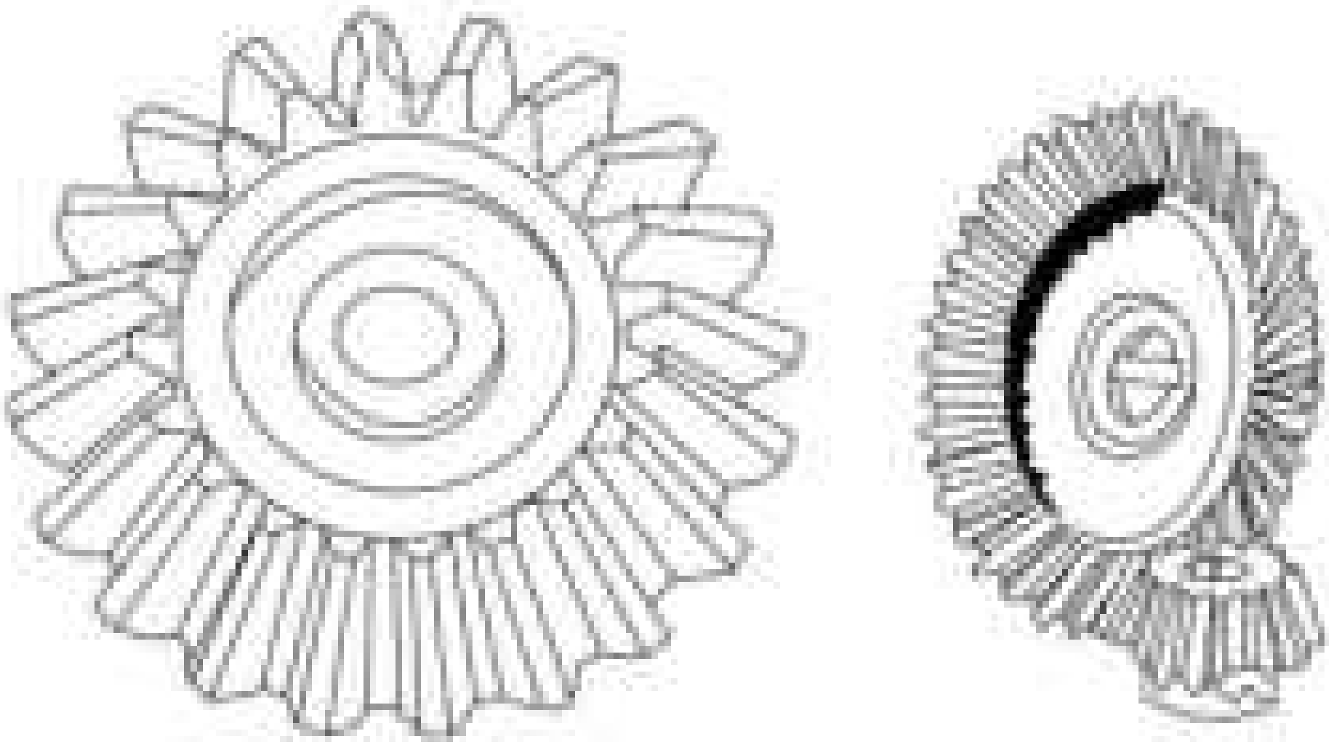
(a) (b) (c)

Fig. 1.34 *Gears: (a) Spur (b) Helical (c) Herringbone*

Rotation is transmitted between parallel shafts by means of spur, helical and herringbone gears (Fig. 1.34). Spur gears have teeth parallel to the axis of rotation, while in helical gears the teeth are inclined with respect to the axis of rotation at an angle known as the helix angle. The herringbone gear is essentially a pair of helical gears in which the helix angle is oppositely directed. Spur gears are used in sliding gear blocks, while helical gears are preferred when the gear pairs are permanently in meshing.

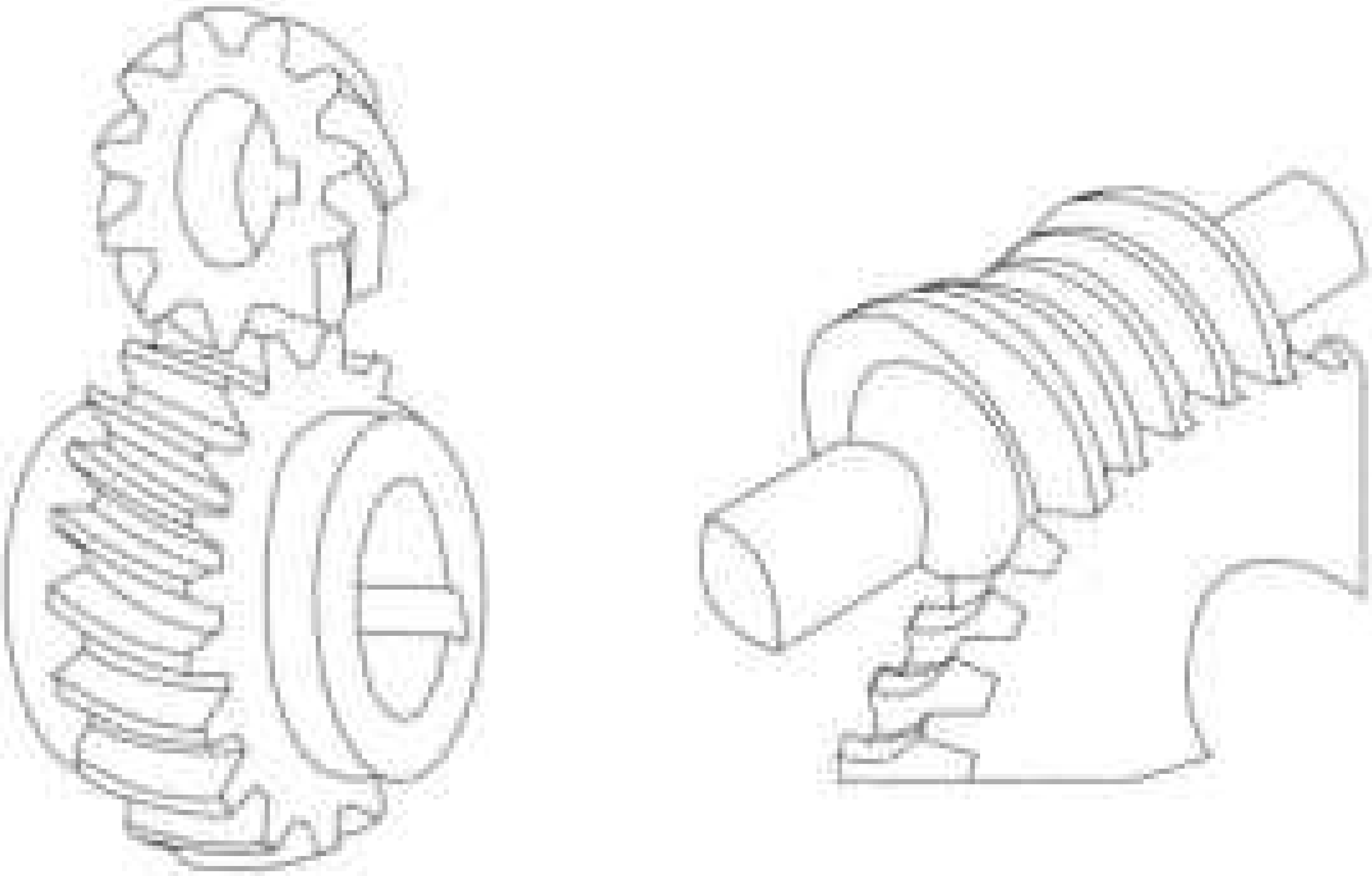
Transmission of rotation between inclined intersecting axes is done with the help of bevel gears. A bevel gear is shown in Fig. 1.35a. The angle between the inclined axes is generally 90° and the bevel-gear transmission (Fig. 1.35b) is commonly employed for transmitting rotation between perpendicular shafts.

Transfer of rotation between skewed axes, i.e., axes that are inclined to each other but do not intersect, is achieved by means of a spiral gear transmission (Fig. 1.36a) or a worm-worm gear transmission (Fig. 1.36b). The spiral gear transmission is characterised by point contact between the meshing gears, and therefore, it cannot be employed for transmitting large torques. In machine tools, the worm-worm gear transmission is commonly employed to achieve heavy speed reduction. Also, since the contact between the worm and worm gear is along a line, this pair can transmit large torques. It should be noted that the worm-worm gear transmission is irreversible and rotation may be transmitted from the worm to the worm gear, but not vice versa. The worm is, in principle, a helical screw and the rpm of the worm gear can be determined by the relationship,



(a) (b)

Fig. 1.35 *(a) Bevel gear (b) Bevel gear pair*



(a) (b)

Fig. 1.36 *(a) Spiral gear pair (b) Worm-worm gear pair*

$$n_2 = n_1 \cdot \frac{k}{Z}$$

where n_2 = rpm of the worm gear
n_1 = rpm of the worm
Z = number of teeth of the worm gear
k = number of passes of the worm

For a single pass worm, $k = 1$, for a double pass worm, $k = 2$.

If a transmission chain consists of a number of elementary gear transmissions connected in series, the overall transmission ratio of the chain is obtained as the product of transmission ratios of the elementary transmissions. In general, the transmission ratio of a gear drive may be >1 (speed increase) or <1 (speed reduction), except the worm-worm gear transmission which always has a transmission ratio <1.

Belt Transmission The belt transmission is used for transmitting rotation between shafts that are located at a considerable distance from each other. It is distinguished by smooth and jerk-free rotation which enables its application in high-speed machine tools, e.g., grinding machines. Belt transmission can be employed for transmitting rotation between parallel and skewed shafts. The most commonly used arrangements are shown in Fig. 1.37.

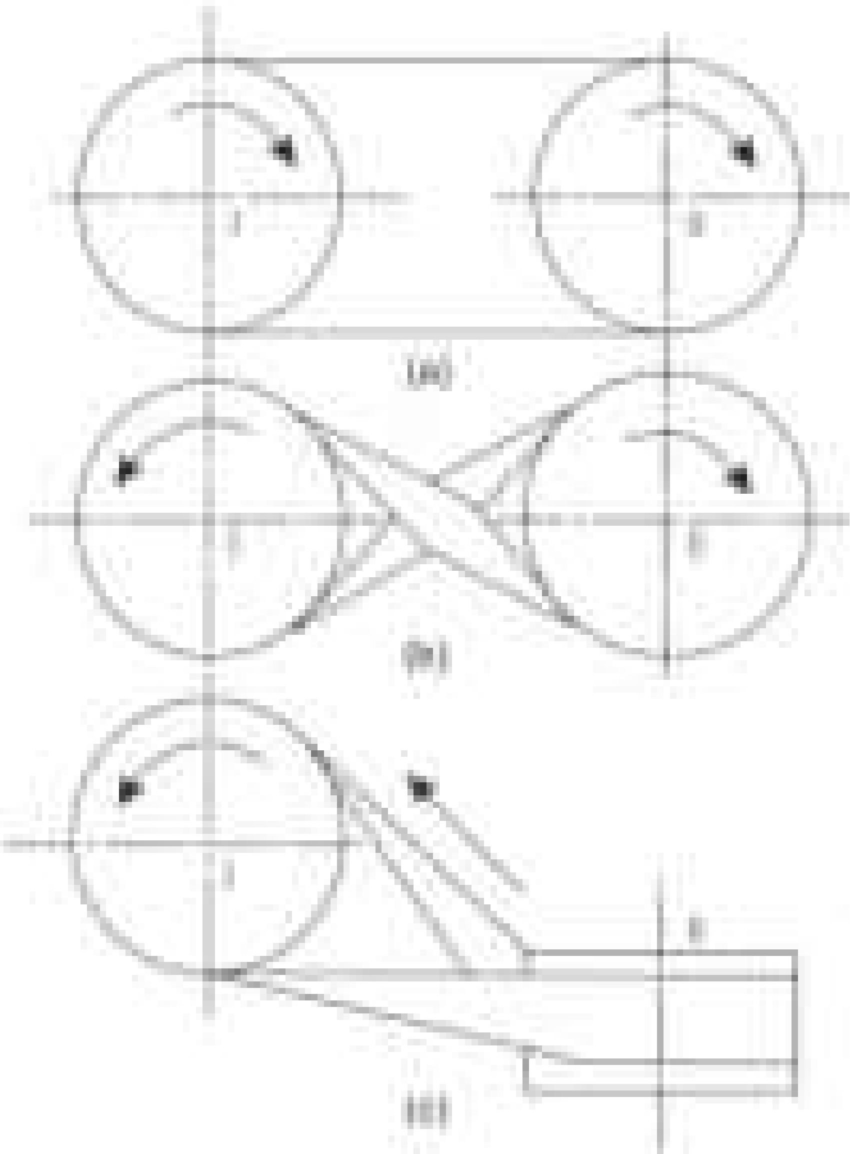


Fig. 1.37 *Belt drives: (a) Open-belt arrangement (b) Cross-belt arrangement (c) Quarter-turned arrangement*

The open-belt arrangement (Fig. 1.37a) is employed for transmitting motion between parallel shafts rotating in the same direction. The cross-belt arrangement (Fig. 1.37b) is used when rotation is transmitted between parallel shafts rotating in opposite directions and the quarter-turned arrangement (Fig. 1.37c) is used for transmitting rotation between skewed shafts.

In machine-tool drives flat, V-shaped and round belts are used. Round belts find application in table model machine tools in which torques are of small magnitude. Flat belts are the most versatile as they can be employed in all the three arrangements shown in Fig. 1.37. The load-carrying capacity of the flat belt can be improved by increasing its width, and therefore, in flat belt drives only one belt is used. In V-belt transmission a number of V-belts (generally two to four) are used for varying the load-carrying capacity in order to avoid large bending stresses in one V-belt, which would otherwise be of unduly large dimensions. V-belts are usually employed only in the open-belt arrangement.

For proper functioning of the belt drive, it is essential to provide some mechanism which keeps the belt tight during operation; this increases their cost. Other major drawbacks of the belt transmission are its relatively large dimensions and inability to guarantee constant transmission ratio due to unavoidable slip between the belt and pulleys.

The rpm of the driven shaft in the belt drive may be determined by the relationship,

$$n_2 = n_1 \frac{D_1(1-\xi)}{D_2}$$

where n_2 = rpm of the driven shaft
n_1 = rpm of the driving shaft
D_1 = diameter of the driving pulley
D_2 = diameter of the driven pulley
ξ = relative slip between belt and pulley

The value of ξ varies between 0.01–0.02 depending upon the belt material.

The belt transmission can be employed to provide transmission ratios >1 as well as <1.

Chain Transmission The chain transmission (Fig. 1.38) is employed for transmitting rotation only between parallel shafts that are located at a considerable distance. The chain transmission consists of a driving sprocket, driven sprocket and chain. Chain transmission is used in machine tools when it is essential to keep the dimension of the drive within reasonable limits and also ensure transmission without slip. The rpm of the driven shaft is determined as,

$$n_2 = n_1 \frac{Z_1}{Z_2}$$

where n_1 = rpm of the driving shaft
n_2 = rpm of the driven shaft
Z_1 = number of teeth on the driving sprocket
Z_2 = number of teeth on the driven sprocket

The chain transmission is also capable of providing transmission ratios <1 and >1.

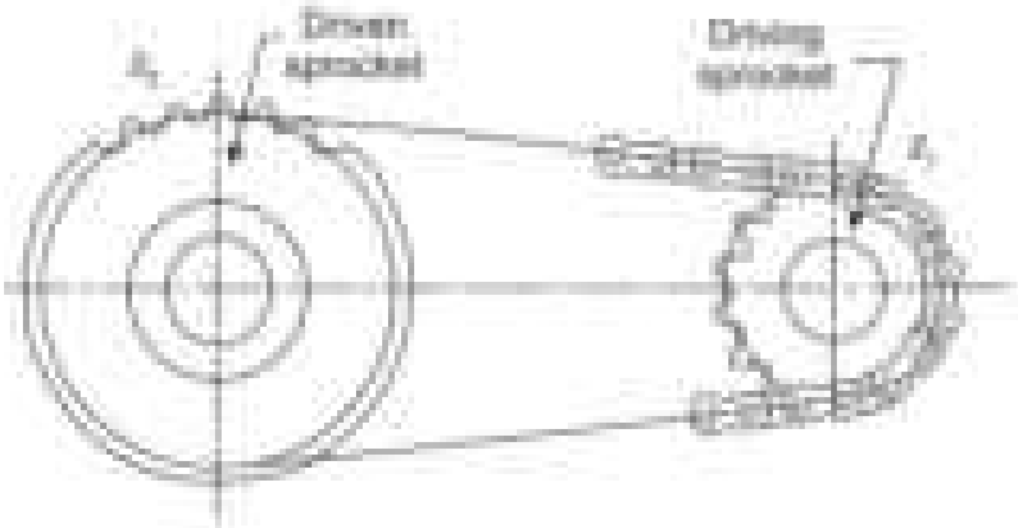


Fig. 1.38 Chain transmission

1.5.2 Elementary Transmission for Transforming Rotary Motion into Translatory

These elementary transmissions are employed in feed mechanisms of most of the machine tools and also in the drives of machine tools having a reciprocating primary cutting motion.

The important elementary transmissions that are used in machine tools for transforming rotary motion into translatory are briefly discussed below.

Slider Crank Mechanism The schematic diagram of a slider crank mechanism is shown in Fig. 1.39. The mechanism consists of a crank, connecting rod and slider. The forward and reverse strokes each take place during half a revolution of the crank. Therefore, the speeds of forward and reverse speeds in the slider crank mechanism are identical. Since metal removal occurs during one stroke (generally the forward stroke), it is desirable from the point of view of productivity to have a higher speed of the other stroke (the reverse stroke). Due to this property, the slider crank mechanism is used only in machine tools with small strokes (< 300 mm), where an increase of the reverse-stroke speed does not result in an appreciable increase of productivity, e.g., in the drive of the primary cutting motion of gear shaping machines. The length of stroke may be changed by adjusting the crank radius and is equal to $L = 2R$, where R is the crank radius.

