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.

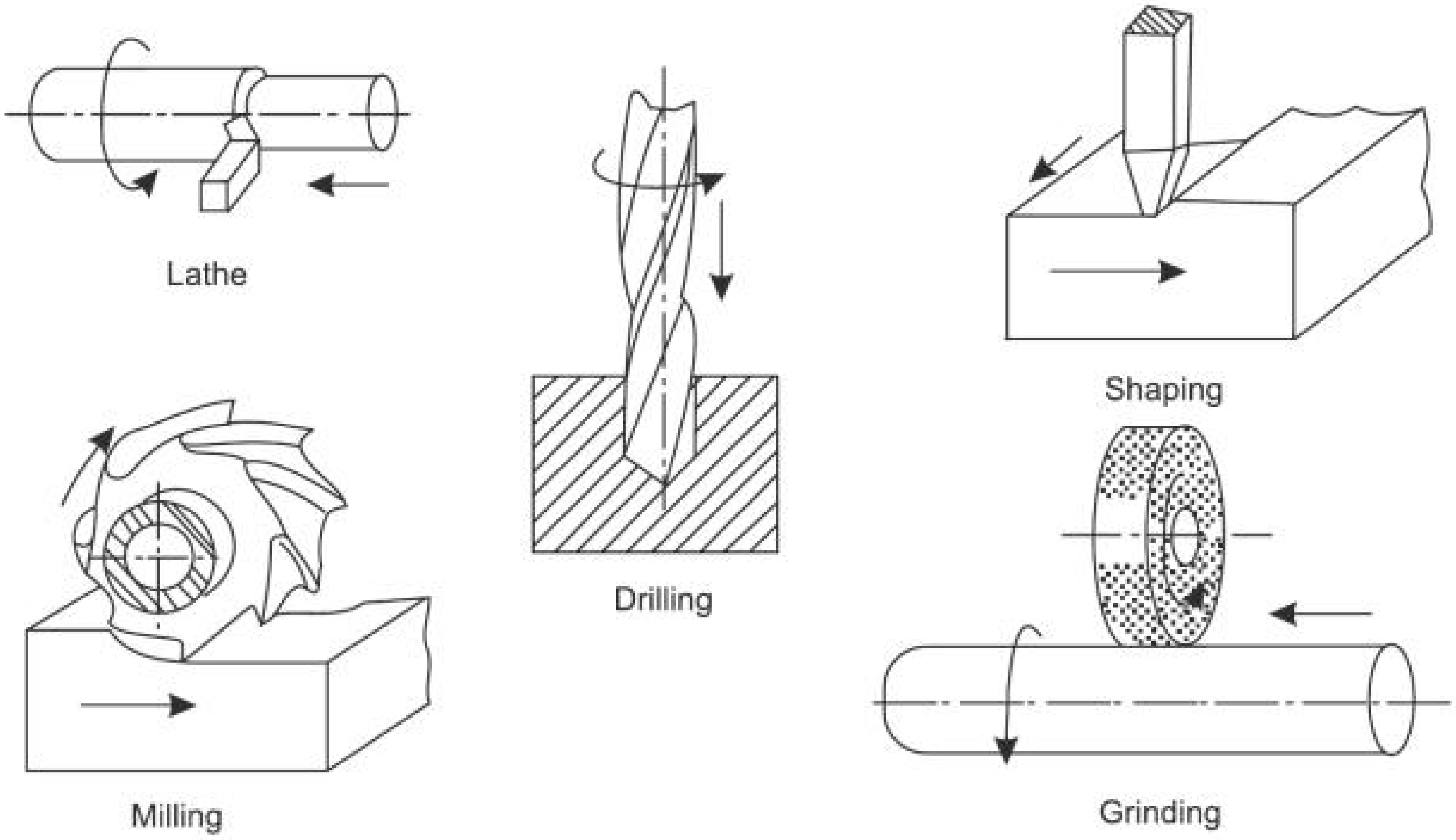


Fig. 1.1 *Working motions for some machine tools*

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. In 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 $\Delta 1$ for safe entry of tool (on commencement of machining) and over travel of $\Delta 2$ for safe exit of tool (on completion of the machining cut). Generally, $\Delta 1$ and $\Delta 2$ 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
$\Delta 1$ = approach; generally equal to 2–3 mm
$\Delta 2$ = over travel; 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 $\Delta 3 = 0$

