DEPARTMENT OF PRODUCTION ENGINEERING

Programme B.Tech

Semester-7th

Course Title: PRINCIPLE OF MACHINE TOOL

COURSE CODE: BPE 403

Module 1

MACHINE TOOL DRIVES

Learning Objectives:

- Introduction to machine tool drives & mechanisms
- Working and auxiliary motions in machine tool
- Parameters defining working motions of machine tool
- Mechanical transmission and its elements
- Broadly Classification of transmission of rotary motion
- Stepped Speed Drives in Machine Tools
    - Belting
    - Pick-Off Gears
    - Gear boxes
- AP &GP for steeping speeds of gears
- Structural formula & structural diagrams
- Feed gear boxes
- Steeples Speed Drives in Machine Tools.
- Hydraulic transmission and its elements

Introduction

The machine tool is a machine that imparts the required shape to 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
    a. Machine tool with manual control
    b. Semi-automatic machine tools and
    c. Automatic machine tool.

2. By weight into
    a. Light duty machine tools weighing up to 1 tonne.
    b. Medium duty machine tools weighing up to 10 tonne and
    c. Heavy duty machine tools weighing greater than 10 tonne.

3. By the degree of specialisation into
    a. General purpose machine tools – which can perform various operations on workpiece of different shapes and sizes.
    b. Single purpose machine tools – which can perform single operation on work pieces of a particular shape and different sizes and

c. Special machine tools – which can perform a single operation on workpiece of particular shape and size.

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 can further be classified as:

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

Working motions in machine tools are generally of two types: rotary & translatory. Working motions of some important groups of machine tools are shown in figure:

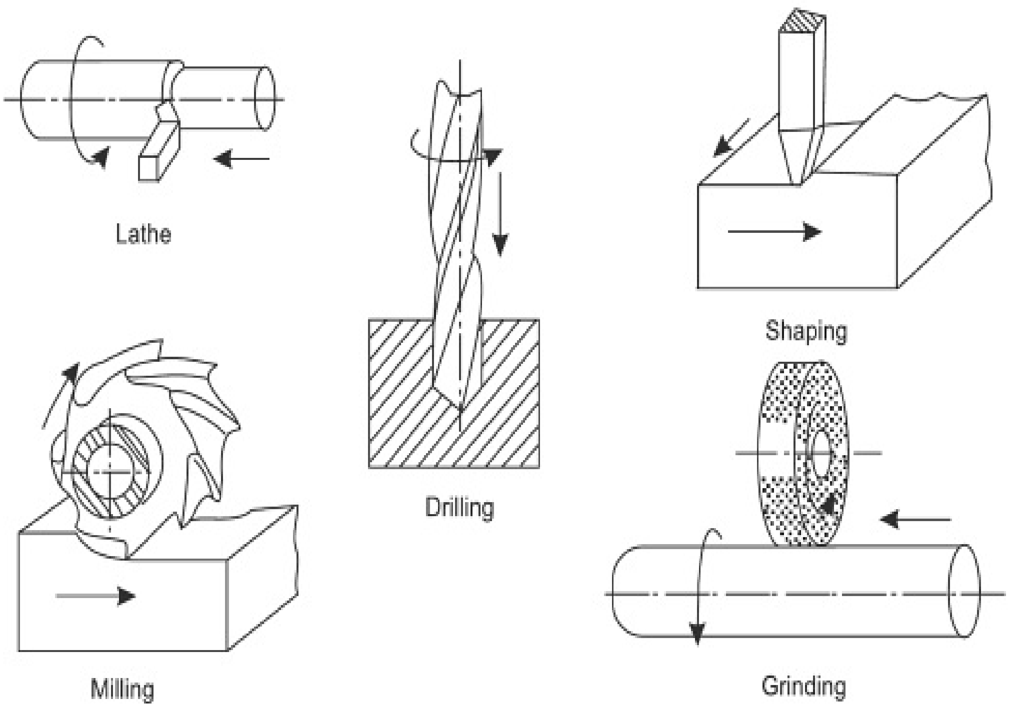

Fig. 1.1 *Working motions for some machine tools*

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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.

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Machine Tools produce desired geometrical surfaces on solid bodies (preformed blanks) and for that they are basically comprised of;

- Devices for firmly holding the tool and work
- Drives for providing power and motions to the tool and work
- Kinematic system to transmit motion and power from the sources to the tool-work
- Automation and control systems
- Structural body to support and accommodate those systems with sufficient strength and rigidity.

For material removal by machining, the work and the tool need relative movements and those motions and required power are derived from the power source(s) and transmitted through the kinematic system(s) comprised of a number and type of mechanisms.

(i) Concept of Generatrix and Directrix

- Generation of flat surface

The principle is shown in Fig. 2.1 where on a flat plain a straight line called Generatrix (G) is traversed in a perpendicular direction called Directrix (D) resulting a flat surface.