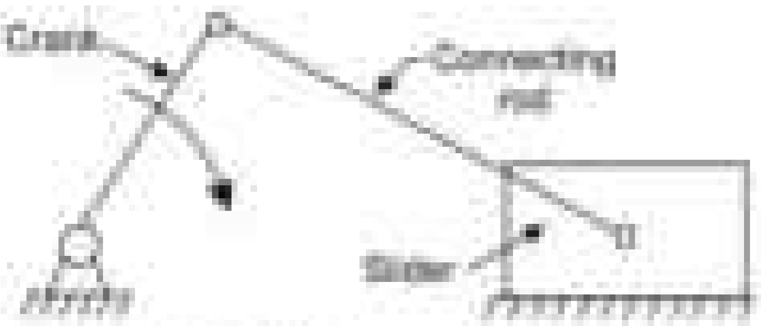


Fig. 1.39 *Slider crank mechanism*

Crank-and-Rocker Mechanism The crank-and-rocker mechanism (Fig. 1.40) consists of a rotating crank which makes the rocker arm oscillate by means of a block sliding along the groove in the rocker arm. The forward cutting stroke takes place during the clockwise rotation of the crank through angle α, and the reverse (idle) stroke during rotation of the crank through angle β. Since $\alpha > \beta$ and the crank rotates with uniform speed, the idle stroke

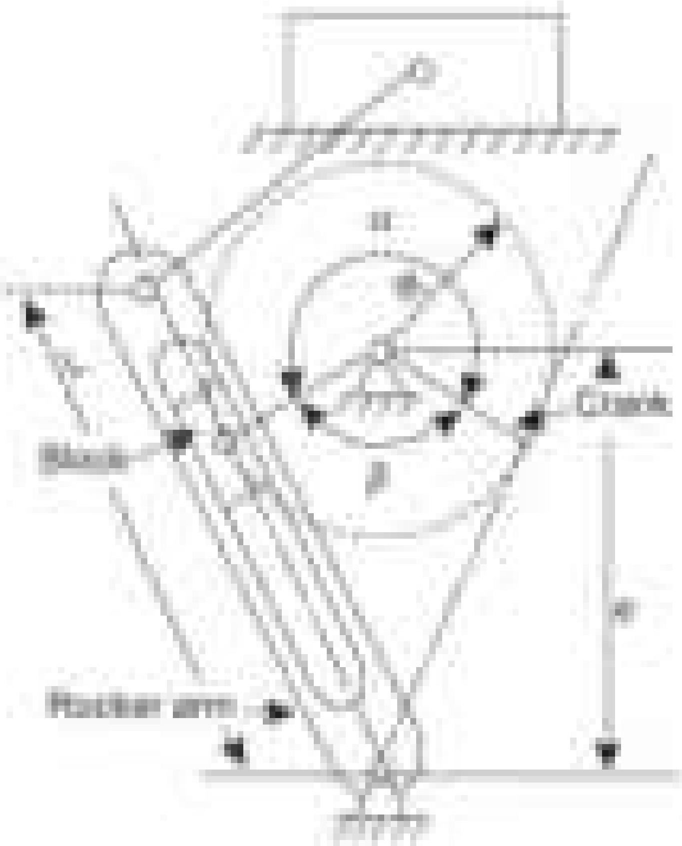


Fig. 1.40 *Crank-and-rocker mechanism*

is completed faster than the cutting stroke. The length of stroke can be varied by adjusting the crank radius. With a decrease in the crank radius, the ratio of angles α/β decreases and the speeds of cutting and reverse strokes tend to become equal. The crank-and-rocker mechanism is, therefore, preferred in machine tools with large strokes (up to 1000 mm) where it can be effectively employed, e.g., in the drive of the primary cutting motion of shaping and slotting machines. The length of stroke can be calculated from the expression,

$$L = 2\left(\frac{L}{e}\right)R \text{ mm}$$

where L = length of the rocker arm, mm

e = off-set distance between the centres of rotation of the rocker arm and crank, mm

R = radius of the crank, mm

Cam Mechanism The cam mechanism (Fig. 1.41) consists of a cam and a follower. The cam mechanism can provide the desired translatory motion if a suitable profile is selected. The profile may be provided

1. on the periphery of a disc—disc type cam mechanism (Fig. 1.41a),
2. on the face of a disc—face type cam mechanism (Fig. 1.41b), and
3. on a cylindrical surface—drum type cam mechanism (Fig. 1.41c).

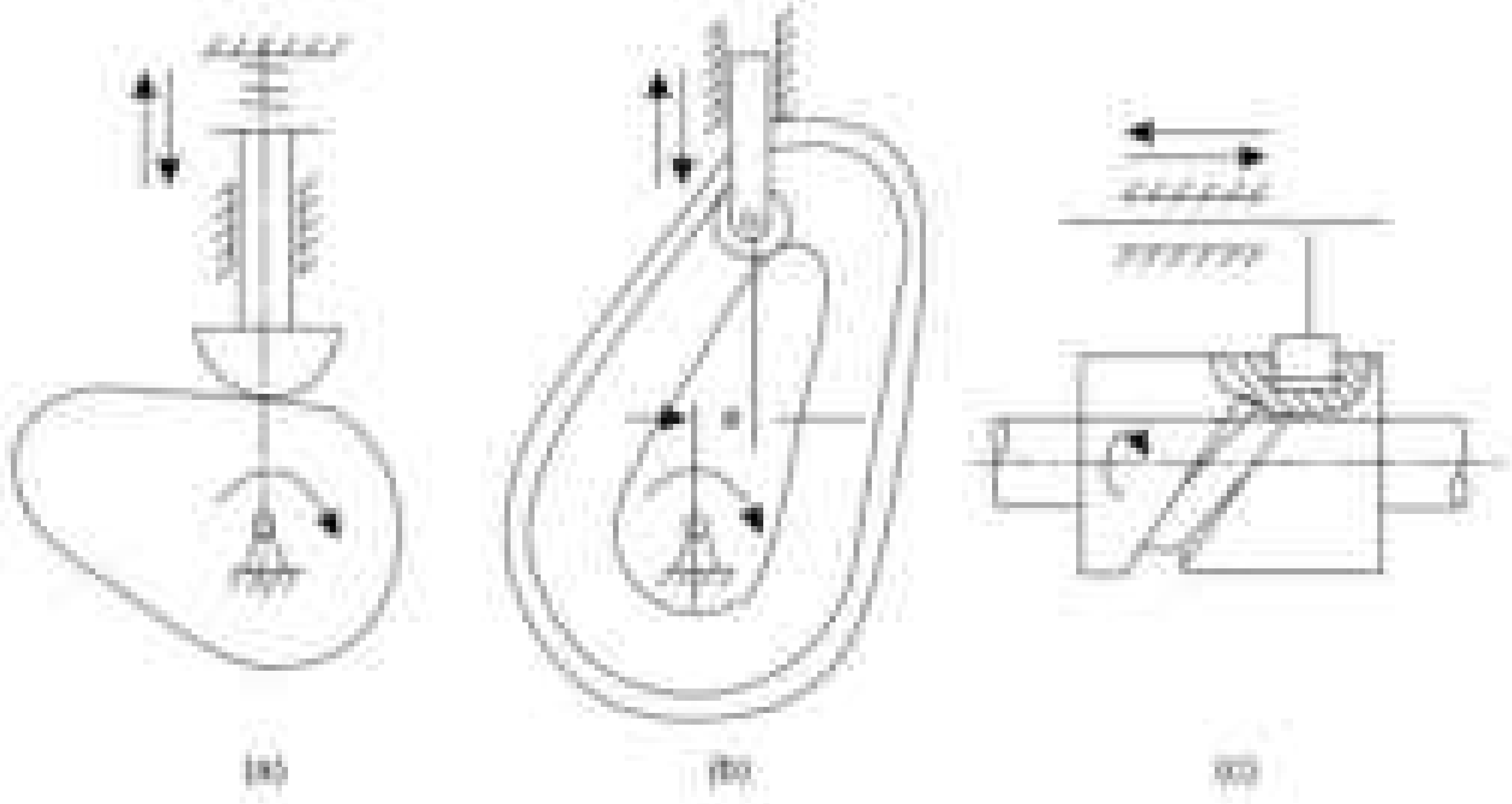


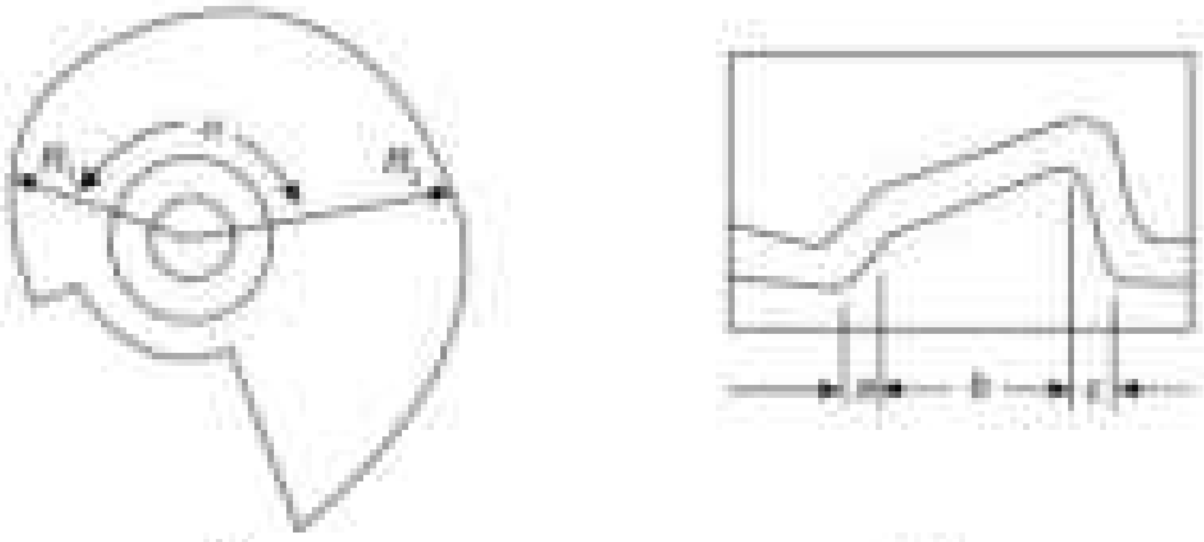
Fig. 1.41 *Cam mechanism: (a) Disc type (b) Face type (c) Drum type*

The main advantage of cam mechanisms is that the velocity of the operative element is independent of the design of the driving mechanism and is controlled by the cam profile. For example, in a disc-type cam, if the radius changes from R_1 to R_2 (Fig. 1.42a) along an Archimedes' spiral while the cam rotates through angle α, the velocity of the follower can be determined from the expression:

$$v = \frac{R_2 - R_1}{\alpha} \cdot 360 \cdot \frac{n}{1000} \text{ m/min}$$

where n = rpm of the cam

R_1, R_2 = radii, mm



(a) (b)

Fig. 1.42 *(a) Profile of a disc-type cam (b) Development of the profile of a drum-type cam*

Similarly, in face- or drum-type cam mechanisms, the speed of the follower depends upon the steepness of the groove. Consider, for instance, the profile development of the drum cam shown in Fig. 1.42b. Segment a depicts the steep rise of the follower corresponding to the rapid advance, segment b depicts the slow rise corresponding to the working stroke and segment c the steep fall corresponding to the rapid withdrawal of the cutting tool. The speed during, say, the working stroke, can be determined by the following relationship:

$$v = \frac{h}{b} \cdot \frac{\pi D}{1000} \cdot n \text{ m/min}$$

where: h = rise during the working stroke, mm

b = length of the working stroke, mm

D = diameter of the drum, mm

n = rpm of the drum

It should be kept in mind that cam mechanisms are costly and a new set is required whenever any change in working conditions is sought to be incorporated. Cam mechanisms are, therefore, generally used in automatic machine tools for mass production of components.

Nut-and-Screw Transmission A nut-and-screw mechanism is schematically depicted in Fig. 1.43. The screw and nut have a trapezoidal thread. When the screw, fixed axially, is rotated, the nut moves along the screw axis. The direction of movement can be reversed by reversing the rotation of the screw. The nut-and-screw transmission is compact, but has a high load-carrying capacity. Its other advantages are simplicity, ease of manufacture, and possibility of achieving slow and uniform movement of the operative member. The speed of the operative member can be found from the relationship,

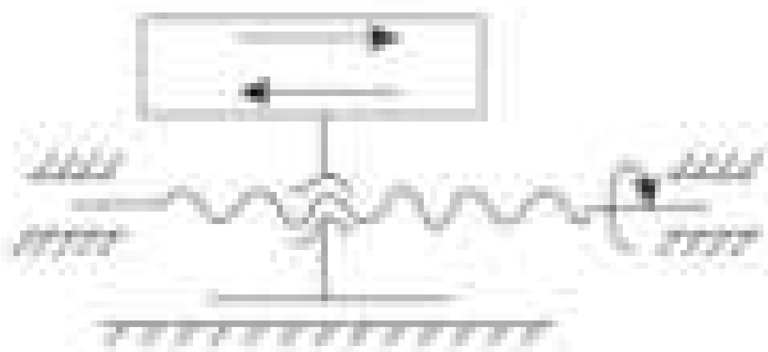


Fig. 1.43 *Schematic diagram of a nut-and-screw transmission*

$$v_{op} = t \cdot K \cdot n \text{ mm/min}$$

where $\quad s_m$ = feed per minute of the operative member

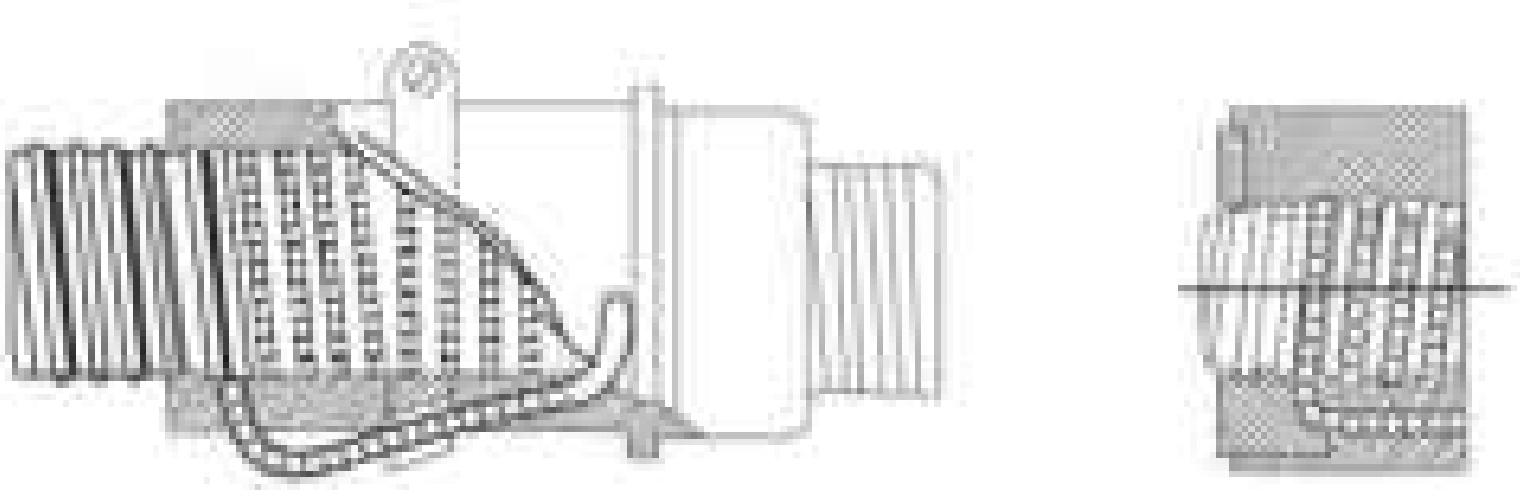
t = pitch of the thread, mm

K = number of starts of the thread

n = rpm of the screw

The major drawback of the nut-and-screw transmission is its low coefficient of efficiency due to large frictional losses. This restricts its application in machine tools to feed and auxiliary motion drives.

Nowadays, rolling friction nut-and-screw transmission is finding increasing application in machine tools. In this transmission, the sliding friction between the nut and screw is replaced by rolling friction by introducing intermediate members, such as balls and rollers. An anti-friction nut-and-screw transmission with balls as rolling members is shown in Fig. 1.44. The balls run along the thread between the screw and the nut and there is provision for their continuous recirculation. For instance, in the transmissions shown on Fig. 1.44, the balls return through an axial channel drilled in the nut (Fig. 1.44b) and through an external return chute (Fig. 1.44a). The thread of the screw and nut in this case is usually half-round and the transmission has provision for backlash elimination by preloading. The efficiency of the anti-friction nut-and-screw transmission reaches 0.9–0.95 as compared to 0.2–0.4 of the sliding-friction transmission. The anti-friction nut-and-screw transmission is mainly used in the feed-motion drive of precision machine tools, such as grinding, jig-boring machines, etc. It is used in numerically controlled machine tools in which backlash is extremely undesirable.



(a) (b)

Fig. 1.44 *Schematic diagram of anti-friction nut-and-screw transmission*

Rack-and-Pinion Transmission A rack-and-pinion transmission is shown in Fig. 1.45. When the rotating gear (pinion) meshes with a stationary rack, the centre of the gear moves in a straight line. On the other hand, if the gear axis is stationary, then the rack executes translatory motion. The direction of motion can be reversed by reversing the rotation of the pinion. The speed of the operative member in this transmission can be found from the relationship,

$$s_m = \pi m \cdot Z \cdot n \text{ mm/min}$$

where $\quad s_m$ = feed per minute of the operative member

m = module of the pinion, mm

Z = number of teeth of the pinion

n = rpm of the pinion

Fig. 1.45 *Rack-and-pinion transmission*

Rack-and-pinion transmission is the simplest and cheapest among all types of transmissions used in reversible drives. It also has high efficiency and provides a large transmission ratio which makes it possible to use it in the feed as well as main drive motions of machine tools. Lack of uniformity in movement due to unavoidable meshing errors between rack-and-pinion teeth preclude its application in precision machine tools. Also, due to absence of self-locking, rack-and-pinion transmission cannot be applied for vertical movement of the operative element.

1.5.3 Devices for Intermittent Motion

In some machine tools, it is required that the relative position between the cutting tool and workpiece should change periodically. This requirement is generally essential in

1. machine tools with a reciprocating primary cutting motion, e.g., shaping machines in which the workpiece must be fed intermittently upon completion of one full stroke of the cutting tool, and
2. machine tools with reciprocating feed motion, e.g., grinding machines, in which the workpiece must be infed intermittently after each half or full stroke of the reciprocating table.

In machine tools, intermittent motion of the operative element is generally obtained with the help of the mechanisms discussed as follows.

Ratchet-Gear Mechanism The ratchet-gear mechanism is schematically shown in Fig. 1.46. It consists of a pawl mounted on an oscillating pin. During each oscillation in the anti-clockwise direction, the pawl turns the ratchet wheel through a particular angle. During the clockwise oscillation in the opposite direction, the pawl simply slides over the ratchet teeth and the latter remains stationary. The ratchet wheel is linked to the machine-tool table through a nut-and-screw transmission. Therefore, the periodic rotation of the ratchet wheel is transformed into the intermittent translatory motion of the table. For a particular nut-and-screw pair of some constant transmission ratio, the feed of the table during each oscillation depends upon the swing of the oscillating pawl. Generally, the rotation of the ratchet wheel in one stroke of the pawl should not exceed 45°. The ratchet-gear mechanism is most suitable in cases when the periodic displacement must be completed in a short time, e.g., in feed mechanisms of shaping, planing and grinding machines in which the intermittent feed motion takes place during the over travel of the cutting tool or during the reverse stroke.

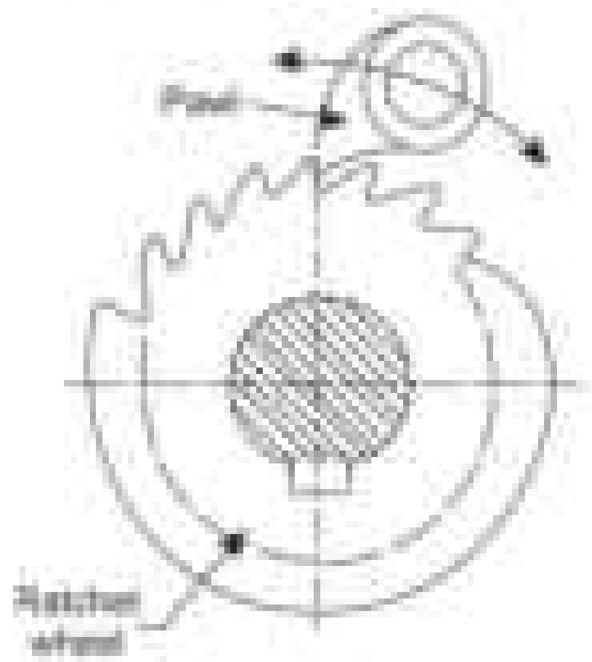


Fig. 1.46 *Pawl-and-ratchet mechanism*

Geneva Mechanism The schematic diagram of the Geneva mechanism is shown in Fig. 1.47. It consists of a driving disc which rotates continuously and a wheel with four radial slots. The arcs on the driving disc and wheel provide a locking effect against rotation of the slotted wheel, e.g., in the position shown in Fig. 1.47a, the wheel cannot rotate. As the disc continues to rotate, point *A* of the disc comes out of contact with the arc and immediately thereafter pin *P* mounted at the end of the driving arm enters the radial slot. The wheel now begins to rotate (Fig. 1.47b); when it has turned through an angle 90°, the pin comes out of the radial slot and immediately thereafter point *B* comes in contact with the next arc of the wheel preventing its further rotation. Thus the wheel makes $1/K$ revolutions, where K is the number of radial slots.