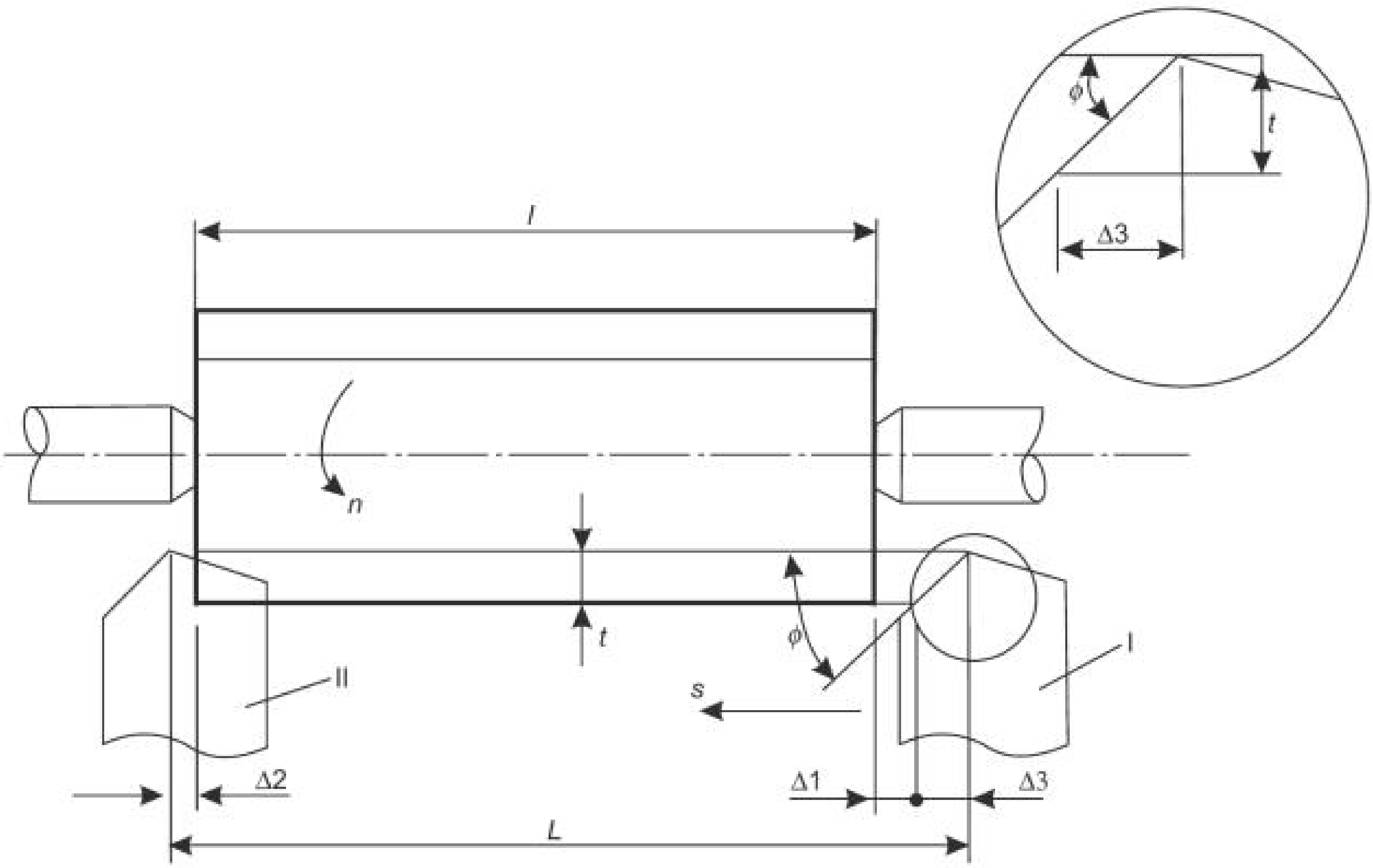


Fig. 1.2 *Turning operation on workpiece supported between centres*

(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

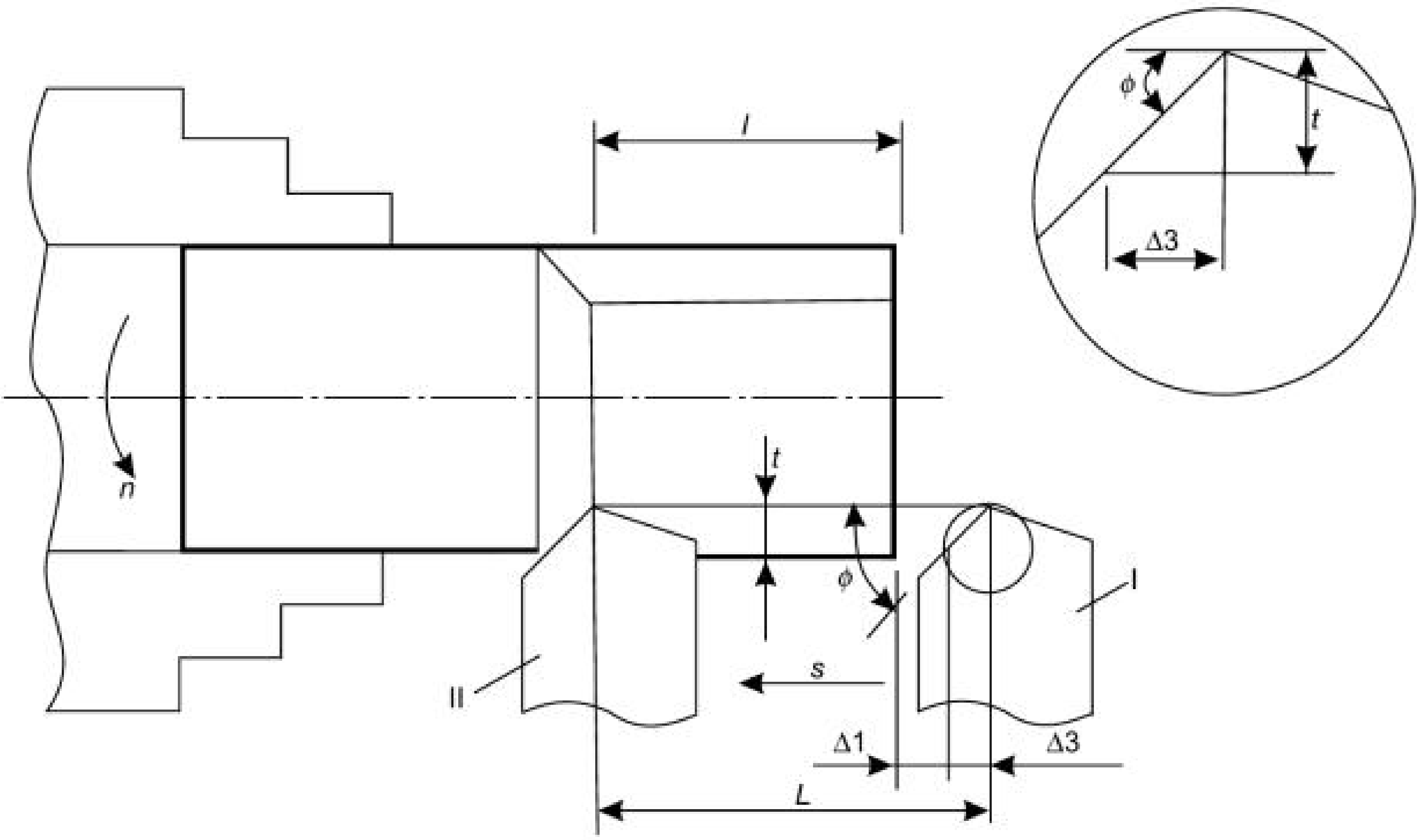


Fig. 1.3 *Turning operation on workpiece clamped in chuck*

(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.