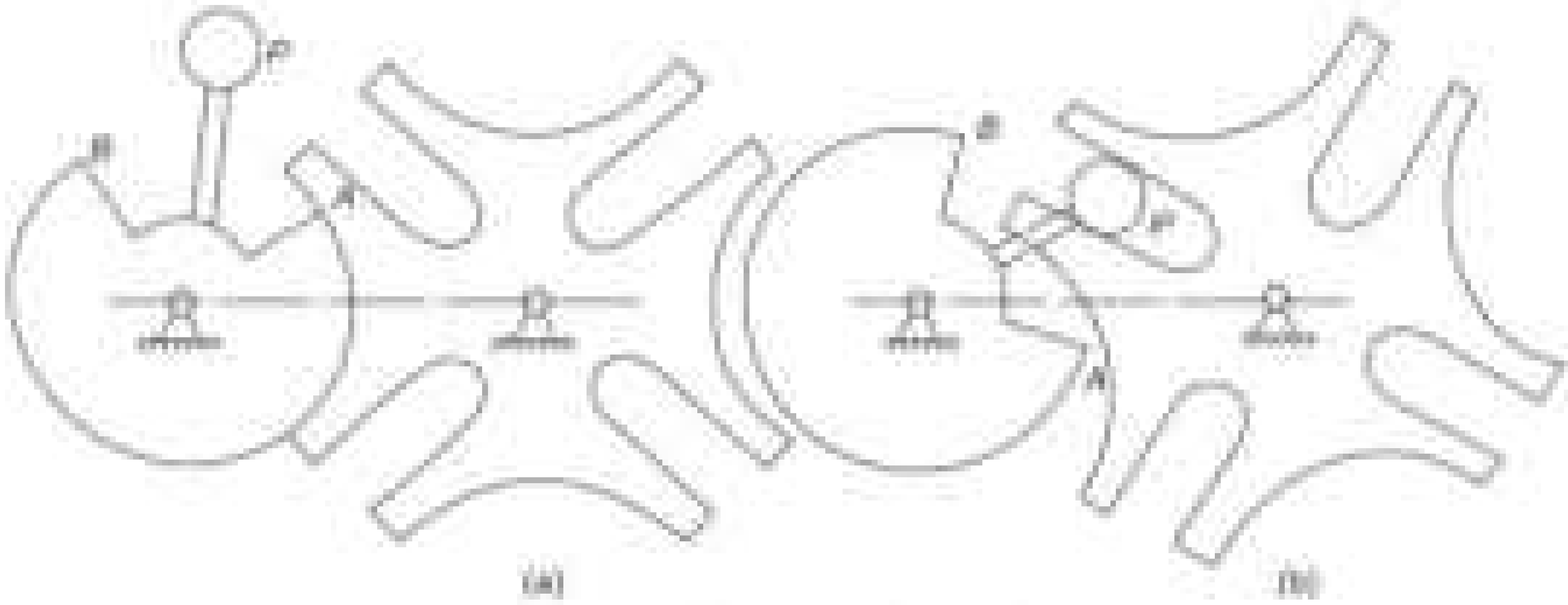


Fig. 1.47 *Geneva mechanism*

In the Geneva mechanism, the angle of rotation of the wheel cannot be varied. Therefore, this mechanism is mainly used in turrets and single-spindle automatic machines for indexing cutting tools and in multiple-spindle automatic machines for indexing spindles through a constant angle.

1.5.4 Reversing and Differential Mechanisms

Reversing Mechanism Reversing mechanisms are used for changing the direction of motion of the operative member. Reversing is accomplished generally through spur and helical gears or bevel gears. A few reversing arrangements using spur and helical gears are shown in Fig. 1.48. In the arrangement of Fig. 1.48a the gears on the driving shaft are mounted rigidly, while the idle gear and the gears on driven shaft III are mounted freely. The jaw clutch is mounted on a key. Rotation may be transmitted to the driven shaft either through gears $(A/B) \cdot (B/C)$ or through D/E depending upon whether the jaw clutch is shifted to the left to mesh with gear *C* or to the right to mesh with gear *E*. In the transmission $(A/B) \cdot (B/C)$ the direction of rotation of the driving and driven shafts will coincide, whereas in the transmission D/E the direction of rotation of the driven shaft will be opposite to that of the driving shaft. In this arrangement, use of helical gears should be preferred.

In the second arrangement shown in Fig. 1.48b, the gears on the driving shaft are again rigidly mounted, and the idle gear is free. On the driven shaft, a double cluster gear is mounted on a spline. By sliding the cluster gear, transmission to the driven shaft may again be achieved either through gears (A/B), (B/C) or through gear pair D/E. Only spur gears may be used in this reversal mechanism.

In the arrangement of Fig. 1.48c gear *A* on the driving shaft and gear *D* on the driven shaft are both rigidly mounted. A quadrant with constantly meshing gears *B* and *C* can be swivelled about the axis of the driven shaft. By swivelling the quadrant with the help of a lever, transmission to the driven shaft may be achieved through $(A/C) \cdot (C/D)$ or through $(A/B) \cdot (B/C) \cdot (C/D)$. In the first case, the direction of rotation of the driving and driven shafts will coincide while in the second it will be opposite. In this mechanism also only spur gears can be used.

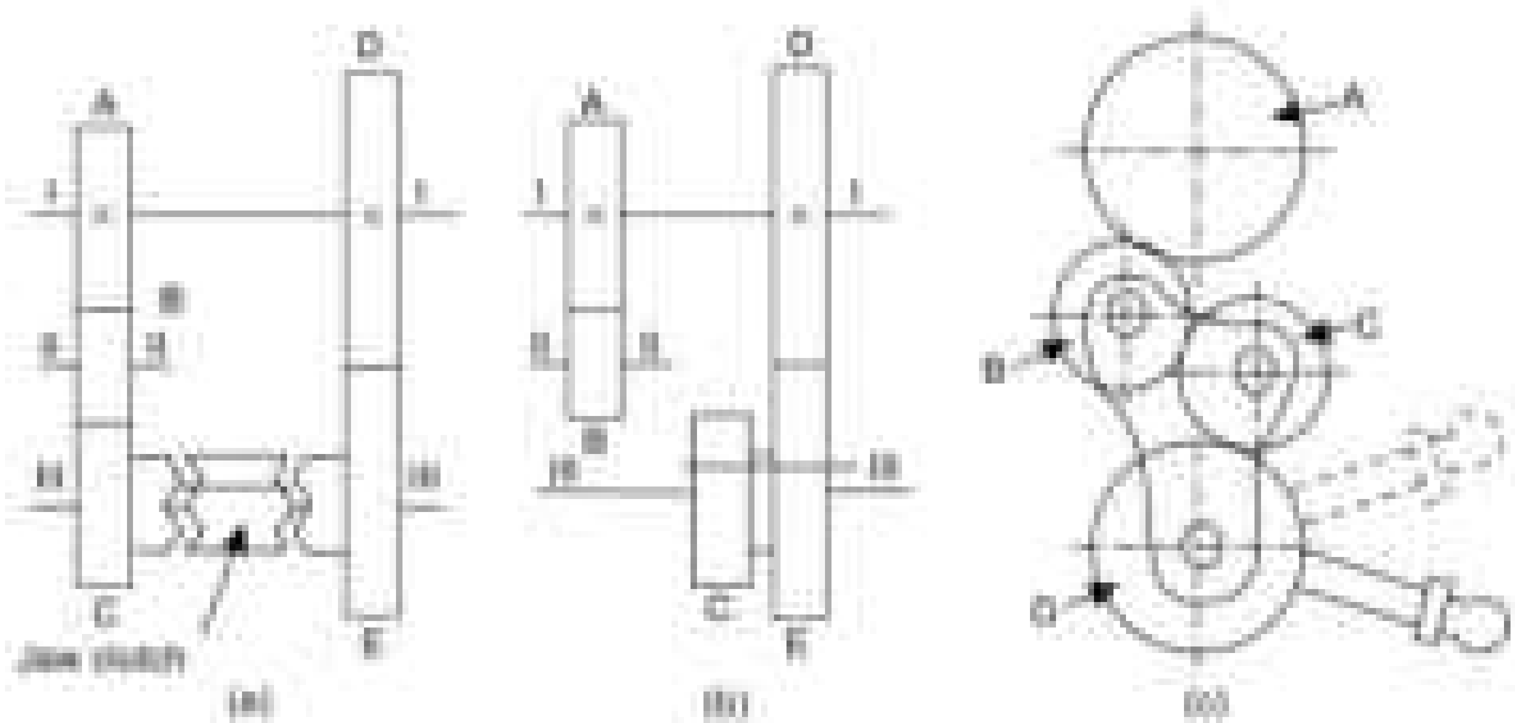


Fig. 1.48 *Reversing mechanisms: (a) using spur gears (b) using helical gears*

It should be noted that in the reversing mechanisms of Fig. 1.48a and b the ratio of direct and reversal speeds will depend upon the transmission ratio of gear pairs *A/C* and *D/E*. By selecting *A/C* = *D/E* we can ensure identical speeds in both directions. However, if desired, a faster reversal speed can be achieved by selecting a larger transmission ratio for the gear pair used in the reversal train (gear pair *A/C*, as the transmission with the idler gear is usually employed for reversal).

Examples of reversal mechanisms using bevel gears are shown in Fig. 1.49. In these devices, shaft I is the driving shaft and shaft II the driven shaft. In the arrangement of Fig. 1.49a, the double-cluster bevel gear is mounted on a splined shaft, and by shifting it the direction of rotation of shaft II can be changed by getting either gear *B* or gear *C* to mesh with bevel gear *A* which is rigidly mounted on the driving shaft.

In the arrangement of Fig. 1.49b, gears *B* and *C* are freely mounted on the driven shaft, while the jaw clutch is mounted on splines. By shifting the clutch to the left or right, rotation to shaft II can be transmitted either through bevel gear pair *A/B* or *A/C* and thus the direction of rotation of the driven shaft can be reversed.

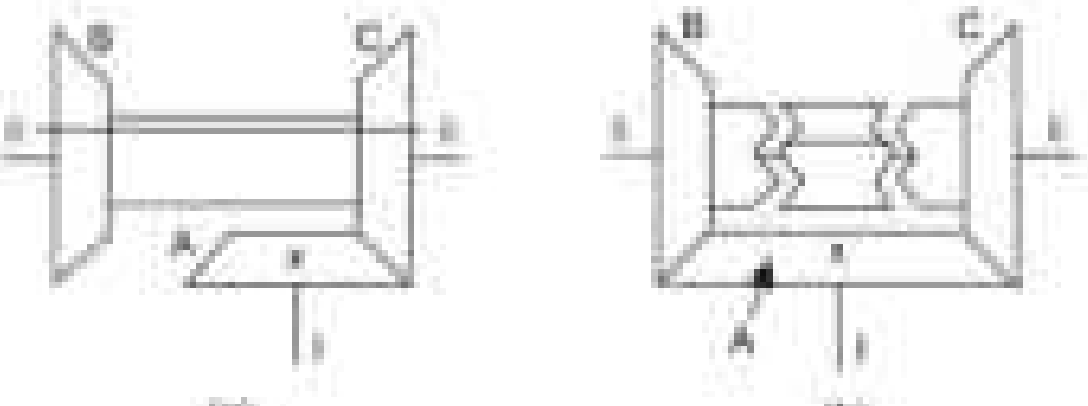


Fig. 1.49 *Reversing mechanisms using bevel gears*

Differential Mechanism Differential mechanisms are used for summing up two motions in machine tools, in which the operative member gets input from two separate kinematic trains. They are generally employed in thread-and-gear cutting machines where the machined surface is obtained as a result of the summation of two or more forming motions.

A simple differential mechanism using spur or helical gears is shown in Fig. 1.50. The mechanism is essentially a planetary gear mechanism consisting of sun gear *A*, planetary gear *B* and arm *C*. The planetary gear is mounted on the arm which can rotate about the axis of gear *A*. Suppose gear *A* makes n_A and arm *C*, n_C revolutions per minute in the clockwise direction. The relative motion between the elements of the mechanism will remain unaffected if the whole mechanism is rotated in the anti-clockwise direction with n_C revolutions per minute. Then the arm becomes stationary and the mechanism is reduced to a simple gear transmission with gear *A* making $n_A - n_C$ revolutions per minute and gear *B* making $n_B - n_C$ revolutions per minute. The transmission ratio of the mechanism may be written as

$$\frac{n_A - n_C}{n_B - n_C} = -\frac{Z_B}{Z_A} \quad \text{(the minus sign denotes the external gear pair)}$$

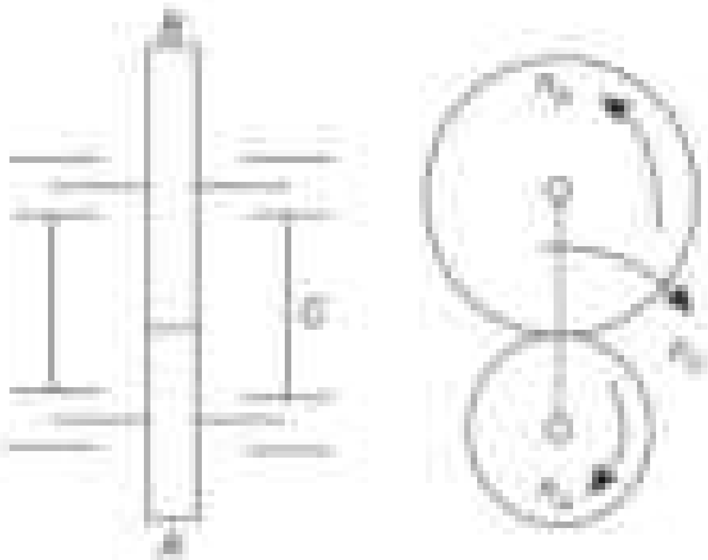


Fig. 1.50 *Differential mechanism using spur or helical gears*

where Z_A and Z_B are the number of teeth of gear *A* and *B*, respectively. The above expression may be rewritten as follows:

$$n_B = n_C\left(1 + \frac{Z_A}{Z_B}\right) - n_A \cdot \frac{Z_A}{Z_B}$$

i.e., the rpm of any one element of the differential mechanism is a function of independent motions of the remaining two elements.

Differential mechanisms using a double-cluster planetary gear are shown in Fig. 1.51. The mechanisms consist of gear *A*, cluster gear block *B*-*B'* mounted on arm *C* and gear *D*. If n_A, n_C and n_D are the rpm's of gear *A*, arm *C* and gear *D*, respectively, then the transmission ratio of the kinematic train between gears *A* and *D* may be expressed as

$$\frac{n_D - n_C}{n_A - n_C} = \frac{Z_A}{Z_B} \cdot \frac{Z_{B'}}{Z_D} \quad \text{(for Fig. 1.51a)}$$

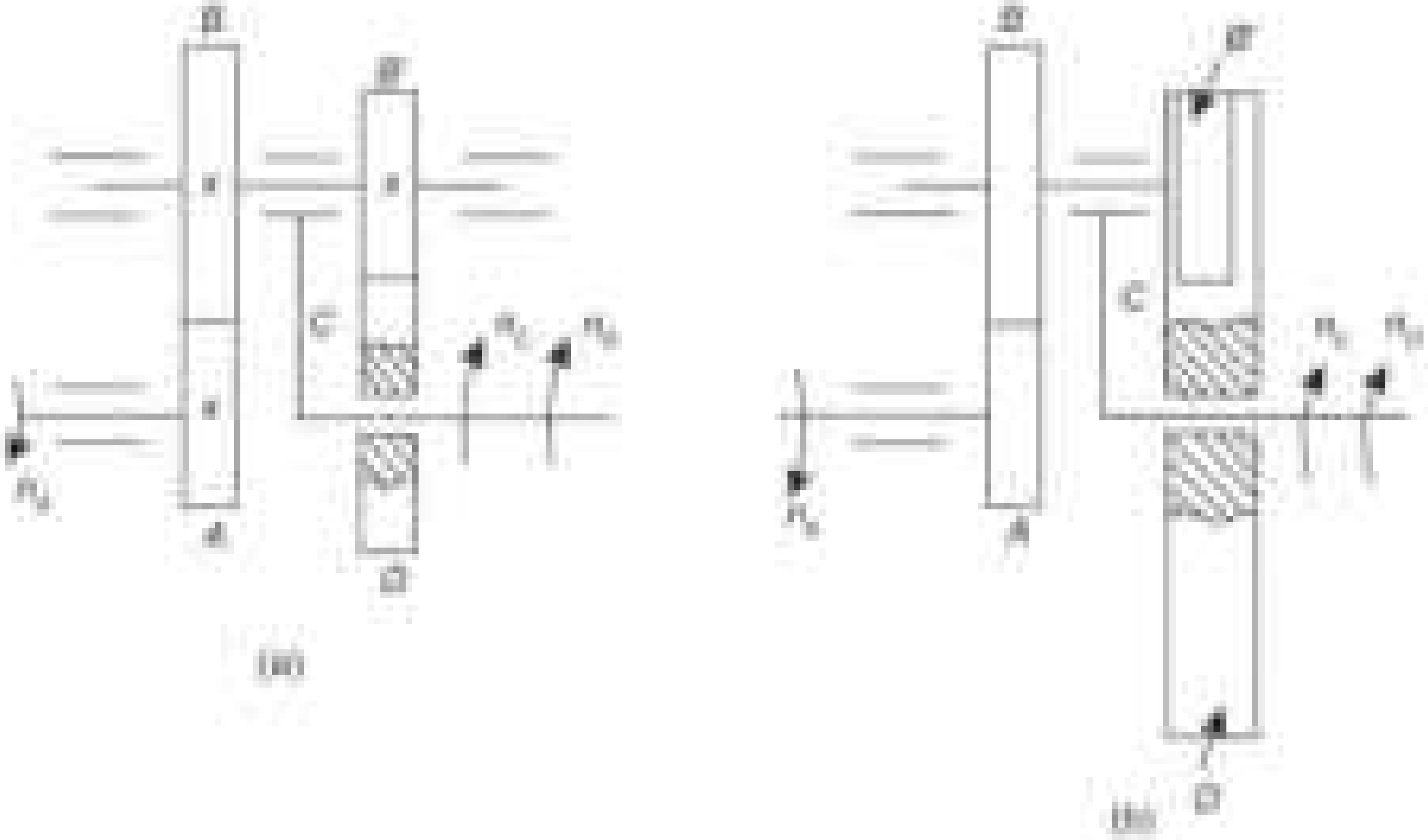


Fig. 1.51 *Differential mechanisms using double-cluster planetary gears*

Differential mechanisms consisting of bevel gears are shown in Fig. 1.52. These mechanisms are widely used in automobiles to provide different rotational speeds to the wheels powered by a single source. This is essential for the functioning of an automobile because, while tackling a turn, the outer wheel of the automobile must rotate faster than the inner wheel. This mechanism is also widely used in machine tools on account of its compactness.

The mechanism consists of bevel gears *A* and *D* and planetary bevel gears *B* and *C*. Planetary gears can be rotated about the common axes of gears *A* and *D*

1. by means of a ring gear (Fig. 1.52a)—this differential is used in automobiles, and
2. by means of a T-shaped shaft (Fig. 1.52b)—this differential is used in machine tools.

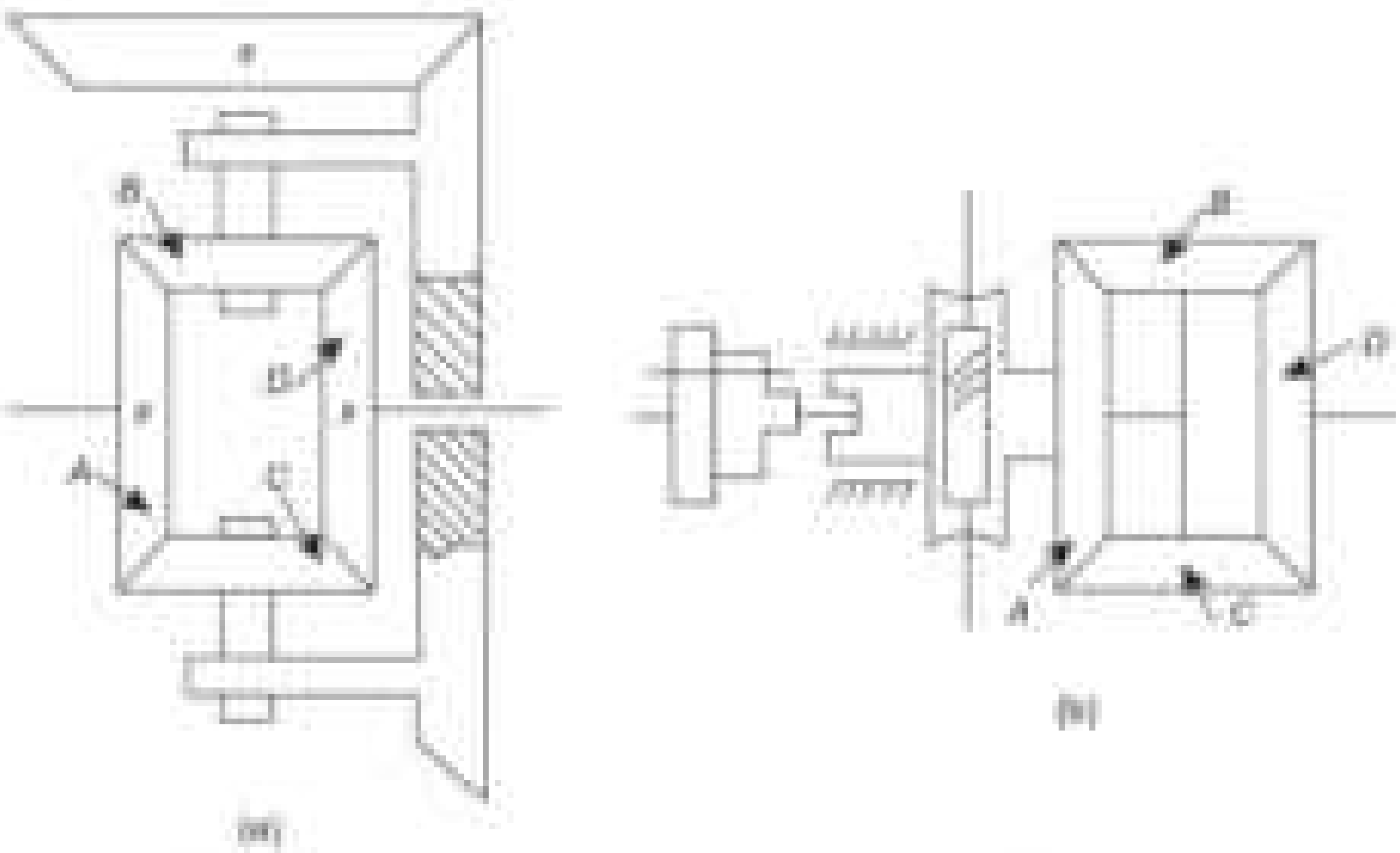


Fig. 1.52 *Differential mechanisms: (a) used in automobiles (b) used in machine tools*

If gears *A*, *B* and *D* make n_A, n_B and n_D revolutions per minute, respectively, then the transmission ratio of the kinematic train between gears *A* and *D* can be written as

$$\frac{n_A - n_B}{n_D - n_B} = -\frac{Z_A}{Z_B} \cdot \frac{Z_B}{Z_D}$$

where Z_A, Z_B and Z_D are the number of teeth of gears *A*, *B* and *D*, respectively. The minus sign indicates that gears *A* and *D* rotate in opposite directions if the rotation of the arm is stopped, *i.e.*, $n_B = 0$.

If $Z_A = Z_D$, the expression becomes

$$\frac{n_A - n_B}{n_D - n_B} = -1$$

wherefrom

$$n_A + n_D = 2n_B$$

In the automobile differential, the constancy of the sum $n_A + n_D$ indicates that when the vehicle is taking a turn a reduction in the rpm of one wheel is accompanied by an increase in the rpm of the other. If the automobile is travelling on a straight line, $n_A = n_D = n_B$, but if on a bend $n_A = 0$, wheel *D* begins to rotate at twice the speed of the ring gear, i.e., $n_D = 2n_B$.

1.5.5 Special Mechanisms and Devices

Special mechanisms and devices are employed in machine tool feed boxes. These mechanisms are:

1. Gear cone with sliding key
2. Norton gear mechanism
3. Meander's mechanism

They are discussed in Sec. 2.8.2.

1.5.6 Couplings and Clutches

Couplings and clutches are devices used for connecting one rotating shaft to another. If two shafts are permanently connected so that they can be disengaged only by disassembling the connecting device, the latter is known as a *coupling*. Devices that can readily engage shafts to transmit power and disengage them when desired are known as *clutches*.

Couplings Couplings are of two types:

1. Rigid
2. Flexible

Rigid couplings require that axial alignment between the connected shafts be maintained strictly. In flexible couplings, there is provision for compensating slight misalignments between the coupled shafts. A rigid coupling is shown in Fig. 1.53a and a flexible coupling in Fig. 1.53b.

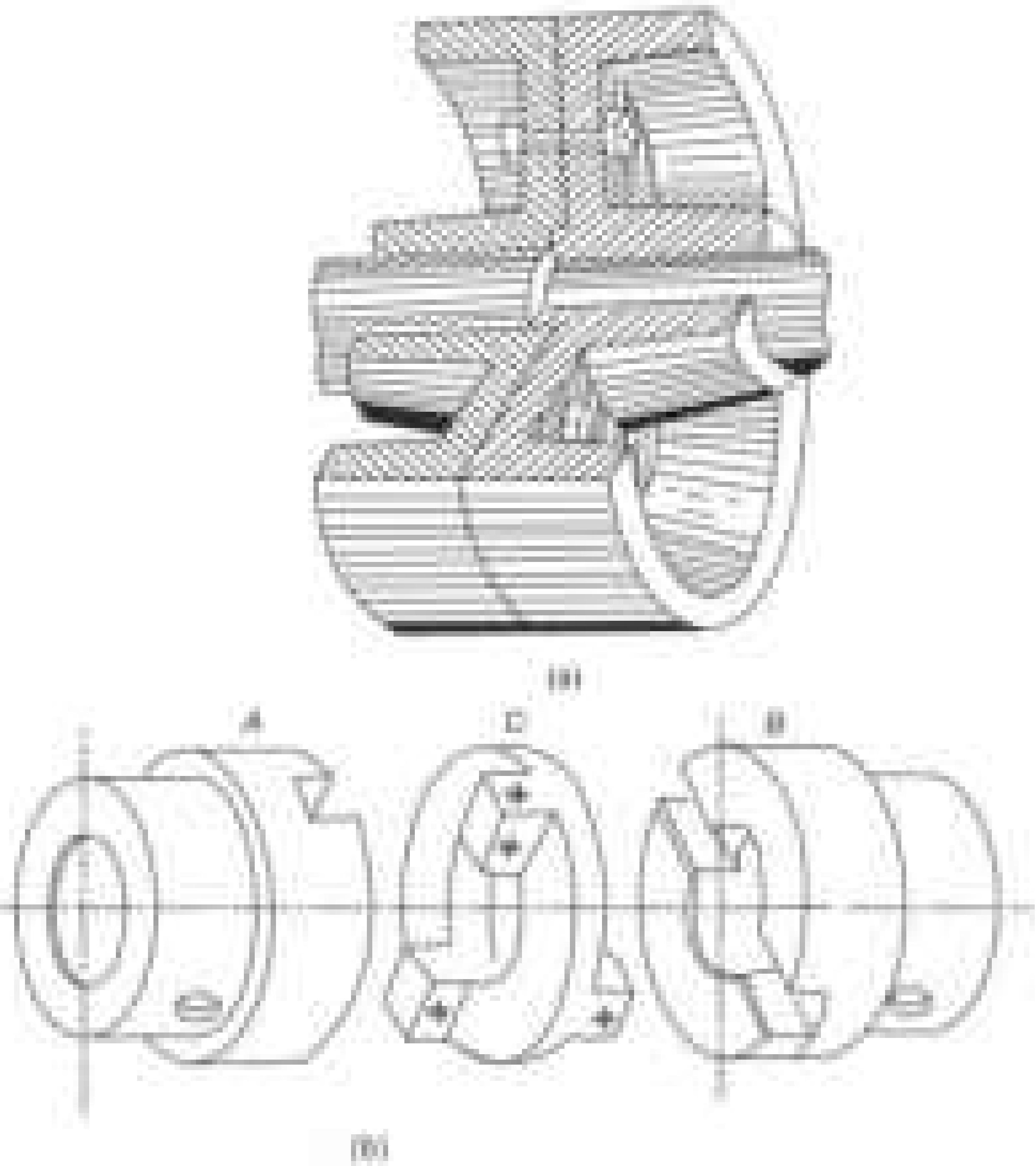


Fig. 1.53 *(a) Rigid coupling (b) Flexible coupling*

The flange coupling (Fig. 1.53a) consists of two flanges which are either press fitted on the ends of the shafts to be connected or mounted on keys. The flanges are drawn together by means of bolts. Torque is transmitted from one shaft to another either by the friction force between the faces of the flanges or by bolts.

The double slider or Oldham coupling consists of flanges *A* and *B* with diametrical slots and an intermediate plate *C* with projections that correspond to the slots of flanges *A* and *B*. Slight misalignment between the connected shafts is compensated by the plate sliding along the slots in the flanges.

If there is considerable misalignment between the shafts to be connected, an elastic flexible coupling (Fig. 1.54) can be used. In this coupling, the shafts are connected through a Cardan or Hooke's joint, which consists of yokes that are mounted on the ends of the shafts and a cross that provides a pivot joint between the yokes.

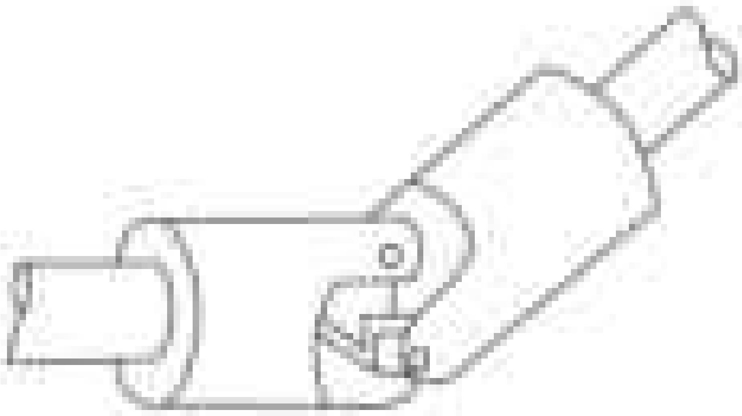


Fig. 1.54 *Elastic coupling*

Couplings are generally used in machine tools for connecting the motor shaft to the first shaft of the speed or feed box.

Clutches Clutches can be roughly classified into two major groups:

1. Positive-action clutches
2. Friction clutches

A positive-action clutch is incapable of slipping. It can be engaged only when the shafts to be connected are stationary or are rotating at identical speed. The most commonly used positive-action clutch is the jaw clutch (Fig. 1.55). The clutch consists of two halves, of which one is rigidly fixed on one of the connected shafts and is stationary, while the other is mounted on the second shaft on a key or splines and is moved into engagement. The faces of both the halves have projections, or so-called jaws and recesses such that the jaws of one fit into the recesses of the other and vice versa.

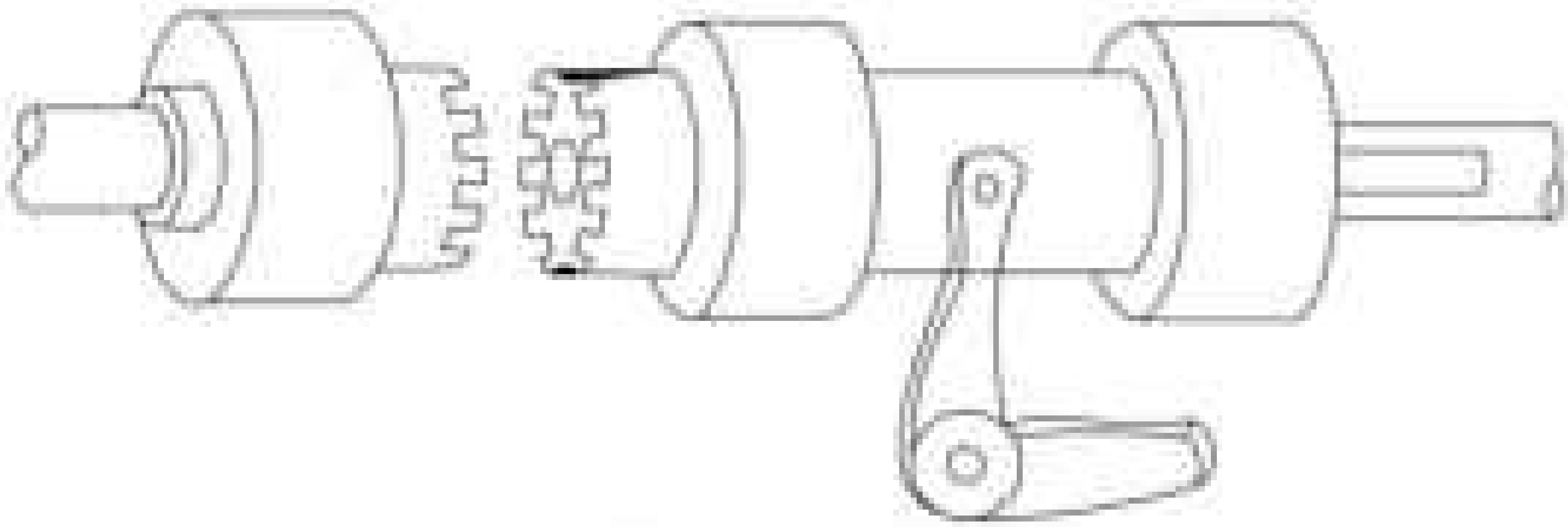


Fig. 1.55 *Jaw clutch*

A friction clutch, as the name implies, transmits torque by virtue of friction between the two halves. It can engage shafts rotating with different speeds or a rotating shaft with a stationary shaft. Friction clutches are generally not capable of transmitting large torques on account of slip. The commonly used friction clutches are discussed below.

A disc-type friction clutch consists of one or more discs which are pressed against each other between the flanges. Accordingly, the clutch is known as a single-disc or multiple-disc clutch. A multiple-disc clutch is schematically shown in Fig. 1.56. It consists of a cylindrical housing 1 with internal splines, flanged hub 2 with external splines, outer discs 3 with splines on their periphery and inner discs 4 with slines on their bore hole. The housing is rigidly mounted on one of the shafts and the sleeve on the other. Now, the discs are assembled by slipping them alternately along the splines of the housing and the hub. Thus, the outer discs rotate with the housing but are free to slide axially along its internal splines. Similarly, the inner discs rotate with the hub but can slide along its external splines. If the discs are to operate in oil, they are made of hardened steel. Since oil greatly reduces friction between discs, most clutches are operated dry. In such a case, the metal discs experience extensive wear, and therefore, one group of discs (generally outer one) is made of solid asbestos or a layer of asbestos is bonded on the metal discs.

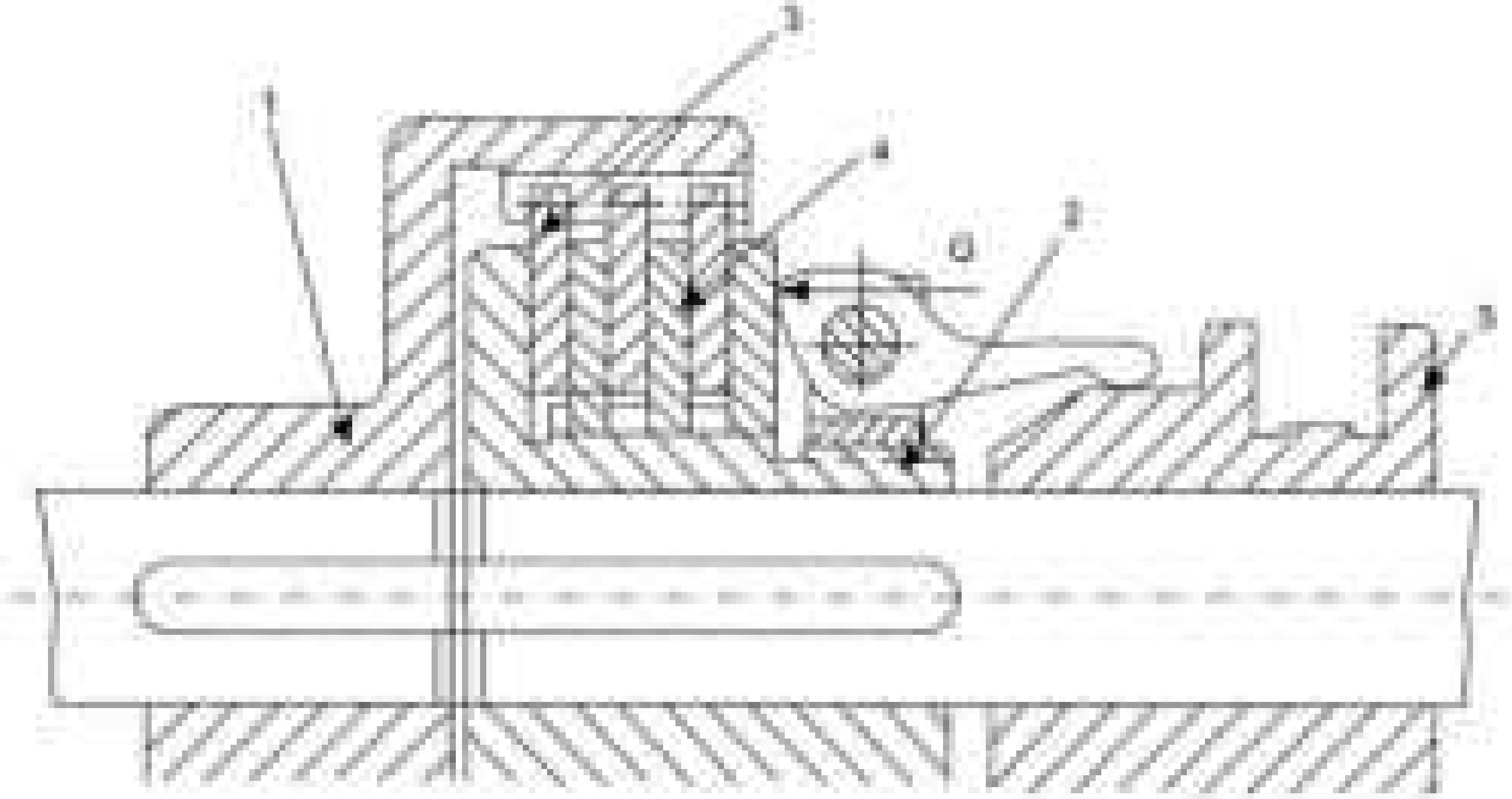


Fig. 1.56 *Multiple-disc friction clutch*

When the engaging sleeve 5 is moved towards the left, it exerts an axial force which is multiplied by the lever arrangement and applied on the friction discs. The discs get pressed against each other and the clutch gets engaged to transmit rotation between the two shafts. The lever system is so designed that it holds the clutch in engagement so that it is not necessary to continuously apply a force on the operating handle.