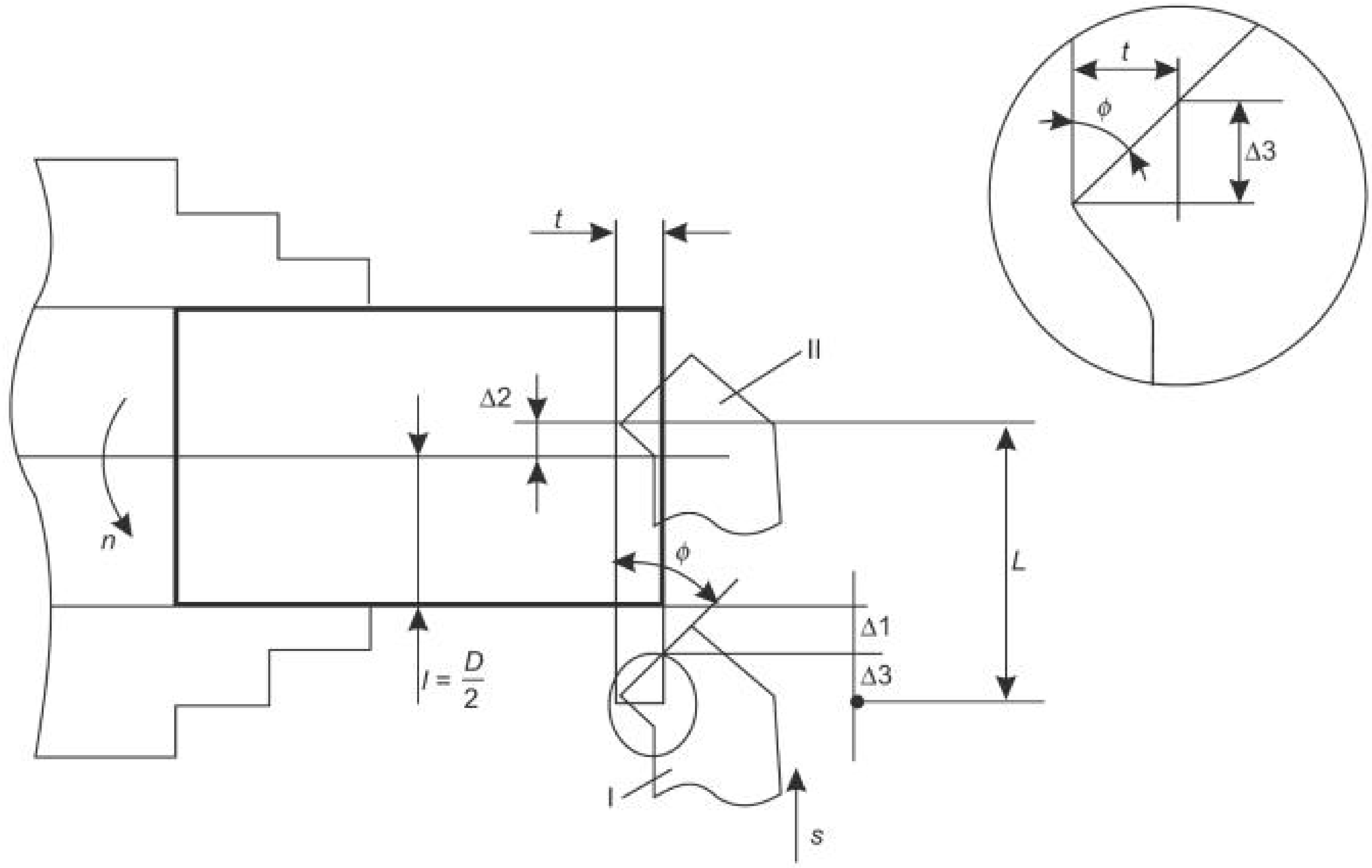


Fig. 1.4 *Facing operation*

(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 $\Delta 3 = 0$

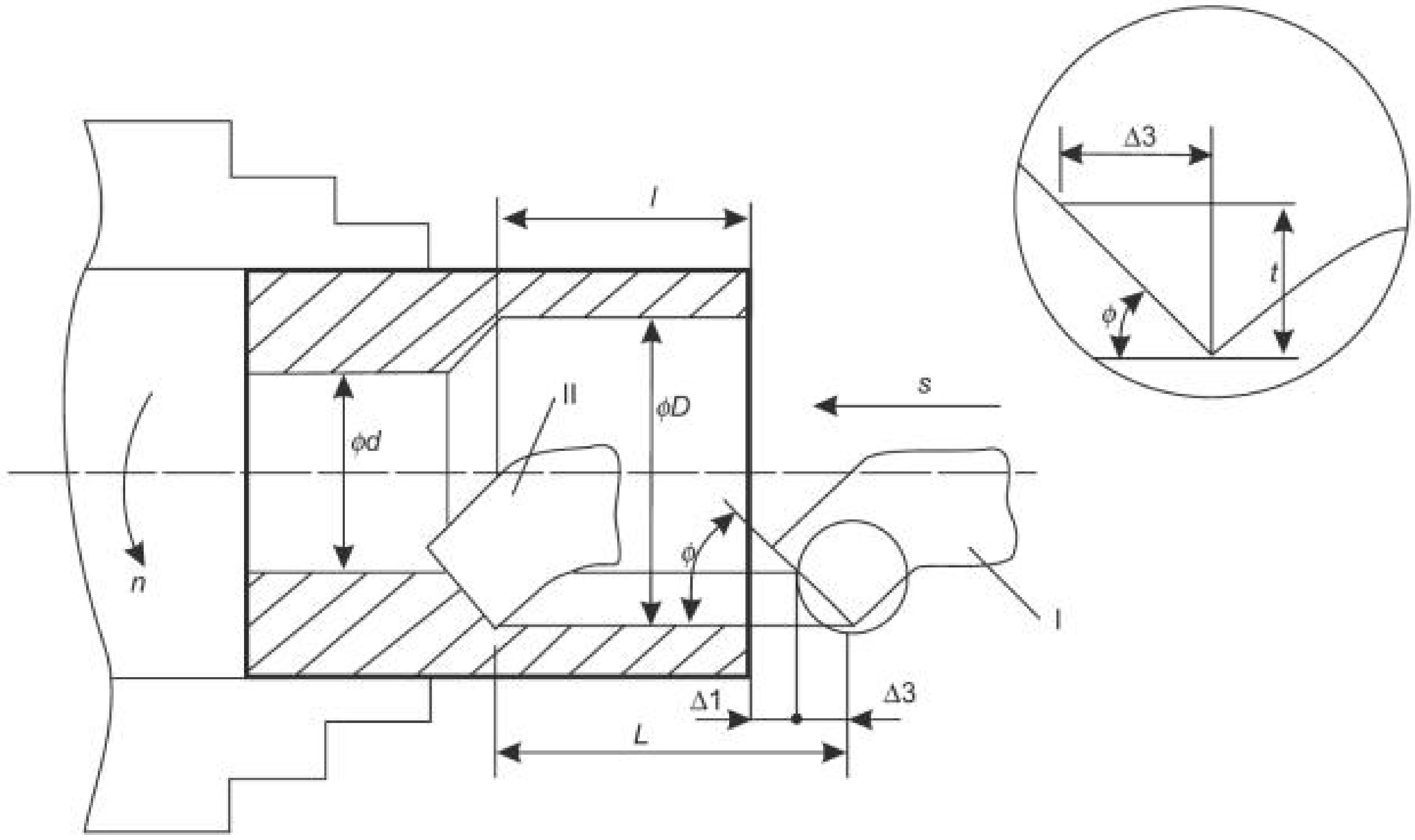


Fig. 1.5 *Boring operation in partial length of workpiece*

(e) *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

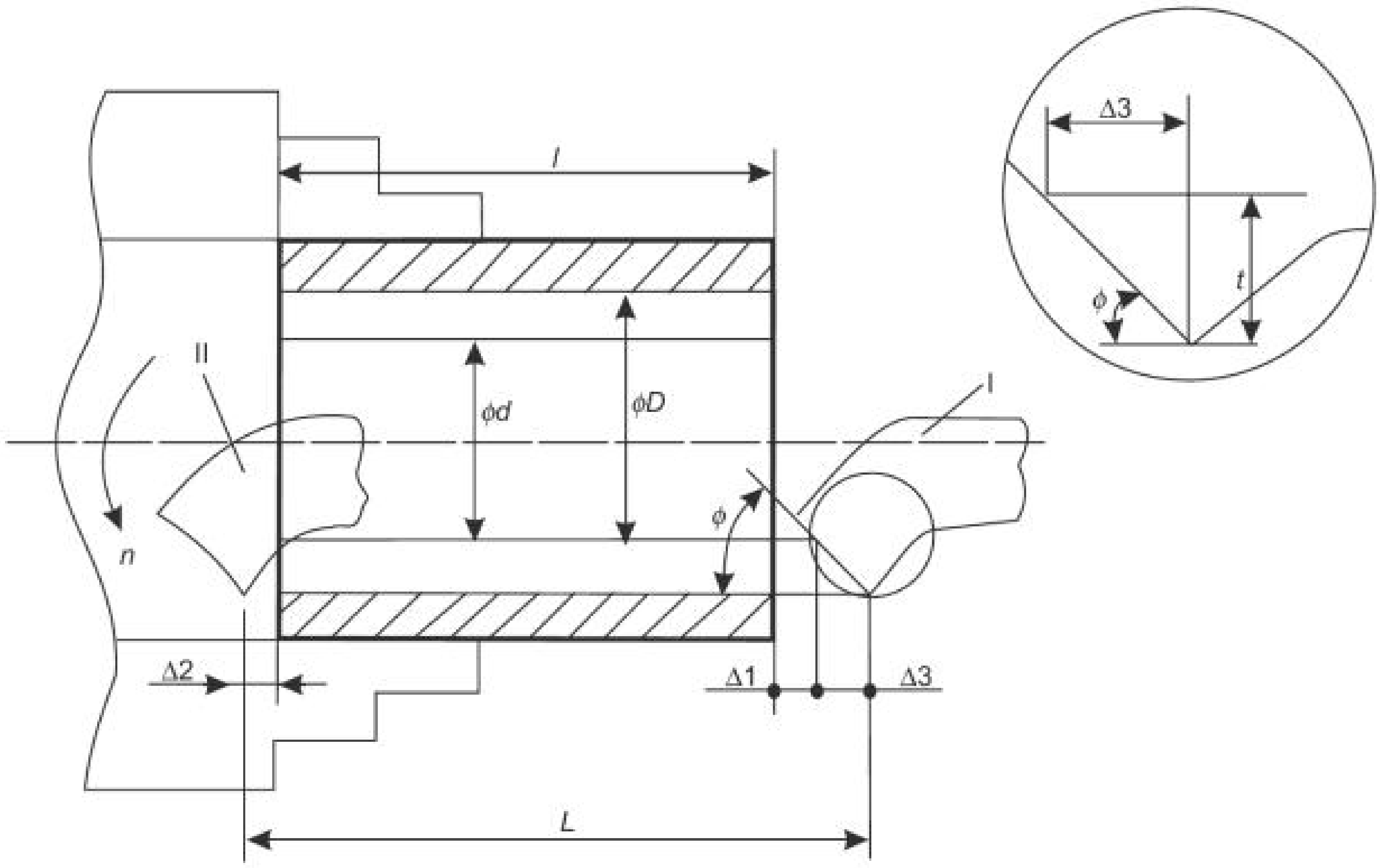


Fig. 1.6 *Boring operation in full length of workpiece*

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:

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

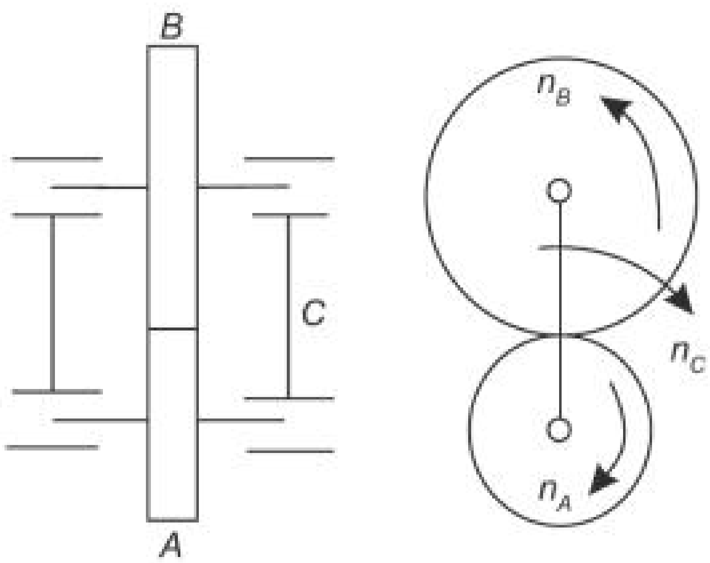


Fig. 1.50 *Differential mechanism using spur or helical gears*

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

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

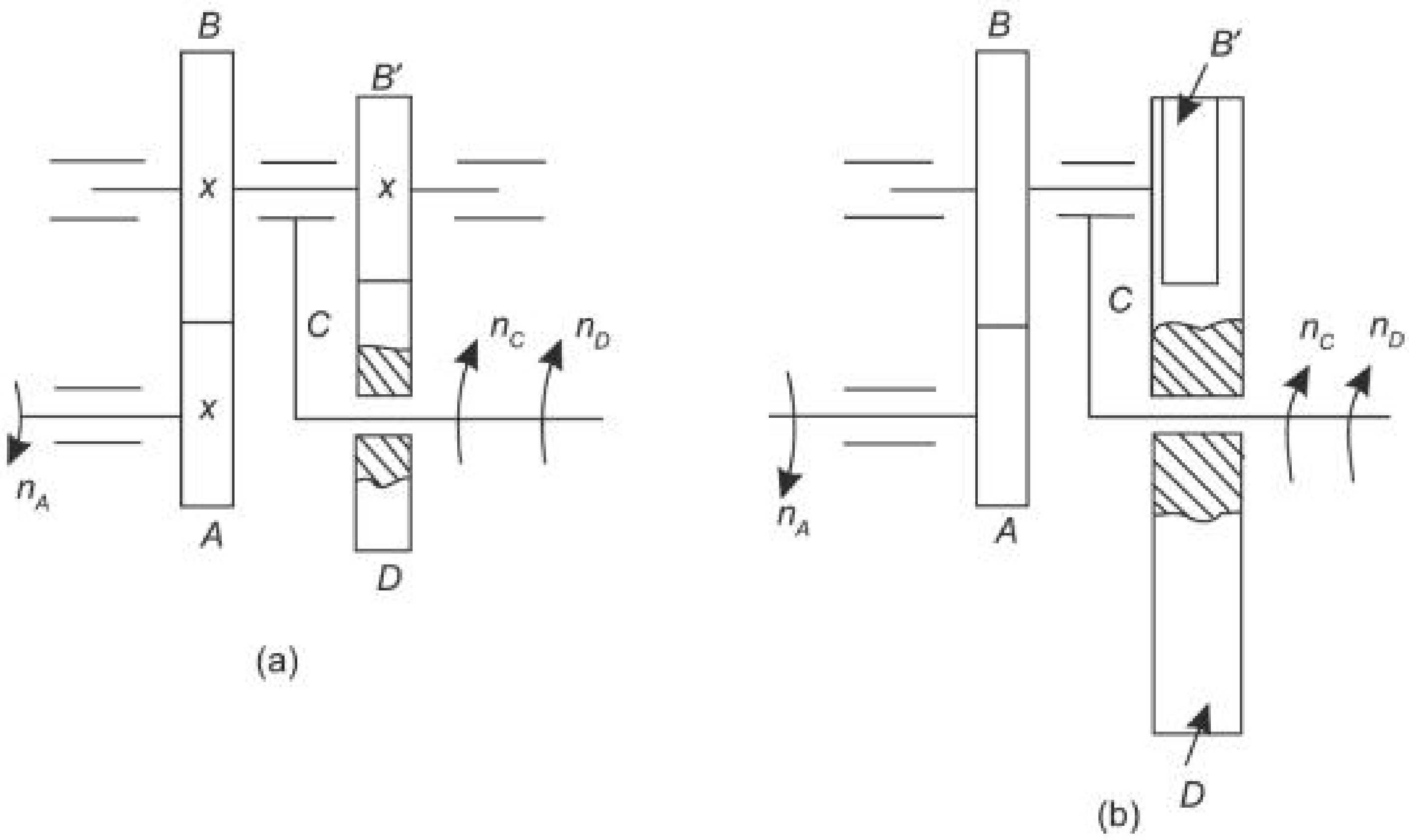


Fig. 1.51 *Differential mechanisms using double-cluster planetary gears*

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.

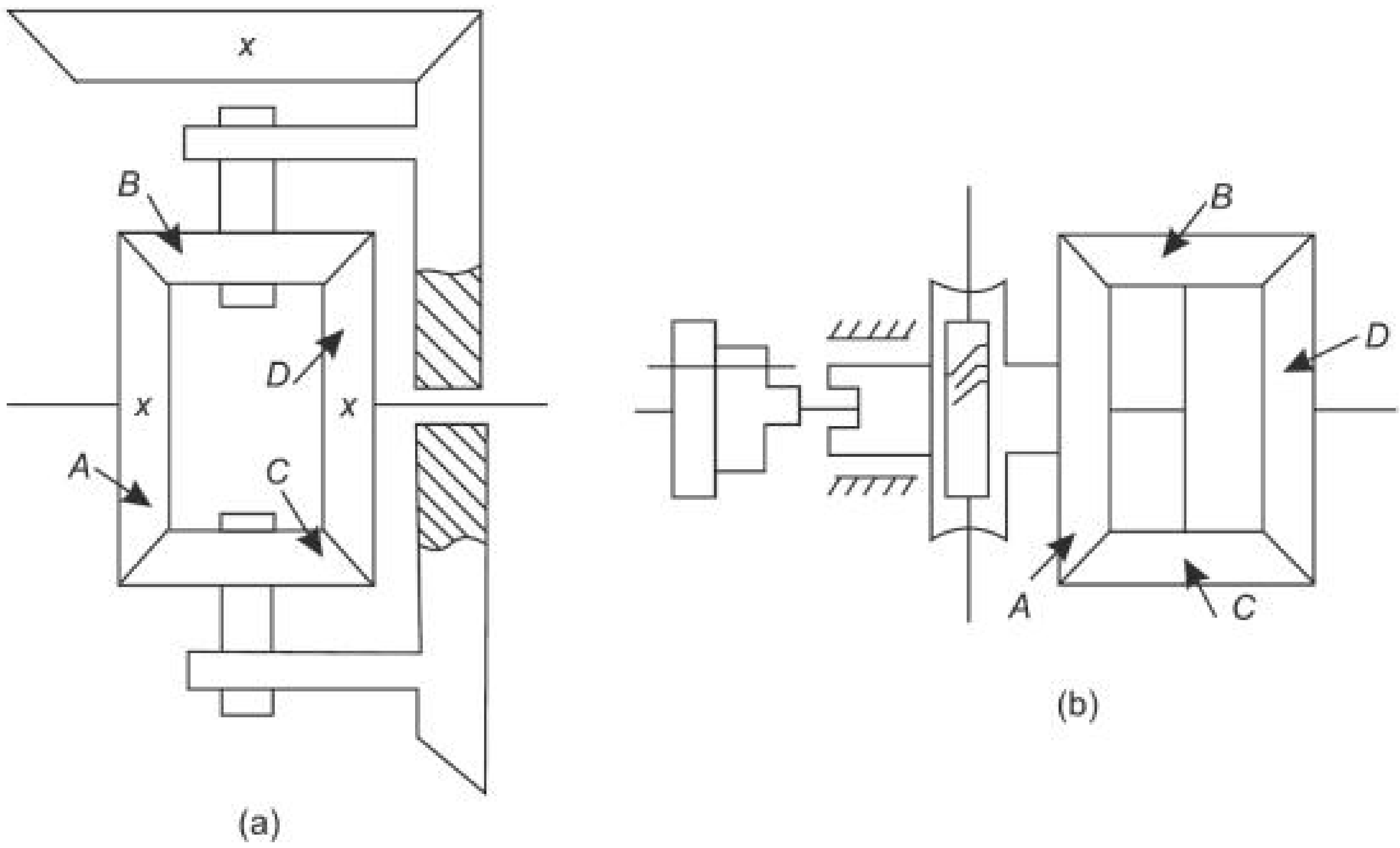


Fig. 1.52 *Differential mechanisms: (a) used in automobiles (b) used in machine tools*

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.