Disc-type friction clutches have large load-carrying capacity with small overall dimensions. They are distinguished by smooth engagement and their capacity can be easily varied by increasing or decreasing the number of discs according to the requirement. Generally, the number of discs does not exceed 10–12 because otherwise there is wear between rotating discs even when the clutch is disengaged.

In machine tools, electromagnetic clutches are lately finding increasing application. The electromagnetic clutch is essentially a multiple-disc friction clutch in which friction discs are pressed by an electromagnet. These clutches are particularly suitable for automatic control and are, therefore, being widely used in numerically controlled machine tools.

A cone-type friction clutch is shown in Fig. 1.57. It consists of two halves: one with an internal tapered surface is mounted on one shaft, while the other with an identical external taper is mounted on the other. One half is mounted rigidly, while the other is mounted on splines to permit axial displacement. The tapered surfaces are made of materials which have a large coefficient of friction and the clutch is engaged by pressing the two halves against each other. If the contacting surfaces are made of hardened steel, half-taper angle $\alpha = 8$–$10°$, while if the surfaces have an asbestos lining $\alpha = 12$–$15°$. Because of the taper of the friction surfaces, a relatively small axial pressing force provides a large force normal to the contacting surfaces which holds them together once the clutch is engaged. Therefore, in cone-type friction

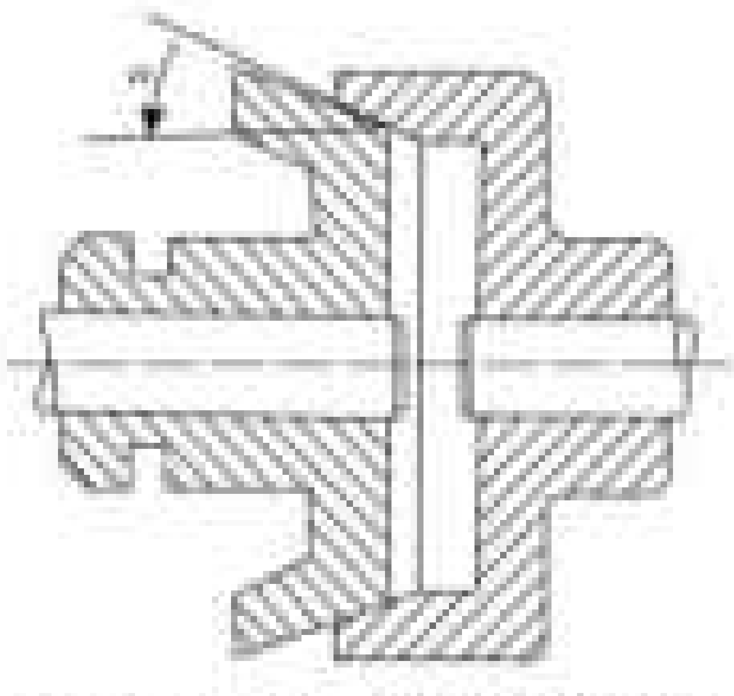


Fig. 1.57 *Cone-type friction clutch*

clutches, an elaborate linkage system is not required. This gives the cone clutch the advantage of simplicity. The major drawbacks that restrict the application of cone clutches in machine tools are their large dimensions and strict requirement of coaxiality between the connected shafts.

1.6 TECHNICO-ECONOMICAL PREREQUISITES FOR UNDERTAKING THE DESIGN OF A NEW MACHINE TOOL

The design and manufacture of a new machine tool can be undertaken only if it is economically justified. The design of a new machine tool can be justified in individual cases on the basis of higher productivity and accuracy, lower metal requirement per machine tool, less floor area per machine tool, etc. In general, all these indices which account for the cost of manufacture of the machine tool and its operation can be unified into a general index of economic effectiveness. The economic effectiveness of a machine tool, and for that matter of any equipment, can be quantitatively expressed through the total annual cost which is represented as

$$C_t = C + k \cdot \mathrm{CI}$$

where C_t = total annual cost

C = annual production cost

CI = capital investment

k = factor of capital recovery along with interest, generally $k = 0.15$–0.2

The design and manufacture of a new machine tool can be considered economically feasible if

$$C_{tn} < C_{te}$$

i.e.,

$$C_n + k_n(\mathrm{CI})_n < C_e + k_e(\mathrm{CI})_e \tag{1.19}$$

where subscript n stands for the new machine tool and e for the existing machine tool that is sought to be replaced or updated.

If the period of recovery of the capital investment is assumed to be the same in both the cases, i.e., $k_n = k_e = k$, then Eq. (1.19) yields

$$\frac{(\mathrm{CI})_n - (\mathrm{CI})_e}{C_e - C_n} < \frac{1}{k} \tag{1.20}$$

Keeping in mind the relationship $T = 1/k$, where T is the period of recovery of the capital investment, Eq. (1.20) can be rewritten as

$$\frac{(\mathrm{CI})_n - (\mathrm{CI})_e}{C_e - C_n} < T \tag{1.21}$$

The total annual cost is a convenient criterion not only for assessing the viability of new equipment, but also for comparing different design versions and methods of implementing these designs.

The factors involved in Eqs (1.19)–(1.21) will now be elaborated to ensure that no important effects are missed while doing the total cost estimation.

Capital Investment (CI) The capital investment consists of

1. Net expenditure on equipment and fixtures for manufacturing the new machine tool

$$E_n = aE_p$$

where E_n = net expenditure on equipment and fixtures
E_p = expenditure on purchase of equipment and fixtures
α = factor that takes into account extra expenditure on transportation and installation of equipment and fixtures; α = 1.1 for machine tools and 1.18 for transfer lines.

The cost of fixtures must always be taken into account when versions using principally different equipment are being compared (e.g., general-purpose machine tools with special-purpose machine tool, machine tools having manual controls with numerically controlled machine tools, etc.).

The expenditure on fixtures is approximately calculated from the following relationship:

$$E_f = \sum_{i=1}^{n} E_{mf} \cdot q$$

where E_f = expenditure on fixtures
n = number of different parts to be machined on the equipment in the course of one year
E_{mf} = mean cost of one fixture
q = number of fixtures required for each part

2. Expenditure on building the production premises

$$E_{bp} = E_{bpm} A_p \cdot \gamma$$

where E_{bp} = expenditure on building the production premises
E_{bpm} = mean expenditure on building 1 m^2 of production premises
A_p = total area under equipment
γ = factor that takes into account the additional area necessary for proper functioning and layout of premises; γ = 1.5–5 depending upon area A. As area A increases, the value of γ decreases; for A = 75 m^2, γ = 1.5

3. Expenditure on building the servicing premises

$$E_{bs} = E_{bsm} \cdot A_s$$

where E_{bs} = expenditure on building the servicing premises
E_{bsm} = mean expenditure on building 1 m^2 of servicing premises
A_s = total area of servicing premises

Thus

$$CI = E_n + E_{bp} + E_{bs}$$

Annual Cost of Production (C) The annual cost of production of the designed machine tool is

$$C = N \cdot \sum_{i=1}^{n} C_i$$

where C = annual cost of production
N = annual output
C_i = cost of the ith part of machine tool
n = number of parts in the machine tool

The production cost of a part includes

1. cost of material of the workpiece; this is fixed cost and may not be taken into account when two versions are being compared,
2. wages paid to labour,
3. overheads: these cover the recurring expenditure on cutting tools, expenditure on the maintenance and repair of equipment and fixtures, expenditure on preparing skilled workers and technical personnel, expenditure on operating the equipment, etc.

All cost factors need not necessarily be taken into account when two versions are being compared; it generally suffices to restrict the analysis only to those expenditures, which substantially differ in the compared versions.

The design and manufacture and subsequent industrial application of new models of machine tools is one of the major factors in increasing productivity. Periodic renovation of production capacities is essential to ensure a normal economic growth rate. Each industrial application of new equipment is accompanied by some initial investment; this must be recovered during the pay-back period, and for the remaining period of its life till obsolescence, the new production capacity should bring in profit due to reduced production cost. The search for new design and production solutions must, therefore, be based upon a thorough economic analysis on the lines discussed above.

1.7 GENERAL REQUIREMENTS OF MACHINE TOOL DESIGN

Any machine tool should satisfy the following requirements:

1. High productivity
2. Ability to provide the required accuracy of shape and size and also necessary surface finish
3. Simplicity of design
4. Safety and convenience of controls
5. Good appearance
6. Low cost of manufacturing and operation

We shall discuss how these requirements are met in the design of machine tools.

1. Productivity Productivity of a metal cutting machine tool is given by the expression,

$$Q = \frac{1}{t_c + t_{np}} \cdot \eta \tag{1.22}$$

where t_c = machining time

t_{np} = non-productive time that includes job handling time, tool handling time, time of idle travel prior to commencement of cut, time of idle travel for guiding the tool to home position after completion of cut, set up time, inspection time and time spent on unscheduled delays.

η = factor that accounts for stoppages for maintenance as well as unscheduled stoppages on account of breakdowns.

Based on Eq. (1.22), productivity of a metal cutting machine tool may be raised by the following methods:

(i) *Cutting down machining time:* This is possible if high cutting speeds and feed rates are available on the machine tool in accordance with the latest developments in cutting tool materials and design. At the design stage itself, the machine tool must be provided with a margin to accommodate future developments so that it does not become obsolete in a short period of time.

The application of stepless mechanical, hydraulic and electrical drives also helps in reducing machining time as the optimum cutting speed can be accurately set without reducing its value to the nearest available rpm on the machine tool with a stepped drive. Machining time can also be reduced by making provision for simultaneous multiple cuts and use of coolants.

(ii) *Cutting down non-productive time:* This can be achieved by using jigs and fixtures that reduce clamping and unclamping time, and mechanising and automating machine tool controls. During the last few decades, developments in machine tool design have been largely directed at reducing the non-productive time through automation. Hard automation in the form of automatic machines, mechanised flow lines and transfer lines reflected this trend till the 60s. However, with growing affluence in the industrialised nations, the consumers became more discerning and since the 70s the demand pattern has changed from mass produced goods to batch produced and custom built goods. This triggered a change in the manufacturing philosophy from one of hard automation to soft automation that is manifested today in the increasing proliferation of numerically controlled and computer numerically controlled machine tools, machining centres, robots, flexible manufacturing systems, etc.

(iii) *Machining with more than one tool simultaneously:* This principle is employed in multiple-spindle lathes, drilling machines, etc.

(iv) *Improving the reliability of the machine tools* to avoid break downs and adopt proper maintenance policy to prevent unscheduled stoppages and delays.

2. Accuracy The accuracy of a machine tool depends upon its geometrical and kinematic accuracy and its ability to retain this accuracy during operation. Accordingly, the ability of a machine tool to consistently machine parts with a specified accuracy within permissible tolerance limits can be improved by the following methods:

(i) *Improving the geometrical accuracy of the machine tool:* This is mainly determined by the accuracy of guiding elements, such as guideways, power screws, etc. It is also essential to ensure uniform, jerk-free movement of the traversing member of the machine tool.

(ii) *Improving the kinematic accuracy of the machine tool:* The kinematic accuracy determines the relationship between velocities of two or more forming motions and it depends upon the length of kinematic trains and the accuracy of manufacture and assembly of components. Obviously, the kinematic accuracy of a machine tool can be improved by using as short kinematic trains as possible, and manufacturing and assembling the components with a high degree of accuracy.

(iii) *Increasing the static and dynamic stiffness of machine-tool structures:* The greater is the static stiffness of the machine-tool structure, the smaller will be its deformation due to the cutting forces and hence the higher will be the accuracy of machining. A high dynamic stiffness reduces the vibrations during machining and hence provides better accuracy and surface finish.

(iv) *Providing accurate devices for measuring distance of travel:* This concerns the accuracy of manufacture of dials, scales, verniers, optical systems, etc. The accuracy of measuring instruments is of paramount importance in machine tools with automatic size control during machining, e.g., automatic machines, machine tools with adaptive controls, etc.

(v) Arranging the machine tool units in such a manner that the thermal deformations during the machining operation result in the least possible change in the relative position between the tool and the workpiece. This factor is especially important in machine tools used for finishing operations, e.g., grinding machines.

3. Simplicity of Design Simplicity of design of machine tools determines the ease of its manufacture and operation. The design of a machine tool can be simplified by using standard parts and subassemblies as far as possible. The complexity of design of a machine tool depends to a large extent upon the degree of its 'universality'. Thus a general-purpose machine tool is, as a rule, more complex than a special-purpose machine tool doing similar operations. The design of a machine tool can, therefore, be simplified by putting restrictions on its range of application, e.g., on the type of different operations that may be carried out, or on the size of parts which may be machined, etc.

4. Safety and Convenience of Controls A machine tool cannot be deemed fit for use unless it meets the requirements of safety and convenience of operation. Safety of controls is achieved by taking, among others, the following measures:

(i) Shielding the rotating and moving parts of the machine tool with hoods.
(ii) Protecting the worker from chips, abrasive dust and coolant by means of screens, shields, etc.
(iii) Providing reliable clamping for the tool and workpiece.
(iv) Precluding the possibility of accidental pressing of push buttons and handles.
(v) Providing reliable earthing of the machine.
(vi) Providing devices for safe handling of heavy workpieces.
(vii) Providing blocking devices which preclude simultaneous engagement of conflicting transmissions.
(viii) Providing travel limiting devices for traversing machine tool members and also devices for overload protection.

The convenience of machine-tool controls is intimately linked with their safety. Convenient controls will protect the worker from excessive fatigue and thus contribute towards safety. The convenience of controls also determines to a large extent the quality of the workers' performance. Machine tool controls should be simplified and made convenient for which a few guidelines are given below:

1. The control system should be rationally selected and should be automated to as large an extent as possible.
2. The control system should be designed by giving due consideration to ergonomic principles.

5. Appearance Good appearance of the machine tool influences the mood of the worker favourably and thus facilitates better operation. It is generally conceded that a machine tool that is simple in design and safe in operation is also good in appearance, although factors, such as external finish, colour, etc., do substantially contribute to the overall aesthetic quality of the machine tool. For instance, painting of machine tools in grey-green or green-blue colours impart a bright and pleasing appearance to the shop. Nowadays, painting of machines in different colours according to the production purpose is becoming popular, e.g., transportation facilities within the shop are painted yellow with black stripes, etc.

6. Low Cost of Manufacturing and Operation The cost of manufacturing a machine tool is determined by the complexity of its design. Therefore, factors that help in simplifying the machine tool design also contribute towards lowering its manufacturing cost. The cost can also be brought down by reducing the amount of metal required in manufacturing the machine tool. This is achieved by using stronger materials and more precise design calculations pertaining to the strength and rigidity of parts to keep the safety margins as low as possible. For instance, considerable saving of metal can be achieved by using welded steel structures instead of cast iron for heavy parts, such as beds, columns, bases, etc. It should also be noted that a reduction of the weight and dimensions of the machine tool also makes transportation and installation of the machine tool easier and cheaper, thus indirectly contributing to a further reduction of the overall cost.

1.8 ENGINEERING DESIGN PROCESS APPLIED TO MACHINE TOOLS

Design is undoubtedly a creative process. Many people mistakenly attribute this creativity to a flair for design in certain persons who become successful designers. In fact, the engineer with a 'flair' for design is, as a rule, a man with a logical decision-making ability by which he explores all possible solutions to a given problem and arrives at an optimum after carefully analysing all the alternatives.

Until a decade or so ago, any design which was technically feasible, *i.e.*, capable of being manufactured, would generally go through to production, irrespective of the time or cost involved. However, the need to find principally new design solutions to keep ensuring higher productivity has in recent times greatly increased the expenditure on design. Design is progressively becoming a team activity as optimum solutions can be found only by considering a large number of factors of diverse nature with which the designer may not always be well conversant.

The block diagram of Fig. 1.58 shows how design is related to different engineering, economic, natural and social sciences.

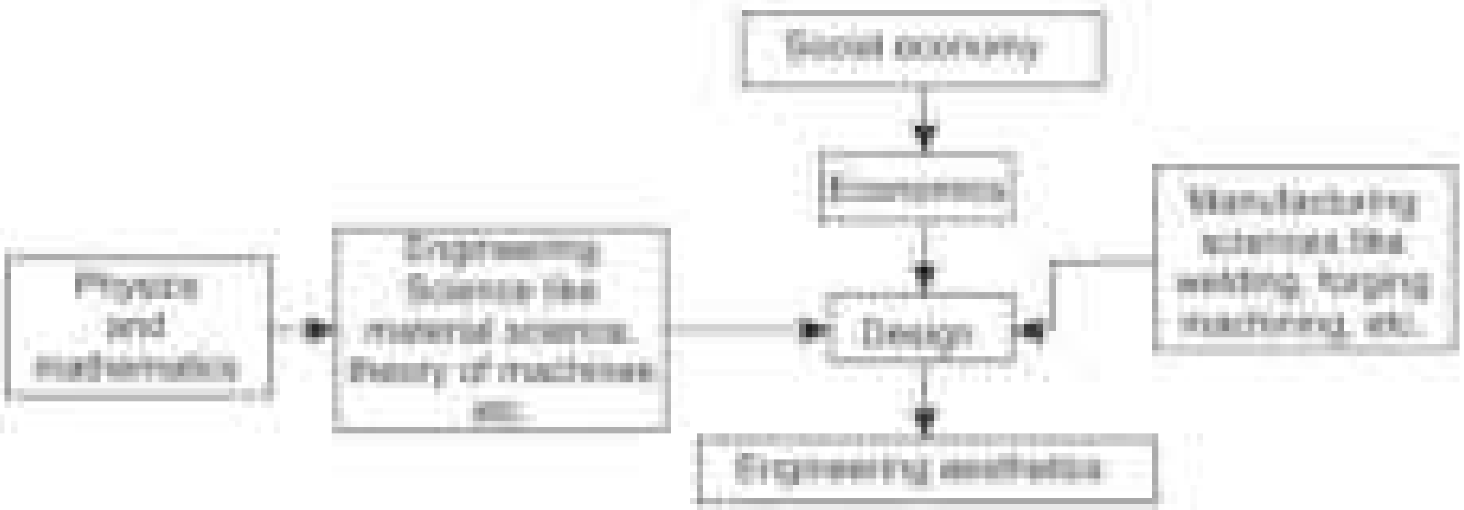


Fig. 1.58 *Block diagram depicting the influence of various sciences on design*

In view of the heavy responsibilities on the designer and the large expenditure involved in designing a new machine, it is necessary to streamline the design process so that a sound design solution is achieved with minimum expenditure.