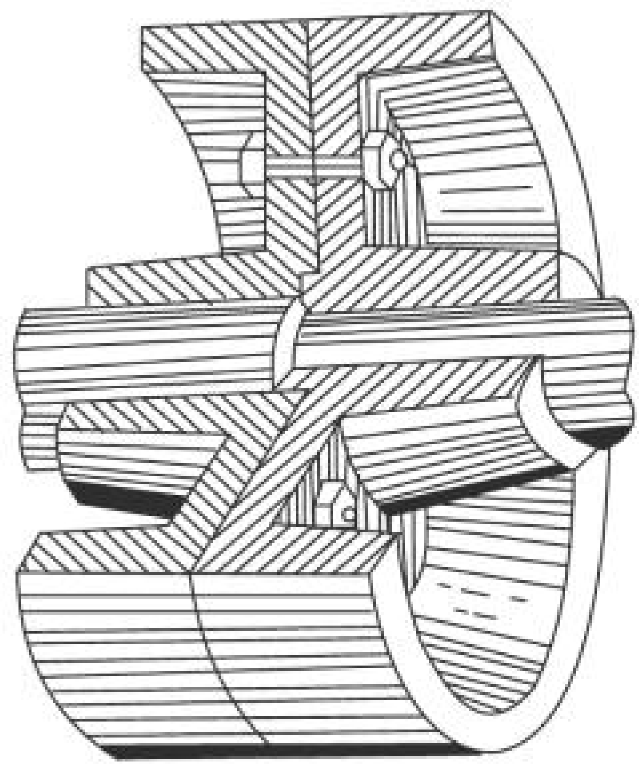
1.5.6 Couplings and Clutches

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*.

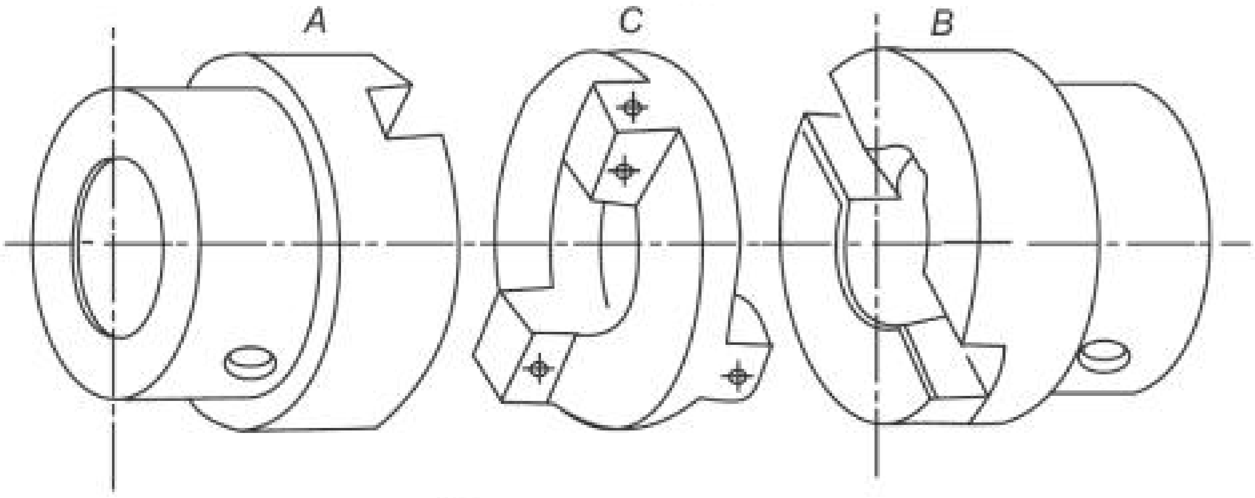
Couplings Couplings are of two types:

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.



(a)



(b)

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.

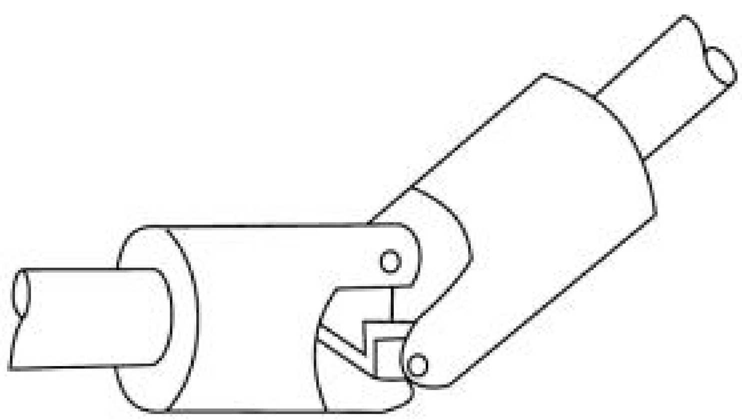


Fig. 1.54 *Elastic coupling*

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.

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.

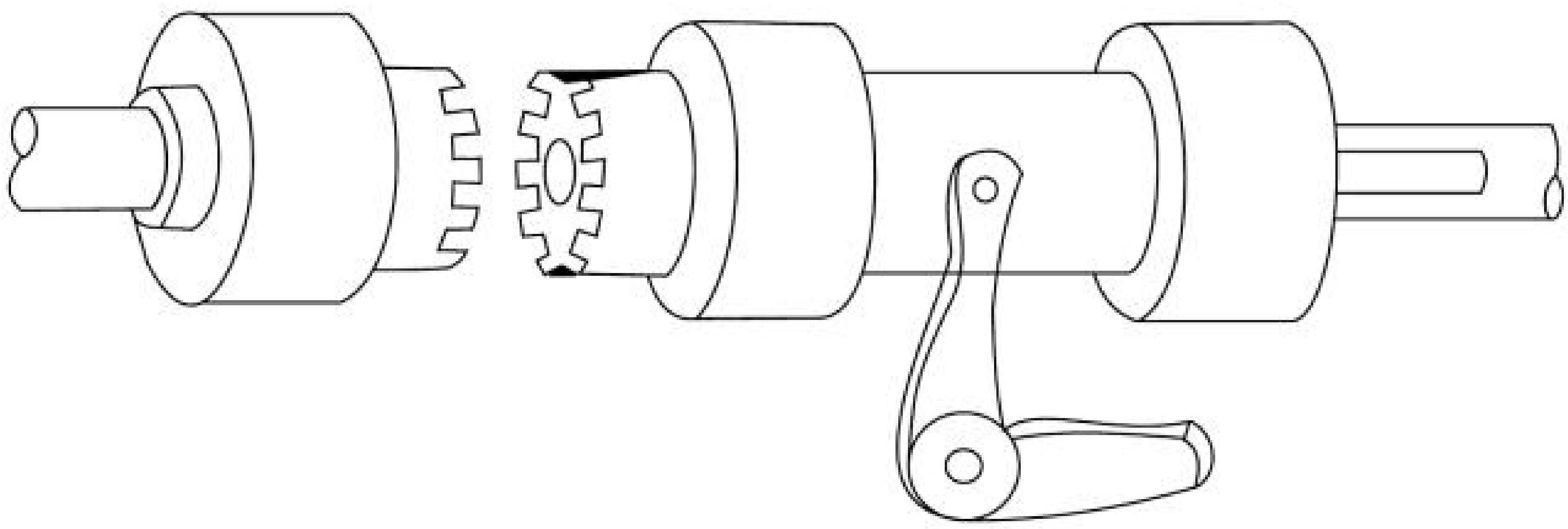


Fig. 1.55 *Jaw clutch*

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.