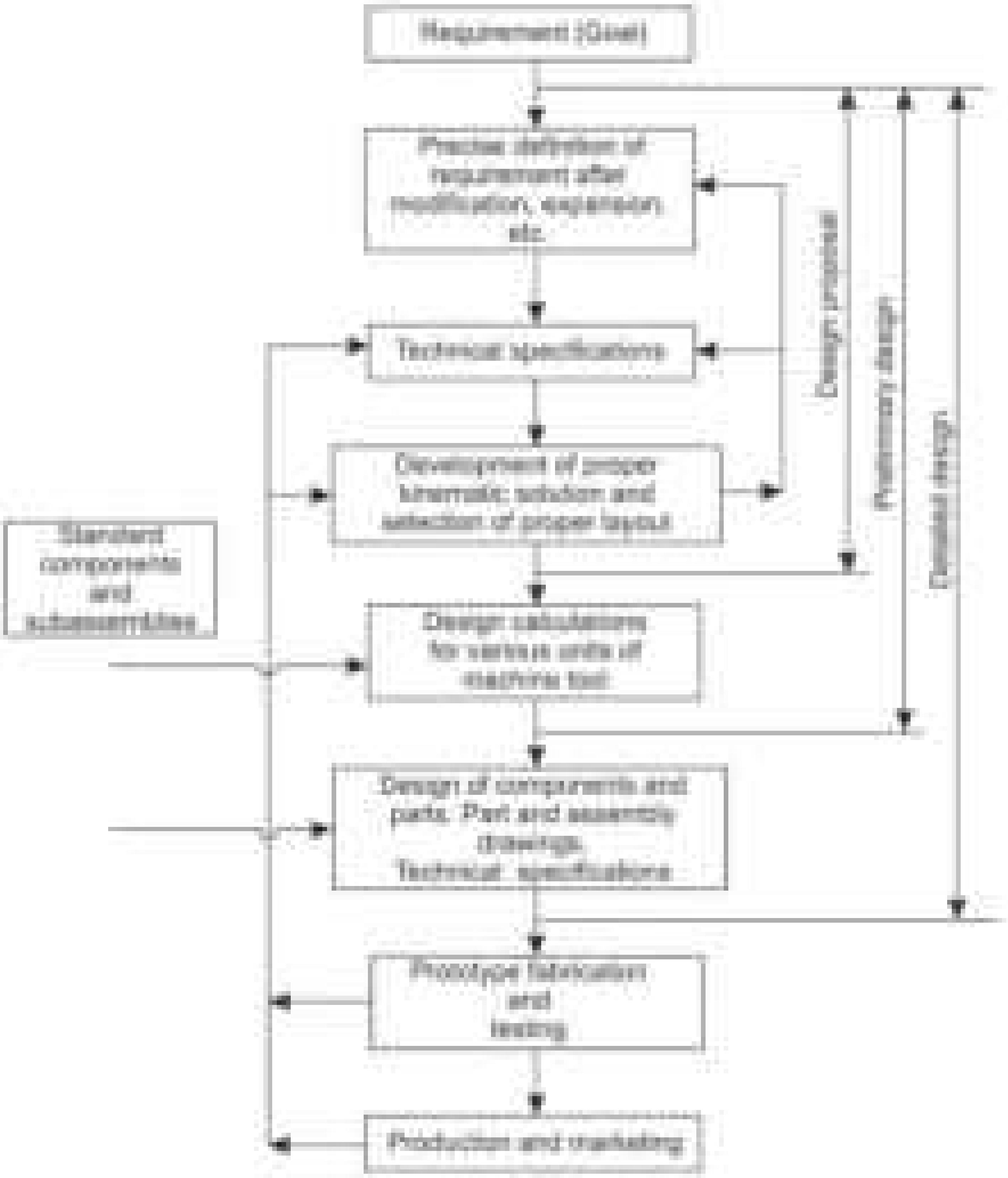


Fig. 1.59 *Block diagram of the design process in respect of machine tools*

The design process for designing a new machine tool is presented in the form of a block diagram in Fig. 1.59. It is evident from Fig. 1.43 that the design process is carried out in three important stages:

1. Design proposal
2. Preliminary design
3. Detailed design

At the end of each stage, the design must be subjected to a critical feasibility analysis, and a technical report prepared and submitted to the customer. The steps involved in the design process will now be elaborated one by one.

1. Requirement The customer outlines the requirement by furnishing information about the parts for machining of which he wants the machine tool to be designed. The information should include the nomenclature of parts and their annual output, the dimensions and shapes of surfaces to be machined, materials of the parts, machining tolerances, and the quality of surface finish required. This information serves as the basis for selecting appropriate machining methods and cutting tools.

The customer, owing to his lack of specialist knowledge, may often not be in a position to define his need exactly. The designer must first of all check that the requirement of a new design is genuine and a suitable product does not already exist. A consideration in undertaking a new design, in the presence of available solutions, may be the need to make the product economically viable by reducing its cost. The designer must, therefore, make a preliminary assessment of the requirement to see whether it is economically feasible. If necessary, he may, in consultation with the customer, modify or expand the requirement to increase the market potential of the designed machine tool.

2. Technical Specifications The technical specification is a listing of parameters that are essential for the design. The information furnished by the customer about the parts forms the basis of determining important machine tool specifications, such as the range of the speed of the main motion, speed of auxiliary motions, power rating of the electric motor, etc. Besides quantifiable items, the designer must also specify factors that in themselves cannot be quantified but are of utmost importance to design, e.g., the method of speed and feed rate regulation, degree of mechanisation and automation to be employed on the machine tool, appearance of the machine tool, etc. In general, the designer should frame the specifications in a manner that does not unnecessarily narrow the range of possible solutions. It should be remembered that incorrect specifications are one of the major sources of overdesign and redundant features in the finished product.

The final design depends to a great extent upon the relative importance of the different specification items. A comparison of the importance of specification items for a vacuum cleaner and aircraft is given in Fig. 1.60[7]. A listing of specifications in this manner helps the designer in properly appreciating his priorities and prevents him from deviating from the basic goal.

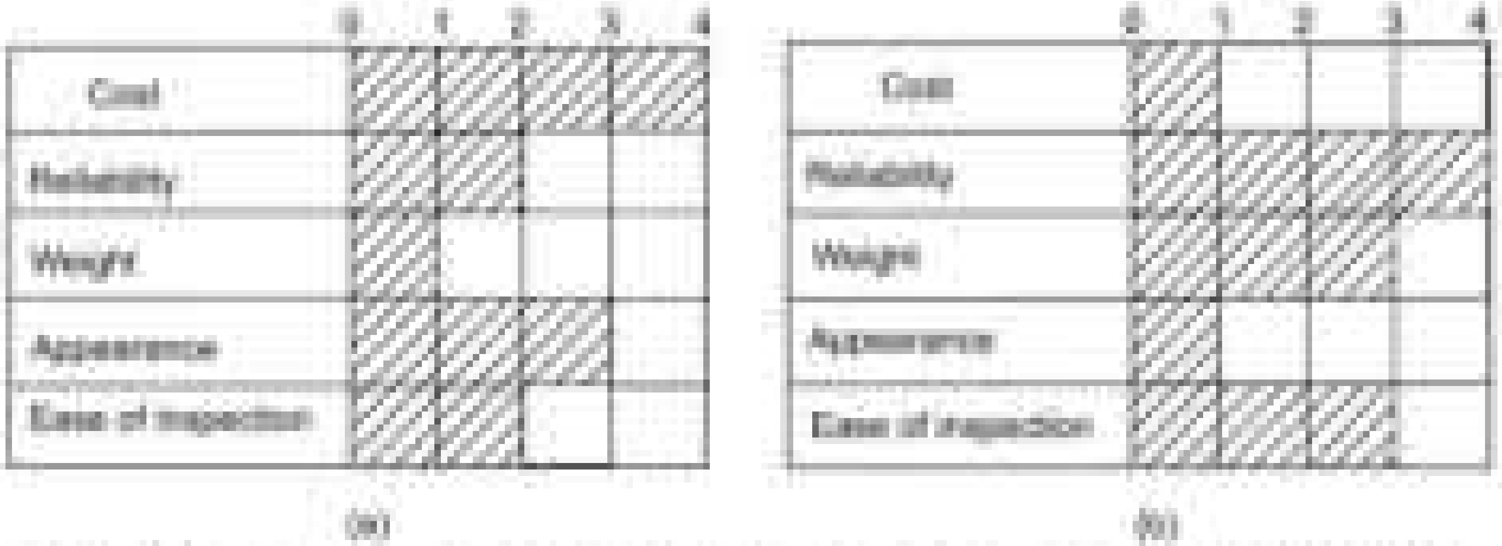


Fig. 1.60 *Importance of specification items for: (a) Vacuum cleaner (b) Aircraft*

3. Selection of Proper Kinematic Solution and Layout After technical specifications have been laid down, the designer explores the combinations of relative motions that can ensure machining of surfaces of required shapes and dimensions. The different possibilities are evaluated and those found technically feasible are selected. Kinematic solutions on the basis of basic motion combinations are now developed. All these solutions are analysed for their technical feasibility and infeasible solutions are screened. A kinematic solution correlates the motions of the workpiece and cutting tool and can be realised in a number of layouts of major machine tool units. A technical feasibility analysis, keeping in mind the constraints of the requirements and technical specifications, is again carried out to select the best possible layout. This aspect will be elaborated in Sec. 1.9.

4. Design Calculations Design calculations cover the design of the major units of the machine tools, such as the speed box, feed box, bed, spindle, etc. These calculations are done in accordance with design procedures only for those versions that are found most suitable on the basis of the preceding analysis. The final version is selected by comparing the economic feasibility of implementation of alternatives.

5. Drawings of Components and Assemblies These drawings are made for the version that is finally selected. The drawings must be complete with dimensions, tolerances and manufacturing specifications (including the manufacturing method to be employed). Special care should be taken during the stages of design calculations and detailed drawing to make use of standard components and assemblies as far as possible.

It should be appreciated that design is essentially an iterative process. The feedback that is received after prototype fabrication and testing, and particularly after marketing the product must be carefully analysed to make appropriate changes in technical specifications and subsequent design. In the design process itself, the designer should adopt a flexible attitude and be prepared to make modifications in the technical specifications and even the requirements, if these are conducive to more sound and/or economic design. These aspects are indicated in the design process (Fig. 1.59) by feedback loops.

1.9 LAYOUT OF MACHINE TOOL

The layout of the machine tool must provide the required combination of forming and setting motions that are necessary for the given machining process. The required relative motions between the cutting tool and workpiece are generally realised by means of a set of translatory and rotary motions. The layout of the machine tool will typically consist of one stationary block and a number of moving blocks divided by linear or circular guideways, the number of guideways being equal to the number of elementary motions provided on the machine tool. A particular layout is obtained by placing the stationary and moving blocks in a particular order. Different layouts are obtained by changing the order of these blocks. It is the task of the designer to analyse the various layout alternatives and select the best possible version, consistent with the constraints of the particular machine tool.

The selection of a suitable layout can best be carried out by structural analysis using the Boolean-algebra technique. In this method, the machine tool structure of any complexity can be represented in the form of a combination of symbols. Let us introduce a set of symbols for this purpose.

Let X, Y, Z represent the basic reciprocating displacements along the corresponding co-ordinate axes and U, V, W the additional displacements in the same directions.

A, B, C represent rotary motions about axes X, Y, Z while D and E represent the additional rotary motions. The lower case letters of all the symbols defined above indicate the auxiliary setting motions for the corresponding coordinate axes, e.g., x represents auxiliary setting motion in the x-direction, while a represents an auxiliary rotary motion about the same axis.

Boolean algebra permits the consecutive linking of blocks which is represented as a conjunction (AND) and parallel linking which is represented as a disjunction (OR). The consecutive linkage of blocks may not be indicated by anything or may be indicated by a full stop (.). Parallel blocks are written in brackets and parallel linking is indicated by a plus sign (+). The layout formula begins with the block carrying the workpiece and ends with the block carrying the cutting tool.

Some examples of machine tool layouts and their layout formulae are given in Fig. 1.61.[8] Figure 1.61a shows the layout of a knee-type vertical milling machine with consecutive linking of blocks. In the layout formula $XYZOC_v$,

X	represents table travel
Y	represents cross-slide travel
Z	represents knee travel
O	represents the stationary block (column)
C	represents rotation of spindle about the Z-axis

subscript v indicates that the spindle is vertical.

The lathe layout shown in Fig. 1.61b also consists of blocks linked in series. In the layout formula $AOXYcwd$,

A	represents rotation of the workpiece clamped in the spindle about the X-axis
O	represents the stationary block (bed)
X	represents carriage travel along bed guideways
Y	represents cross-slide travel
c	represents rotary setting motion of the compound slide
w	represents setting motion of the tool post
d	represents rotary setting motion of the tool post

The layout of the gear-shaping machine is shown in Fig. 1.61c. In the formula $DaOs(CZ)_v$,

D	represents rotation of workpiece about the vertical axis
a	represents setting motion of the table
O	represents the stationary block (column)
s	represents tool head travel along cross-rail guideways
(CZ)	indicates that the cutting tool experiences two simultaneous cutting motions—translatory motion along the Z-axis and rotation about the same axis
v	indicates that the spindle is vertical

The unit built drilling machine shown in Fig. 1.61d consists of blocks linked in parallels. In the layout formula $yaO(X4A + Y4B_h + Z5C_v)$,

y	represents setting motion of the workpiece
a	represents rotary setting motion of the workpiece
$X4A$	indicates that there are four spindles rotating about the X-axis in the spindle head which itself travels in the direction of the X-axis
$Y4B_h$	indicates that there are four spindles rotating about the Y-axis in the spindle head which itself travels in the direction of the Y-axis; subscript h indicates that the spindles are horizontal
$Z5C_v$	indicates that there are five spindles rotating about the Z-axis; subscript v indicates that these spindles are vertical

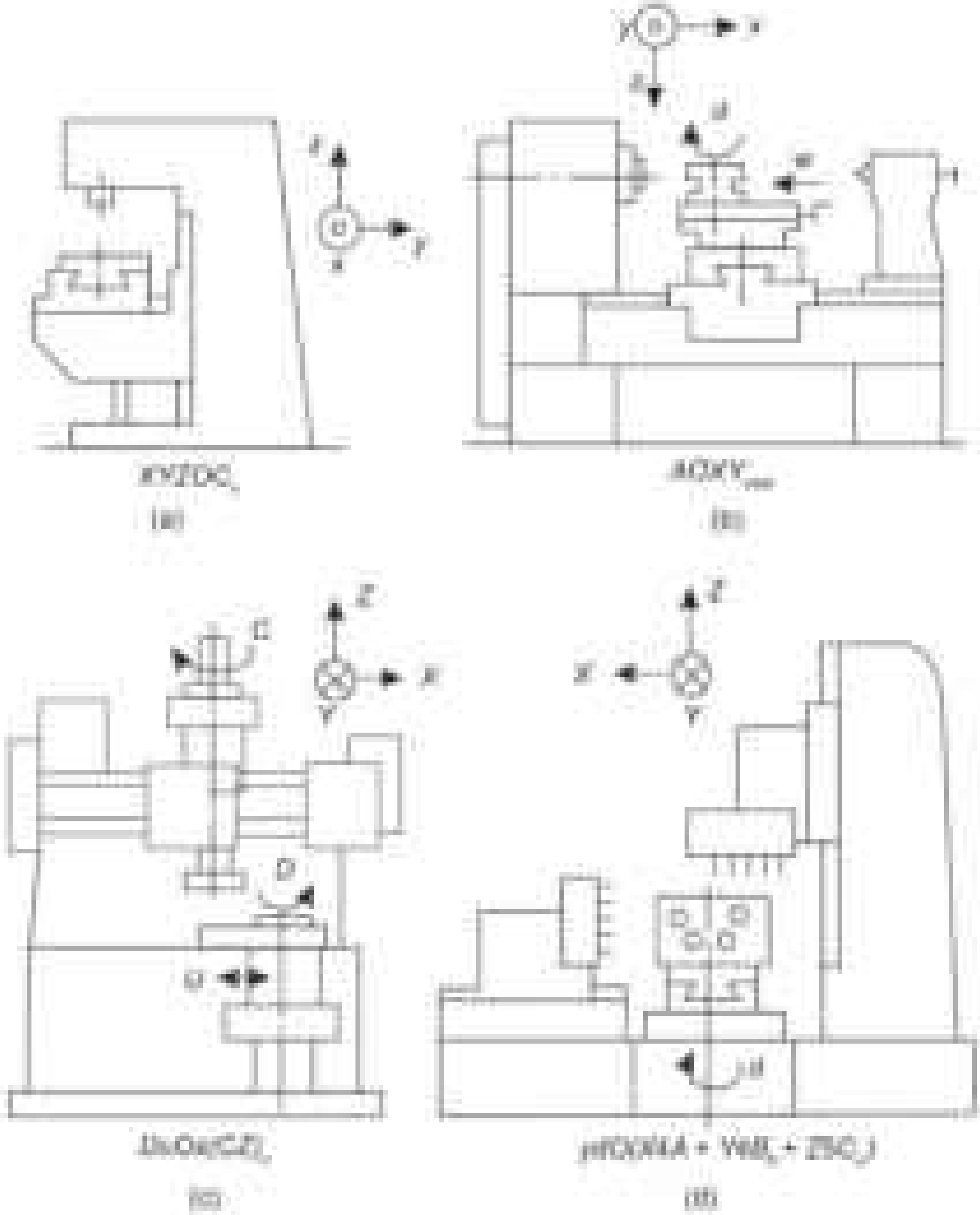


Fig. 1.61 *Layout and layout formulae for: (a) Knee-type vertical milling machine (b) Lathe (c) Gear-shaping machine (d) Unit-built drilling machine*

The examples give a fair idea that an appropriate layout formula can be written for a machine tool of any complexity. The formal representation of the machine tool layout in the form of a layout formula enables us to obtain all possible layout versions by a mere readjustment of the symbols. The best layout is selected by a comparative analysis of all the versions.

Consider a milling machine which is a triple co-ordinate machine consisting of four blocks—one stationary which is denoted by *O* and three moving denoted by *X*, *Y* and *Z*. The spindle rotation is not a forming motion and, therefore, does not affect the layout. The different layout versions are obtained by the permutation of symbols *XYZO*. The total number of possible versions is = 4! = 24. These versions are tabulated in the matrix shown in Fig. 1.62a. The columns of this matrix differ in terms of location of the stationary block, while the lines (in pairs) differ by the location of the vertically moving block *Z*. We can, for instance, isolate a subgroup of layouts with a moving column. These versions correspond to the matrix elements of Fig. 1.62a enclosed within the thick line and are shown in Fig. 1.63.