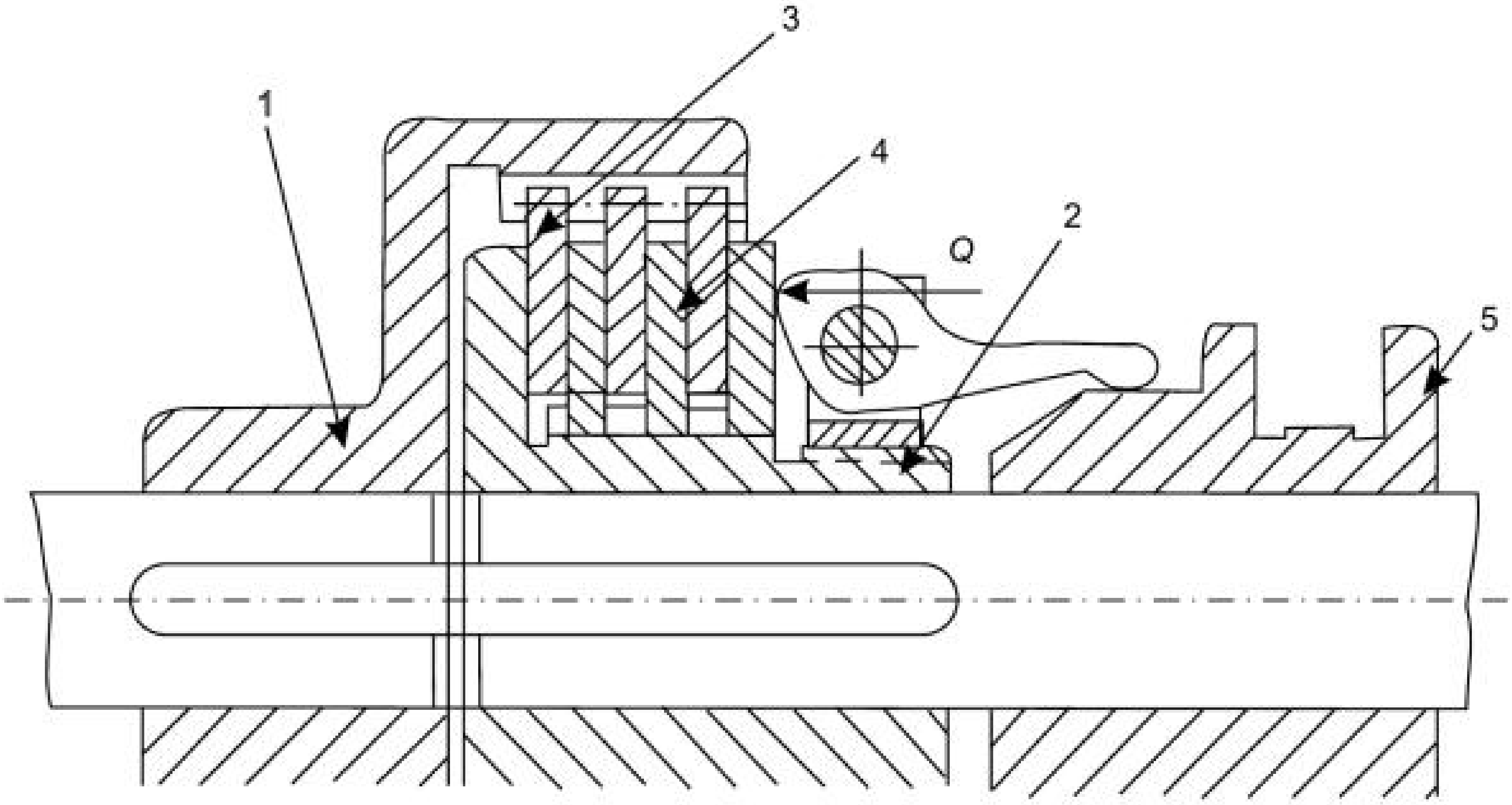


Fig. 1.56 *Multiple-disc friction clutch*

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

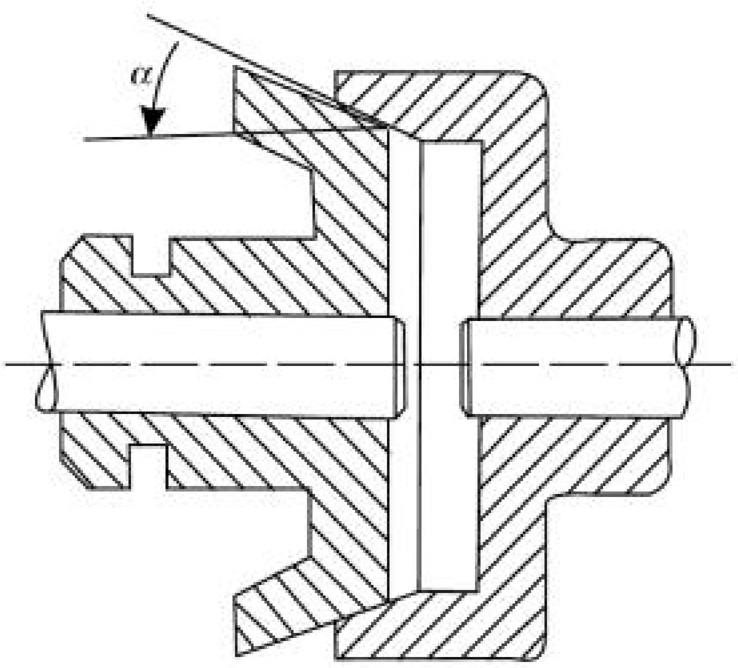


Fig. 1.57 *Cone-type friction clutch*

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 not 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

$$E_n = \alpha E_p$$

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)

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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**

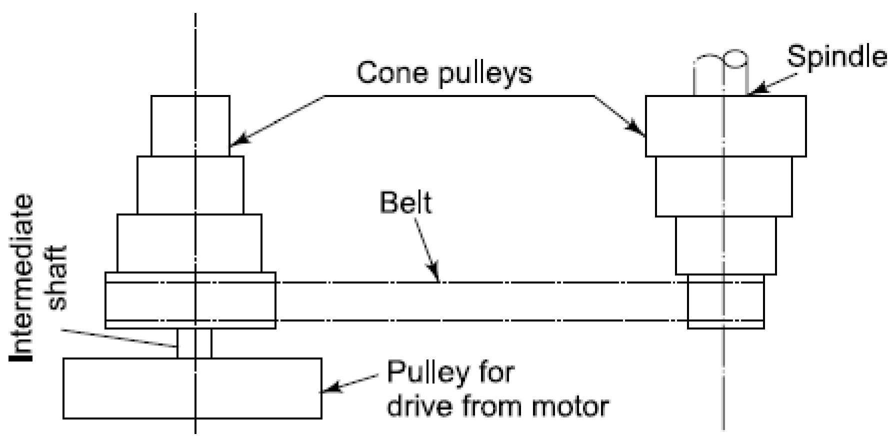
These can be broadly classified as:

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

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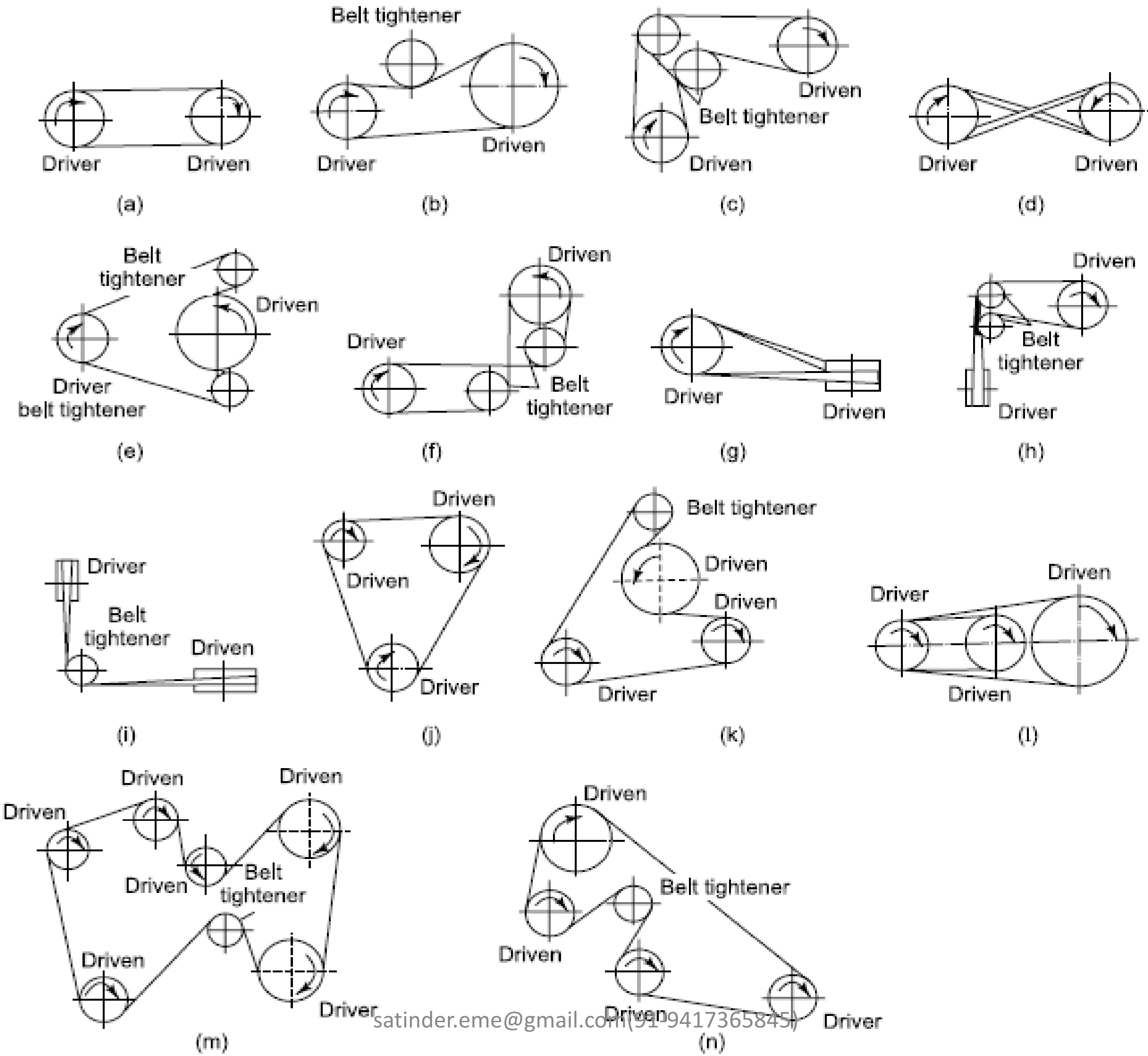
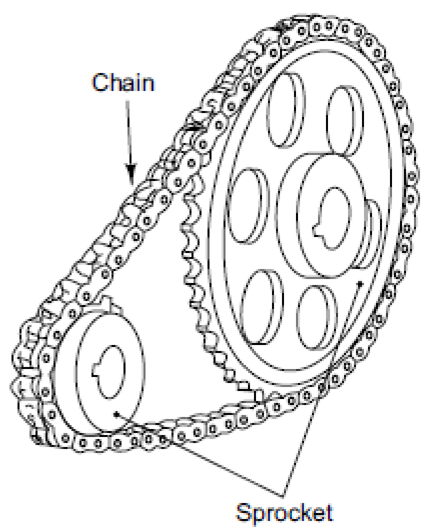
Flat belts are more convenient for drives between non-parallel shafts [Fig. g, h, i]. The driven shaft can be rotated in a direction, opposite to the rotation of the driving shaft, by crossing flat belt or using 2 idlers **[Fig. d, e, f]. The thinner, rectangular section of flat** belts also enables 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)
Driver
Driven
Driven
(l)
Driven
Driven
Driven
Driven
Belt
tightener
Driven
Driver
(m)
Driven
Belt tightener
Driven
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].

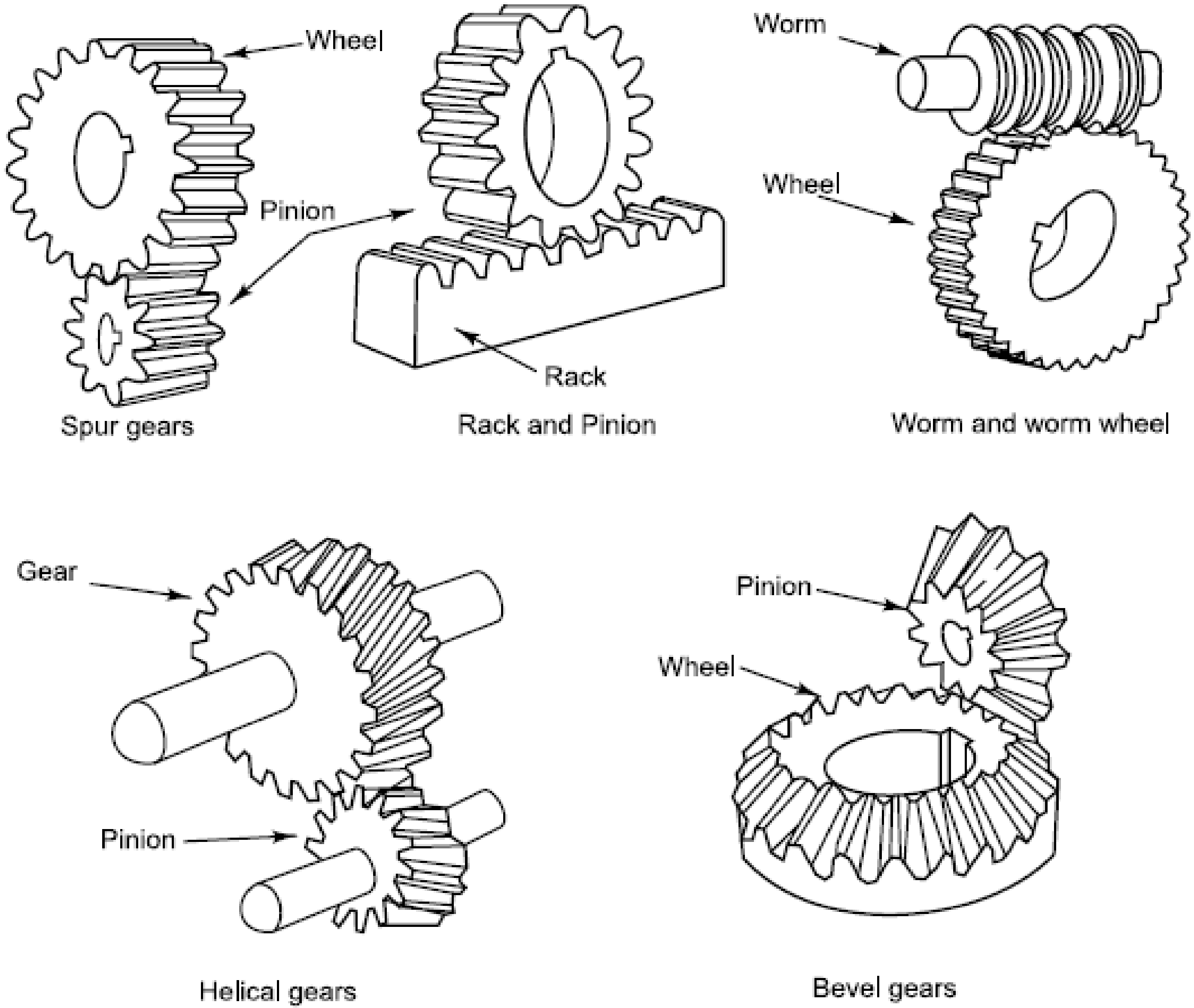


Chain transmission

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.

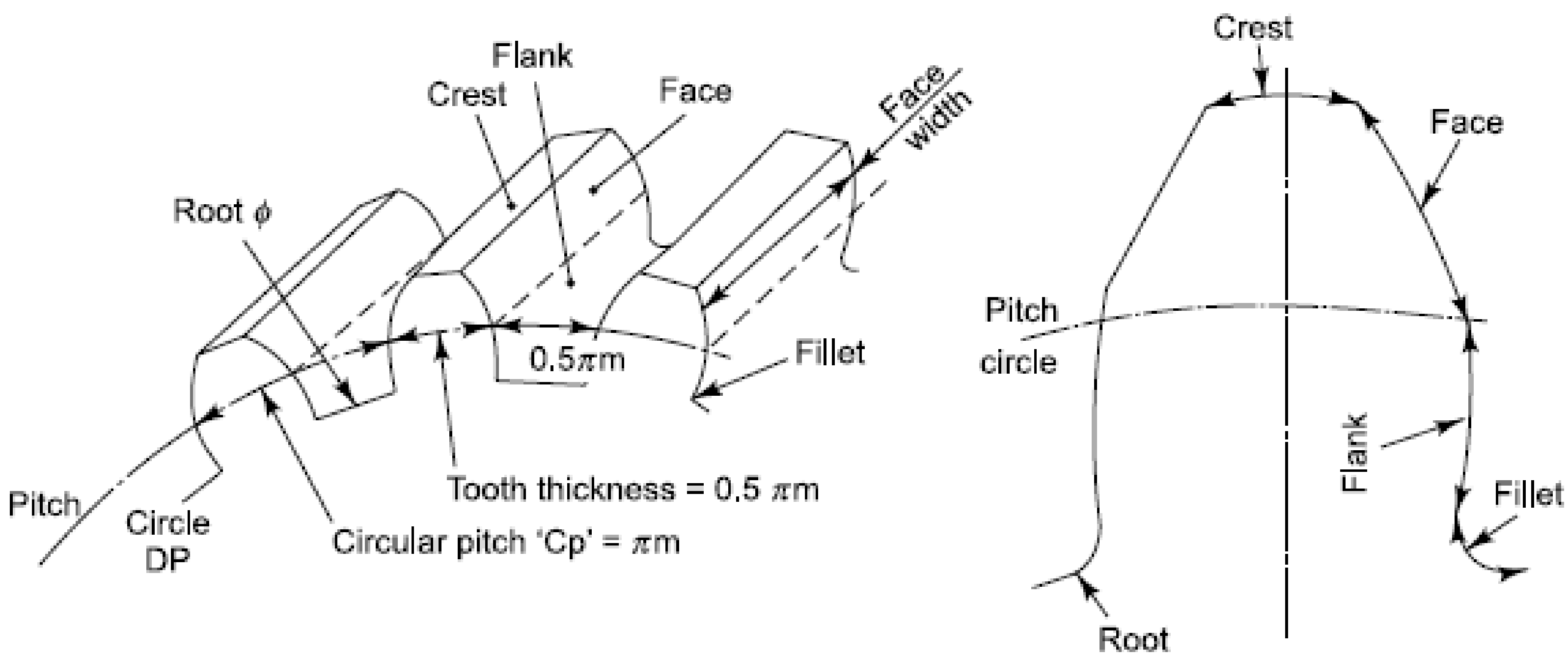
The teeth of the gears engage and intermesh with the teeth of the mating gears. Spur gears have teeth on the cylindrical portion. When the teeth are parallel to the axis of rotation, the gears are called straight spur gears or simply spur gears. Making the teeth twisted with the gear axis, helical **increases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