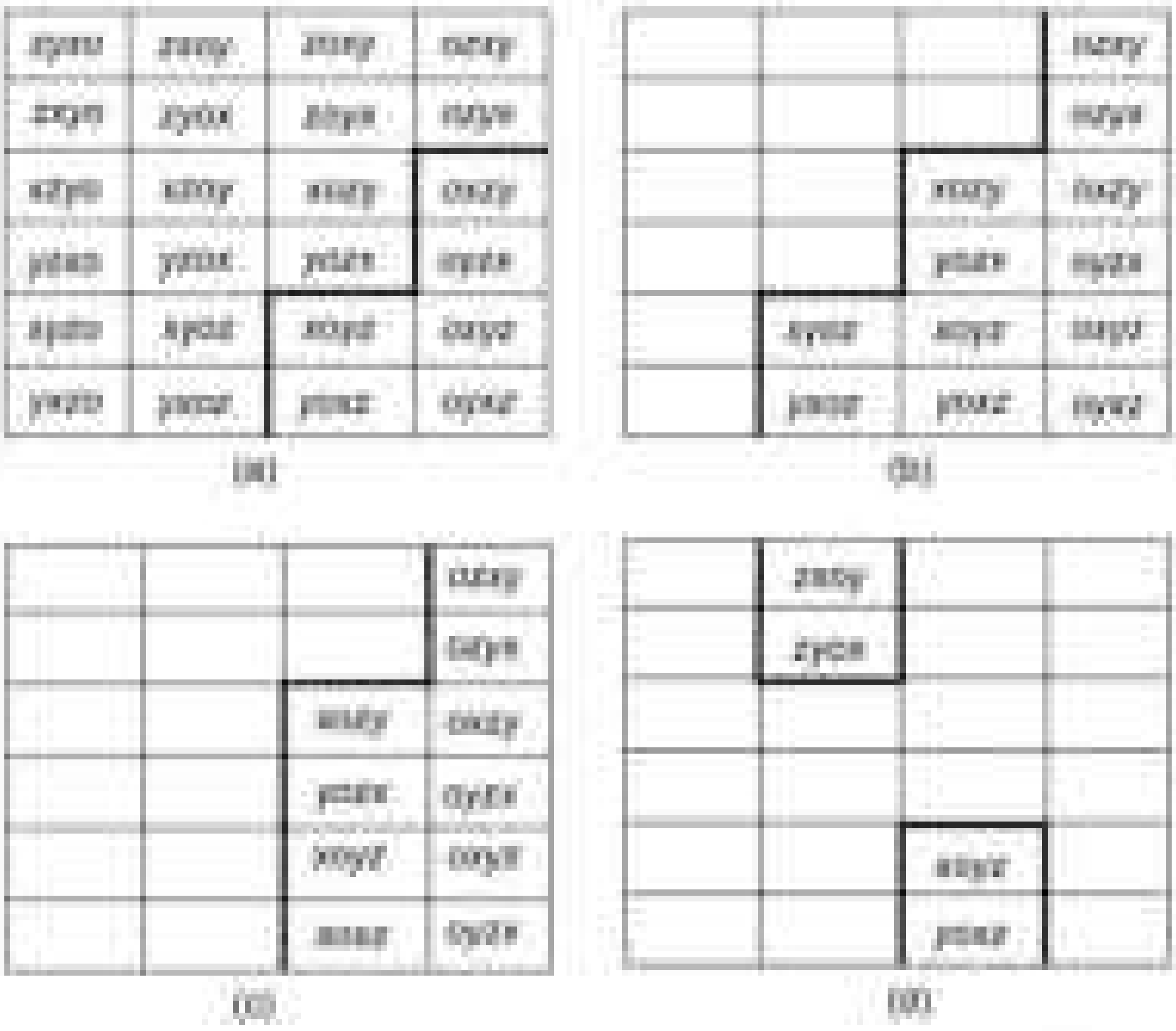


Fig. 1.62 *Matrices of layout versions: (a) Complete matrix for layout formula XYZO (b) Submatrix for layouts in which vertical displacement is absent (c) Submatrix for layouts in which the workpiece receives only horizontal displacement (d) Submatrix for layouts in which the horizontal displacement occurs in the immediate vicinity of stationary block*

The selection of the best layout from among the possible alternatives is done by successive elimination of those versions which do not satisfy the requirements (constraints) necessary for optimum functioning of the designed machine tool. The constraints are also formulated in the form of generalised structural formulae. For this purpose, the following new notations need to be introduced:

- $\vec{O}$ represents a moving block pertaining to cutting tool displacement
- $\bar{Z}$ represents horizontal moving block

Let us write a few requirement (constraint) statements with the help of these symbols.

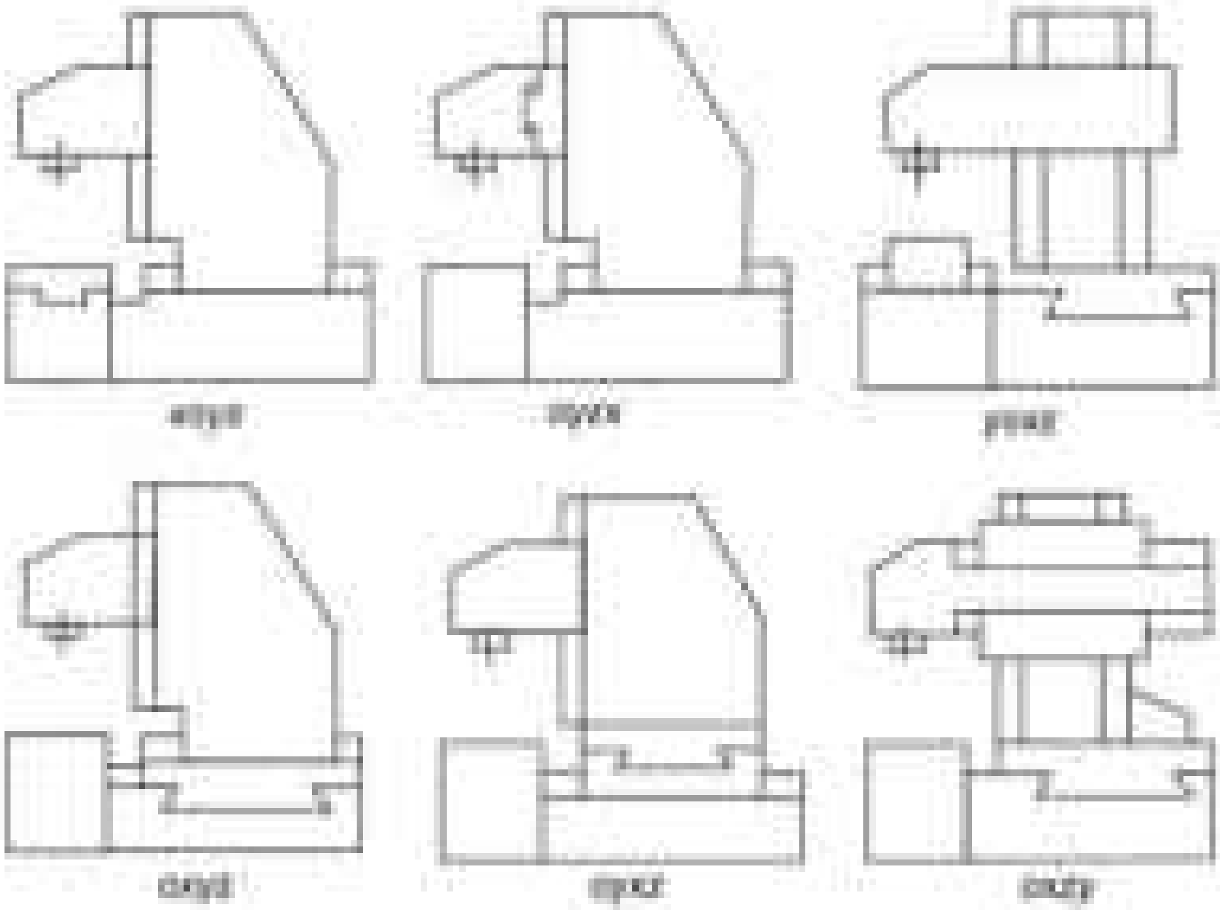


Fig. 1.63 *Layouts of vertical milling machine with a moving column*

Statement 1 The machine tool is meant for machining heavy parts. Therefore, it is not desirable that the machine tool table be given vertical displacement.

The requirement may be stated as

$$R_1 = \bar{Z}\bar{Z}O\bar{O} + \bar{Z}O\bar{O}\bar{O} + O\bar{O}\bar{O}\bar{O}$$

In this statement the "+" sign represents the OR function; the statement can be interpreted as indicating that the requirement can be satisfied by imparting two horizontal motions to the workpiece, followed by the stationary block and then one (obviously vertical) motion to the cutter

or

imparting one horizontal motion to the work piece followed by the stationary block and then two motions (one horizontal and one vertical) to the cutting tool

or

rigidly attaching the workpiece to the stationary block and imparting all three motions (two horizontal and one vertical) to the cutting tool.

Statement 2 The heavy parts must be machined with a high degree of accuracy. Therefore, to prevent the weight of the part from affecting the accuracy, the workpiece should be stationary or have only one horizontal displacement.

The requirement may be stated as

$$R_2 = \bar{Z}O\bar{O}\bar{O} + O\bar{O}\bar{O}\bar{O}$$

Statement 3 To prevent the weight of the moving assembly with the workpiece from affecting the machining accuracy, the horizontal moving block must be adjacent to the stationary block.

The requirement may be stated as

$$R_3 = Z\bar{Z}O\bar{Z} + \bar{Z}O\bar{Z}Z$$

Consider now that the requirements (constraints) formulated in the above statements ought to be satisfied by the triple coordinate milling machine for which the best layout has to be selected.

The layout versions which satisfy requirement 1 are shown in Fig. 1.62b, while those satisfying requirements 2 and 3 are shown in Fig. 1.62c and Fig. 1.62d, respectively.

To locate the versions that simultaneously satisfy all the requirements, the requirement statements are written in the matrix form so that the stationary blocks occupy identical positions.

	1	2		3		4
R_1	–	$\bar{Z}\bar{Z}O\bar{O}$	+	$\bar{Z}O\bar{O}\bar{O}$	+	$O\bar{O}\bar{O}\bar{O}$
R_2	–	–		$\bar{Z}O\bar{O}\bar{O}$	+	$O\bar{O}\bar{O}\bar{O}$
R_3	–	$Z\bar{Z}O\bar{Z}$	+	$\bar{Z}O\bar{Z}Z$		
	ϕ	ϕ		$XOYZ$	ϕ	
				$YOXZ$		

The columns in which there is no formula or which contain contradictory statements are rejected (ϕ). Thus for the given machine tool, the optimum layout lies in column 3. A comparison of this conclusion with Fig. 1.62b, c and d immediately provides the answer that the best layouts are $XOYZ$ and $YOXZ$. These layouts are shown in Fig. 1.64. From among these two versions, which are equal in all respects from the point of view of technical feasibility, the final choice is determined by a thorough analysis of static and dynamic stiffness of the two structures, ease of control, cost of manufacture, etc.

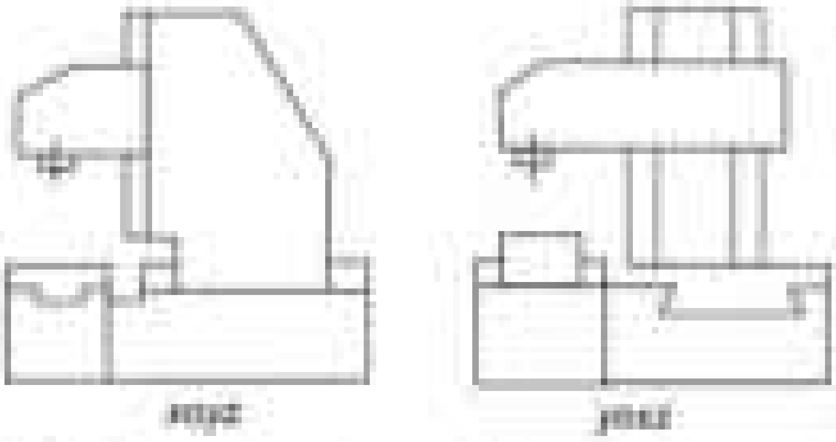


Fig. 1.64 Layout versions which satisfy all constraints

Review Questions

1.1 Determine the rpm of a lathe spindle if a workpiece of diameter 100 mm is to be turned at a cutting speed of 88 m/min.

Machine Tool Drives

- Machine Tools generally draw power from a single source such as electric motor or Engine.
- Engine is used to convert thermal power into electrical energy by using a generator.
- Speed of motor decreases marginally under load ,it is considered constant.
- Maximum torque depends upon the motor power rating (KW/HP) and rotary speed (RPM)

satinder.eme@gmail.com(91-9417365845)

Machine Tool Drives

•Rotary Drive :- For rotary cutting, it is necessary to compute revolutions per minute (R.P.M.) of the machine spindle

• **Mechanical Drives**

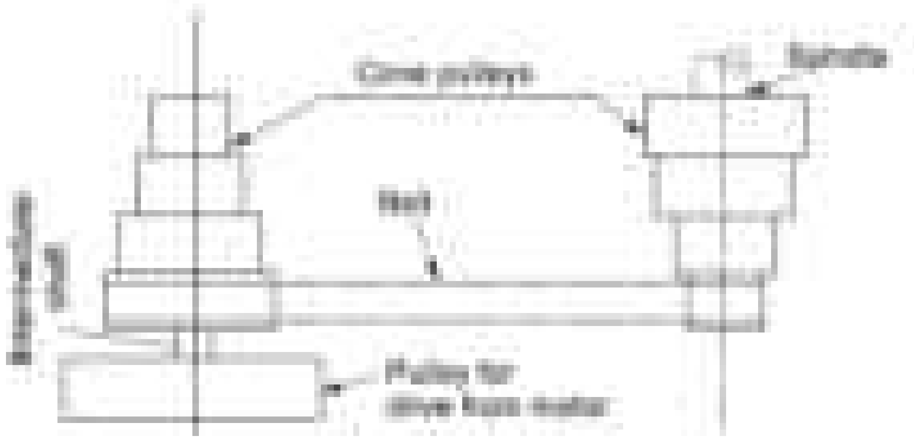
These can be broadly classified as:

1. **Positive drives: In these, the driver and the driven elements, mesh with each other** (gears), or with the power transmitting elements (chains).

2. **Frictional drives: Belts and clutches rely on friction for power transmission. There is** always a possibility of slip, under high speed (above 30 m/sec in belts), or due to overload. Timer belts with a toothed profile on the inside, are used for light load. They provide almost slip-free transmission. **Positive drives are suitable for low-speed (below 6 m/sec), high-torque applications**, while **frictional drives are more convenient and economical for high-speed (above 15 m/sec), low torque applications**.

CHANGING SPEED

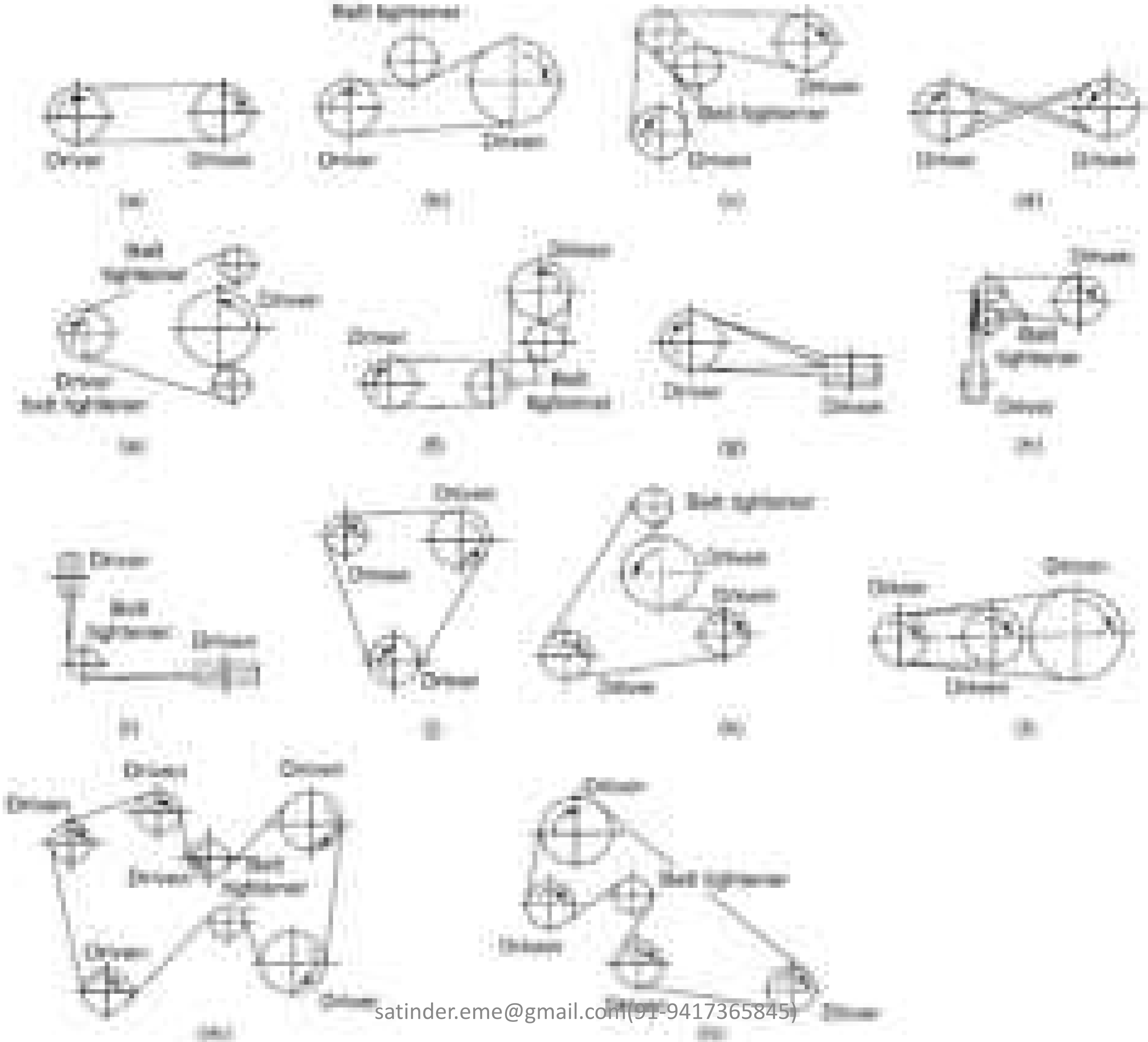
Slow Changeover: General purpose machines are used for a wide variety of workpiece sizes and materials. Higher cost, entailed in quick change of speed is not warranted. Belt-driven cone pulleys or change gears are quite adequate for speed change. Cone pulleys are combined, composite pulleys with 2 to 4 steps. **Figure (a) shows a cone** pulley with 4 steps. Any of the available four speeds can be obtained, by shifting the belt to the required step in a minute or two. Belt drives are more convenient for higher speeds.