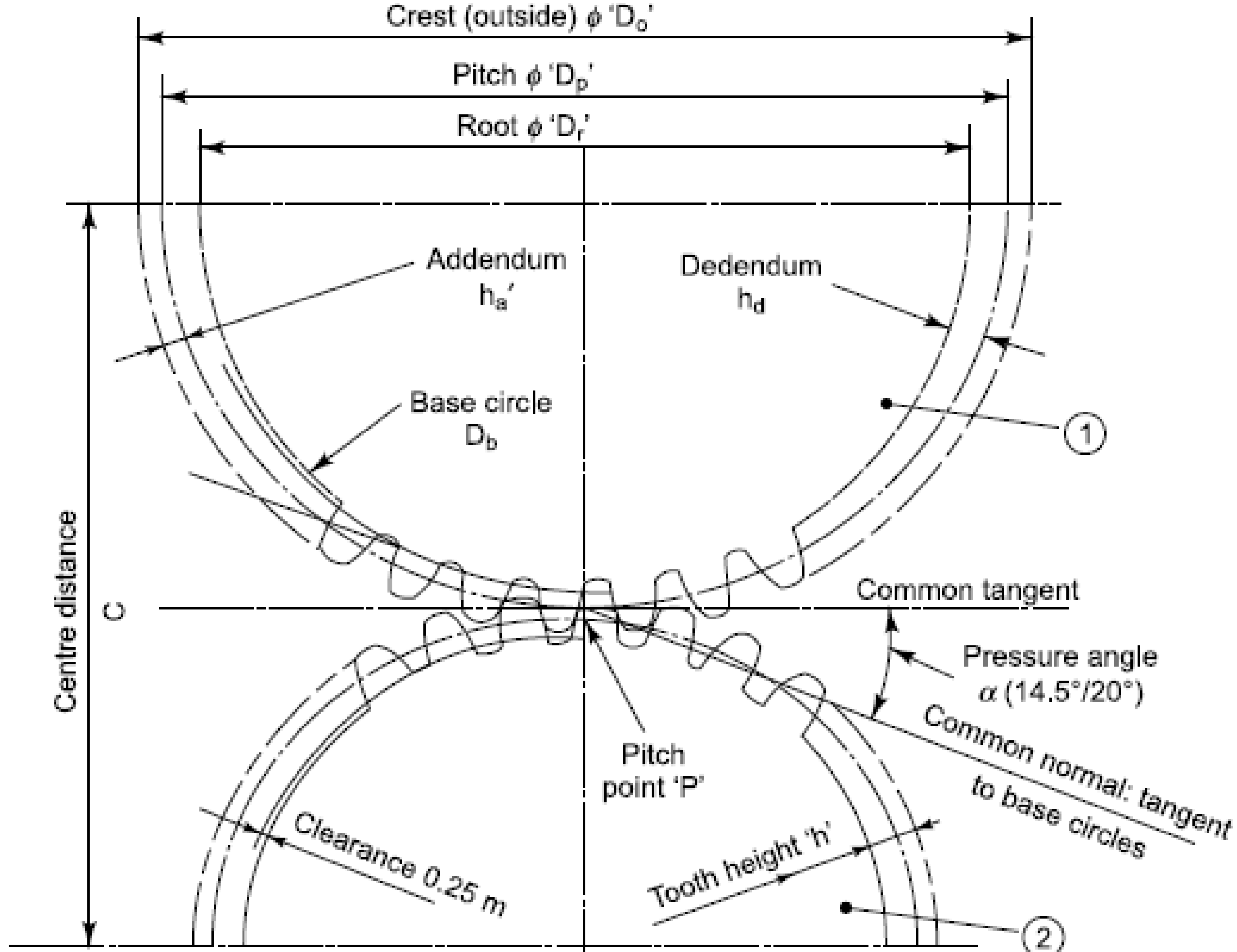
Types of gear drives

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.



Gear tooth terminology



$D_p = tm$; m = module;
t = No. of Teeth

$h_a = m$; $h_d = 1.25\,m$
$D_o = D_p + 2\,m$; $D_r = D_p - 2.5\,m$

$$\text{centre distance 'C'} = \frac{D_p + d_p}{2} = \frac{m\,[t_1 + t_2]}{2}\ \text{mm}$$

Note: Suffix 1 and 2 indicate values of no. of teeth (t) for gear nos. 1 and 2

Important dimensions of mating gears

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) = $0.5\ \pi$ 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.

- Operating conditions and required performance
- Environmental conditions
- Dimensions of shaft and housing

Evaluation of bearing types

	Page number
Allowable space	A18, A38
Magnitude and direction of loads	A18
Vibration and shock	A18
Operating speed, maximum speed	A18, A37
Misalignment of inner and outer rings	A18
Fixing in axial direction and mounting arrangement	A20 to A23
Ease of bearing mounting and dismounting	A19
Sound and torque	A19
Required rigidity	A19, A96
Availability and cost	

Determination of bearing type and mounting arrangement

Determination of bearing size

	Page number
Expected life of machine	A24, A25
Dynamic and static equivalent loads	A30, A32
Speed	–
Permissible static load factor	A32
Permissible axial loads (in the case of cylindrical roller bearings)	A33

Determination of bearing size

Evaluation of accuracy

	Page number
Running accuracy	A19
Rotational stability	A18, A37, A81
Torque fluctuation	A19

Selection of bearing accuracy class

Examination of fitting

	Page number
Operating conditions	A82
Magnitude and characteristics of loads	A82, A83
Temperature range	A83
Materials, size, accuracies of shaft and housing	A84, A100

Determination of fitting

Examination of internal clearance

	Page number
Fitting	
Difference in temperature between inner and outer rings	A95
Speed	
Misalignment of inner and outer rings	A18
Amount of preload	A98

Determination of internal clearance

Study of cage

- Speed
- Noise
- Operating temperature
- External vibration and shock
- Rapid acceleration and deceleration
- Moment load and misalignment

Selection of cage type and material

Examination of special specifications

	Page number
Operating temperature	A57
Environment (seawater, vacuum, gases, chemicals, etc.)	
Type of lubrication	

Selection of special material, heat treatment for dimensional stability

Examination of lubricating methods

	Page number
Operating temperature range	A106, A107, A110, A112
Speed	A37
Lubricating methods	A105
Type of seals	A102
Maintenance and inspection intervals	A123

Selection of lubricating method, lubricant, and type of seals

Examination of ease of mounting/dismounting

	Page number
Procedure for mounting and dismounting	A116, A121
Necessary equipment	A116, A121
Dimensions affecting mounting	A100

Determination of dimensions affecting mounting and procedure for mounting/dismounting

Final specifications for bearing and surrounding parts

Fig. 2.1 Flow Chart for Selection of Rolling Bearings