Cone pulley, belt drive

satinder.eme@gmail.com(91-9417365845)

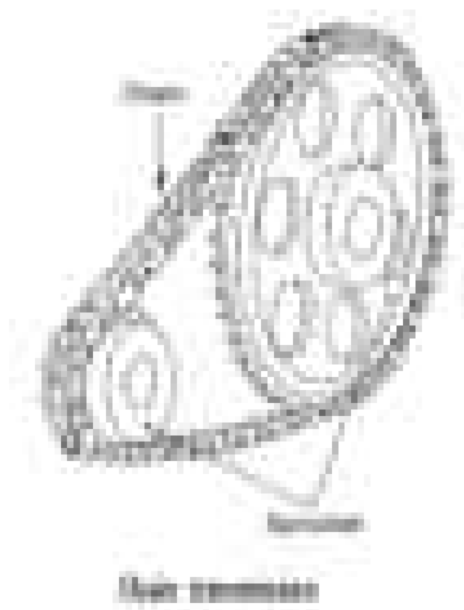
Flat belts are more convenient for drives between non-parallel shafts [Fig. g, h, i]. The driven shaft can be rotated in a direction, opposite to the rotation of the driving shaft, by crossing flat belt or using 2 idlers **[Fig. d, e, f]. The thinner, rectangular section of flat** belts also enables usage of smaller pullies. The width of flat belts varies from 20 to 500 and the thickness from 3 to 13.5. The pullies are crowned, made bigger at the center of the width, by making them convex **[Fig. a] or conical, to facilitate centralizing the flat belts on** ungrooved pullies.



Driver
Driven
(a)
Belt tightener
Driver
Driven
(b)
Driven
Belt tightener
Driven
(c)
Driver
Driven
(d)
Belt tightener
Driven
Driver
Belt tightener
(e)
Driven
Driver
Belt tightener
(f)
Driver
Driven
(g)
Driven
Belt tightener
Driver
(h)
Driver
Belt tightener
Driven
(i)
Driven
Driven
Driver
(j)
Belt tightener
Driven
Driven
Driver
(k)
Driven
Driver
Driven
(l)
Driven
Driven
Driven
Driven
Belt tightener
Driven
Driver
(m)
Driven
Belt tightener
Driven
Driver
(n)

CHAIN DRIVE

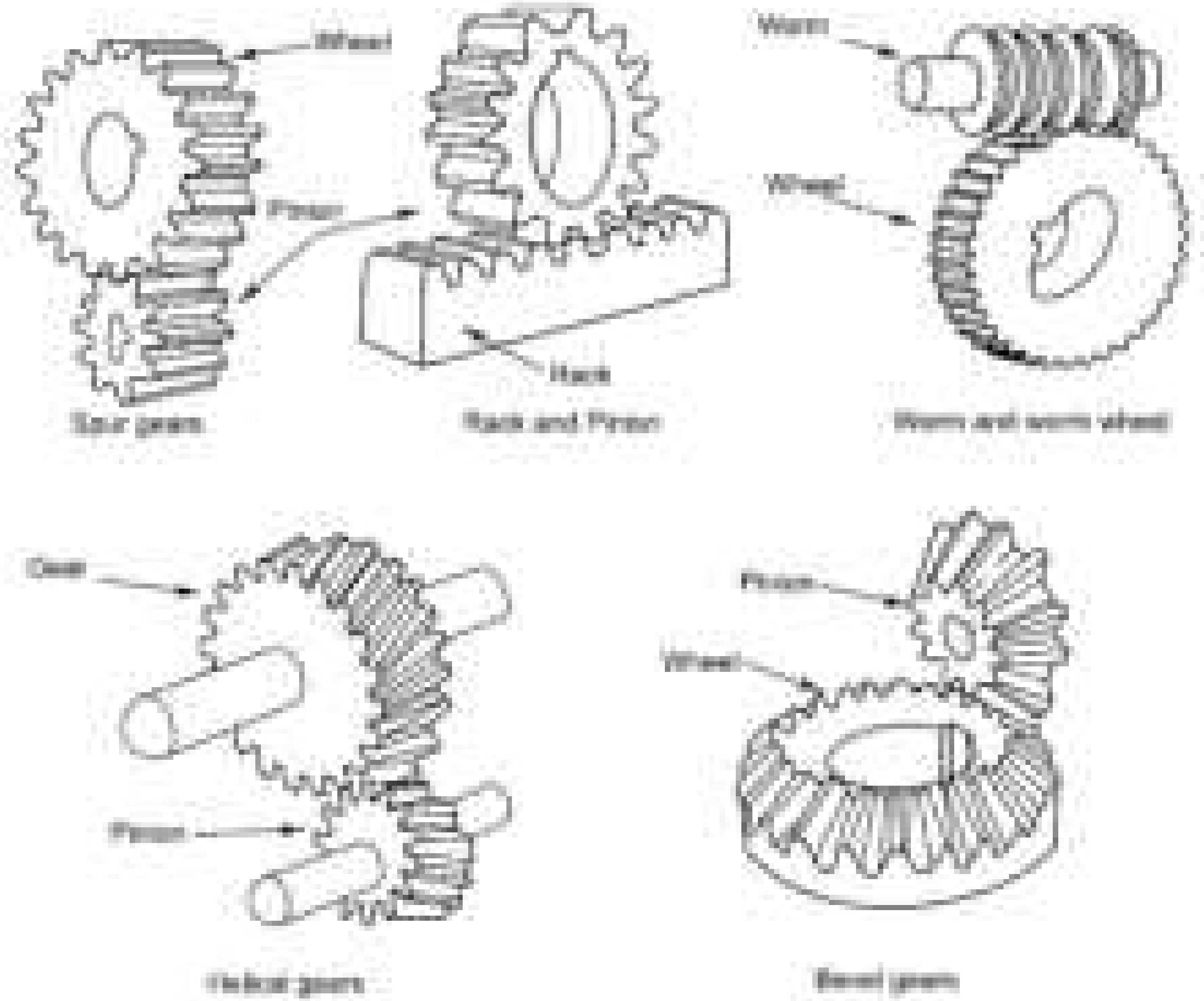
Instead of belts, we can use a more compact and positive roller chain and sprockets if the linear speed is less than 12 m/sec (at the most 20 m/sec) and transmission ratio is less than 7. The flexibility of the chain makes the drive shock absorbent. The number of teeth on the smaller sprocket should not be less than 17 [preferably 21].



GEARS

Some mass production machines like autos, produce similar work pieces for long runs, lasting many shifts. A little extra setting time, spent on changing the pullies, sprockets, or gear does not make much difference in the overall economy. Even in machines with very long, cutting times or slower operations, such as thread-cutting, the machine setting time is only a small fraction of the total running time. Under such circumstances, slow, manual replacement of change gears is quite satisfactory.

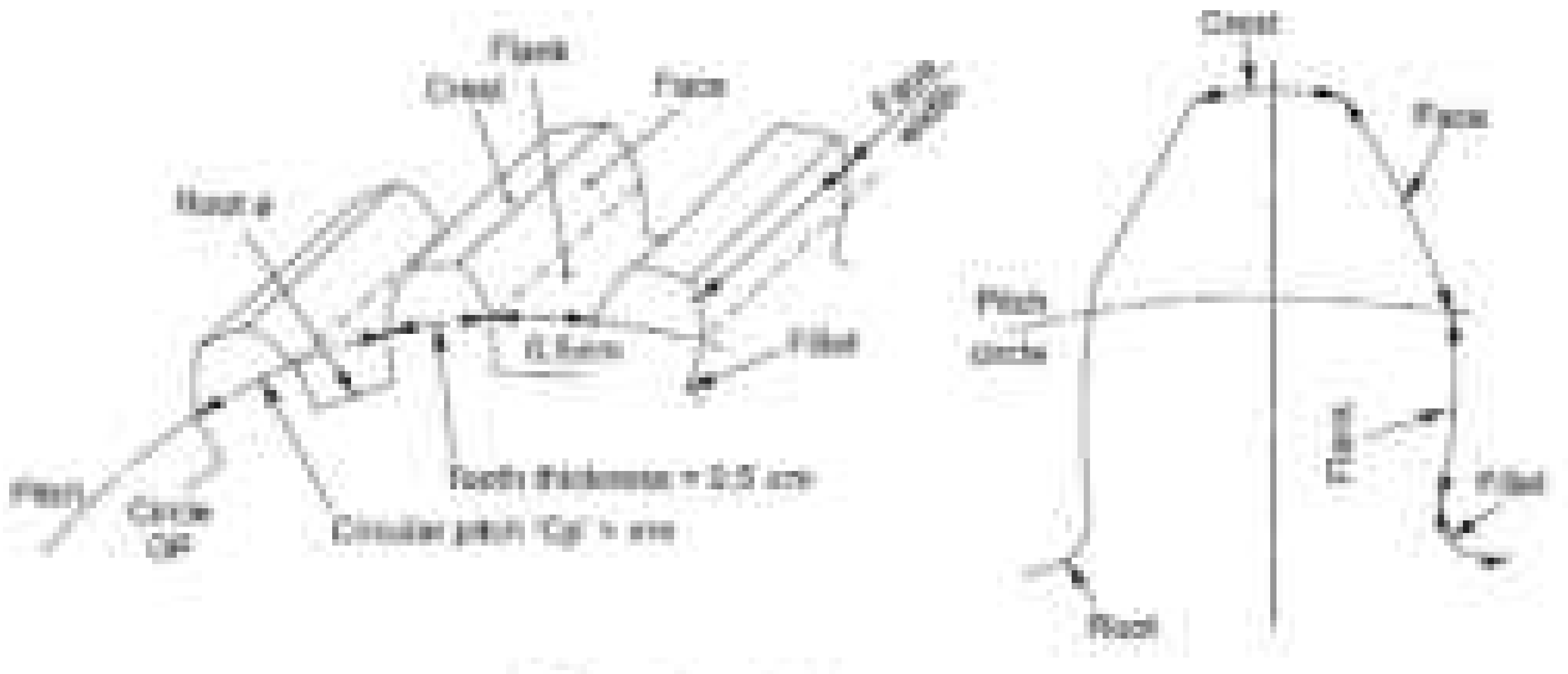
The teeth of the gears engage and intermesh with the teeth of the mating gears. Spur gears have teeth on the cylindrical portion. When the teeth are parallel to the axis of rotation, the gears are called straight spur gears or simply spur gears. Making the teeth twisted with the gear axis, helical **increases the load capacity and promotes a smooth, and gradual** engagement. Straight and helical spur gears are used for transmission between parallel shafts.

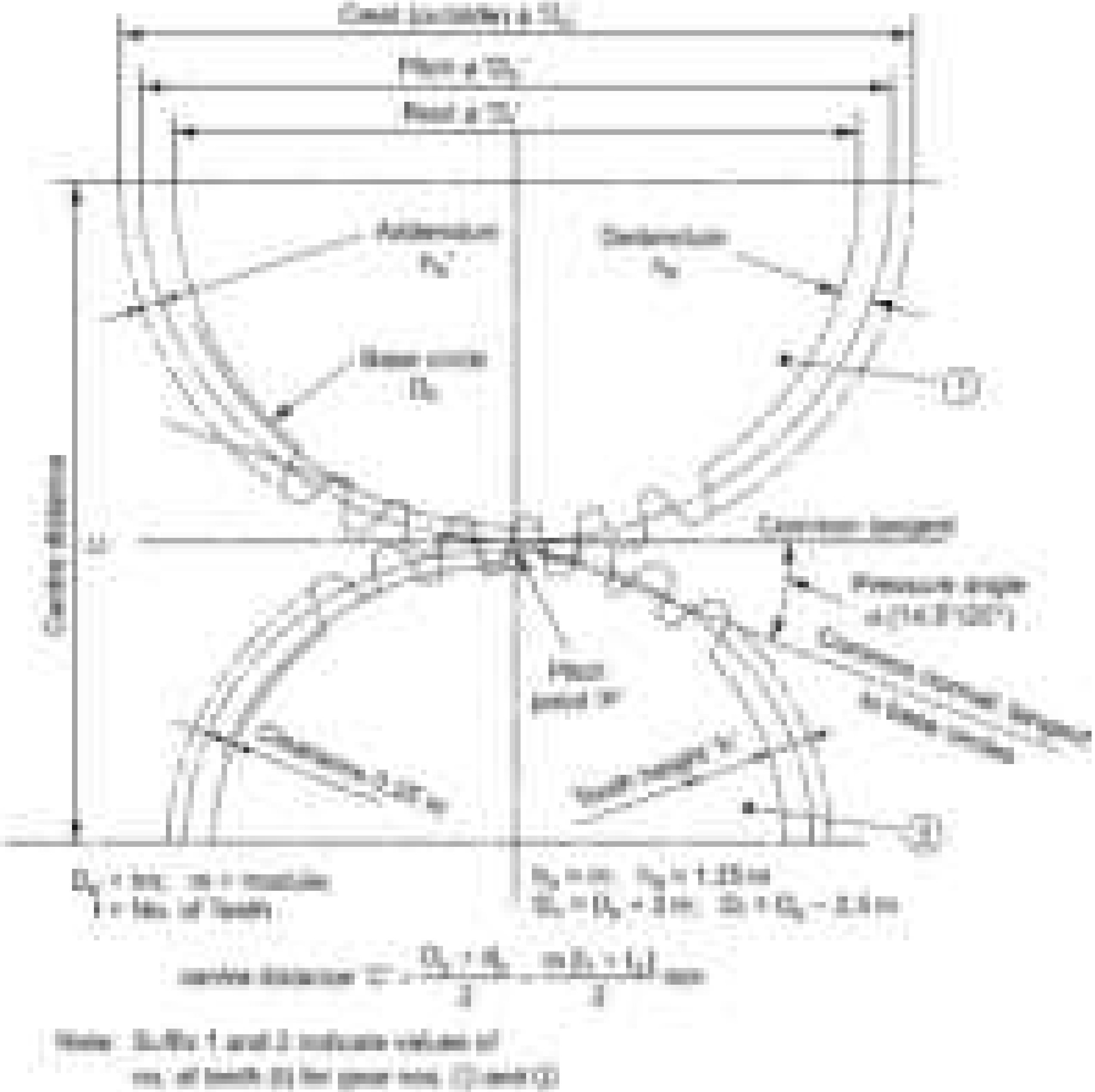


Types of gear drives

GEAR TOOTH TERMINOLOGY

The size of the tooth is specified by module (m). The other important parameter in spur gears is the pitch diameter ,**the diameter where the meshing gear pitch touches** tangentially, and where the tooth thickness is equal to the gap between adjacent teeth. In racks, the tooth thickness and the gap are equal at the pitch line.





d_p = pinion pitch ϕ (mm); D_p = gear pitch ϕ (mm)
t = No. of teeth; m = module (mm)
For standard, unmodified tooth profile:

Tooth height above pitch ϕ = Addendum = $h_a = m$

Tooth height (depth) below pitch ϕ = Dedendum = h_d = 1.25 m

Tooth height = $h_a + h_d = h$ = 2.25 m

Tip clearance = $h_d - h_a$ = 0.25 m

Circular pitch = $P = \pi$ m

Circular tooth thickness (or gap) = 0.5 π m

$\therefore$ Outside or tip ϕ of gear = $d_o = m\,(t + 2)$

Root ϕ of gear = $d_r = m\,(t - 2.5)$

2. BEARING SELECTION PROCEDURE

The number of applications for rolling bearings is almost countless and the operating conditions and environments also vary greatly. In addition, the diversity of operating conditions and bearing requirements continue to grow with the rapid advancement of technology. Therefore, it is necessary to study bearings carefully from many angles to select the best one from the thousands of types and sizes available.

Usually, a bearing type is provisionally chosen considering the operating conditions, mounting arrangement, ease of mounting in the machine, allowable space, cost, availability, and other factors.

Then the size of the bearing is chosen to satisfy the desired life requirement. When doing this, in addition to fatigue life, it is necessary to consider grease life, noise and vibration, wear, and other factors.

There is no fixed procedure for selecting bearings. It is good to investigate experience with similar applications and studies relevant to any special requirements for your specific application. When selecting bearings for new machines, unusual operating conditions, or harsh environments, please consult with NSK.

The following diagram (Fig.2.1) shows an example of the bearing selection procedure.

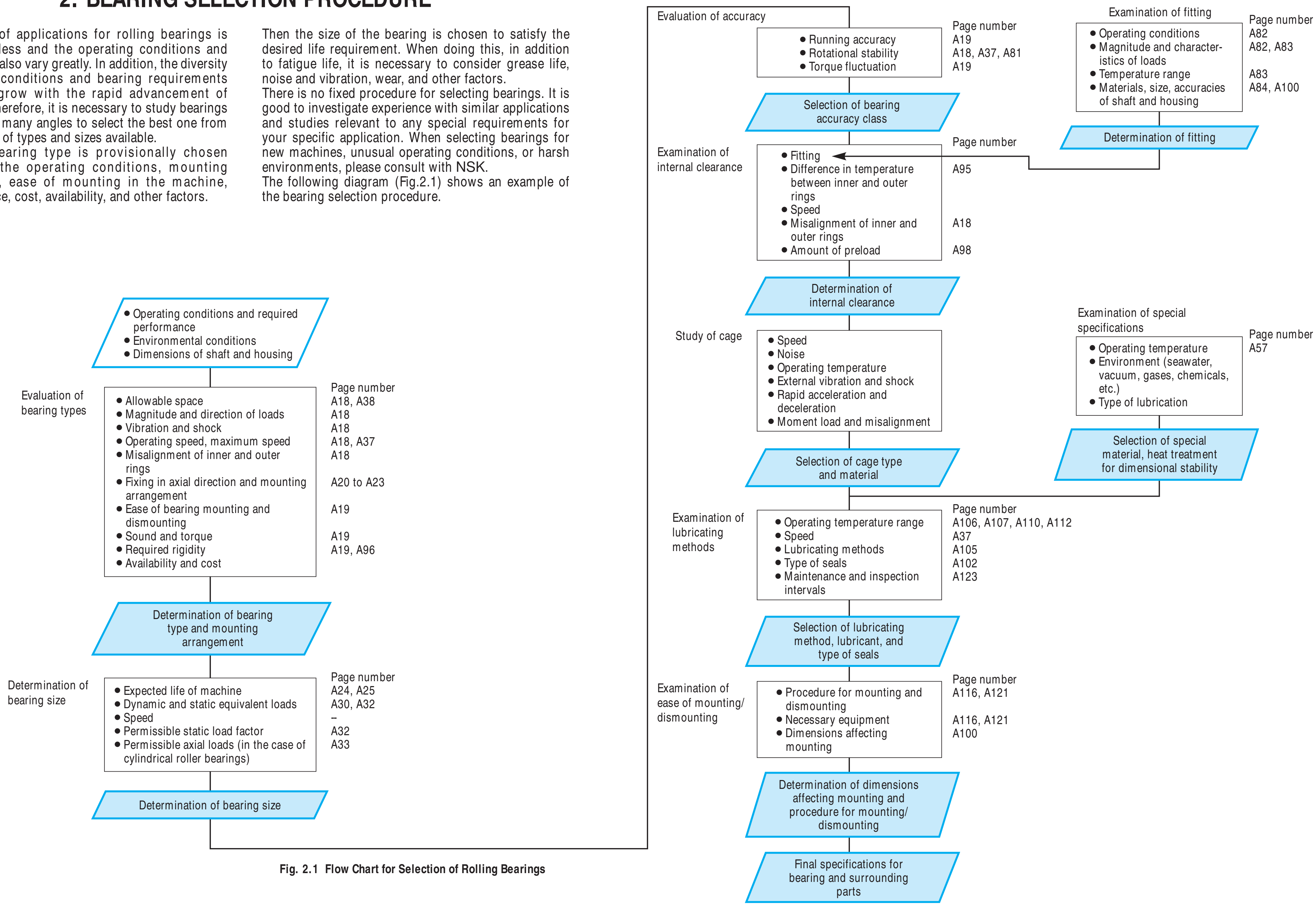


Fig. 2.1 Flow Chart for Selection of Rolling Bearings