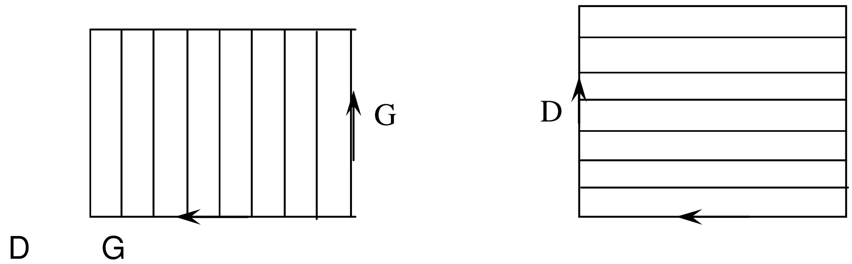
- Generation of cylindrical surfaces

The principles of production of various cylindrical surfaces (of revolution) are shown in Fig. 2.2, where,

– A long straight cylindrical surface is obtained by a circle (G) being traversed in the direction (D) parallel to the axis as shown in Fig. 2.2(a)

– A cylindrical surface of short length is obtained by traversing a straight line (G) along a circular path (D) as indicated in Fig. 2.2(b)

– Form cylindrical surfaces by rotating a curved line (G) in a circular path (D) as indicated in Fig. 2.2 (c and d).



(a) (b)

Fig. 2.1 *Generation of flat surfaces by Generatrix and Directrix.*

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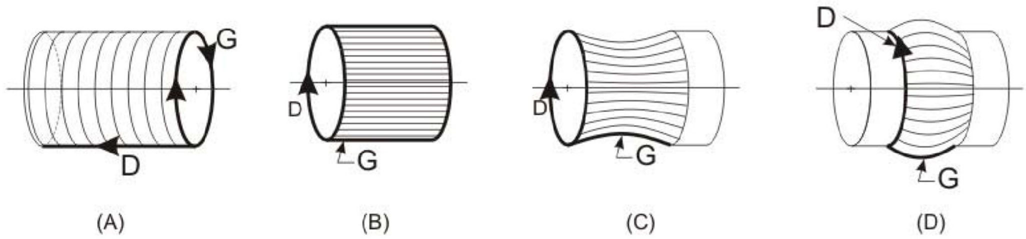


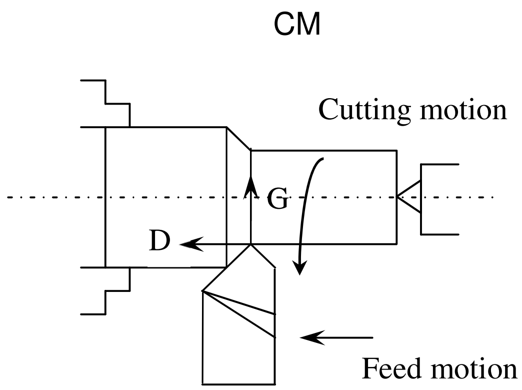
Fig. 2.2 *Generation of cylindrical surfaces (of revolution)*

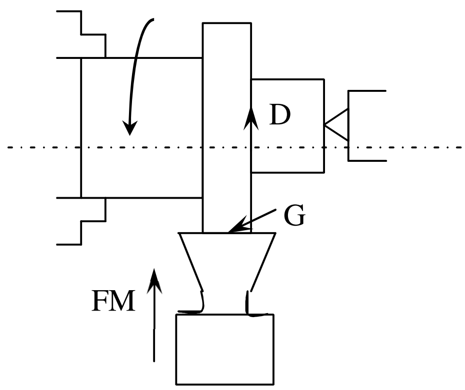
(ii) Tool – work motions

The lines representing the Generatrix and Directrix are usually produced by the locus of a point moving in two different directions and are actually obtained by the motions of the tool-tip (point) relative to the work surface. Hence, for machining flat or curved surfaces the machine tools need relative tool work motions, which are categorized in following two groups:

- Formative motions namely
  - Cutting motion (CM)
  - Feed motion (FM)
- Auxiliary motions such as
  - Indexing motion
  - Additional feed motion
  - Relieving motion

The Generatrix and Directrix, tool and the work and their motions generally remain interconnected and in different way for different machining work. Such interconnections are typically shown in Fig. 2.3 for straight turning and in Fig. 2.4 for shaping.





(a) longitudinal turning (b) transverse turning

Fig. 2.3 *Principle of turning (cylindrical surface)*

The connections in case of straight longitudinal turning shown in Fig. 2.3 (a) are:

Generatrix (G) – Cutting motion (CM) – Work (W)
Directrix (D) – Feed motion (FM) – Tool (T)

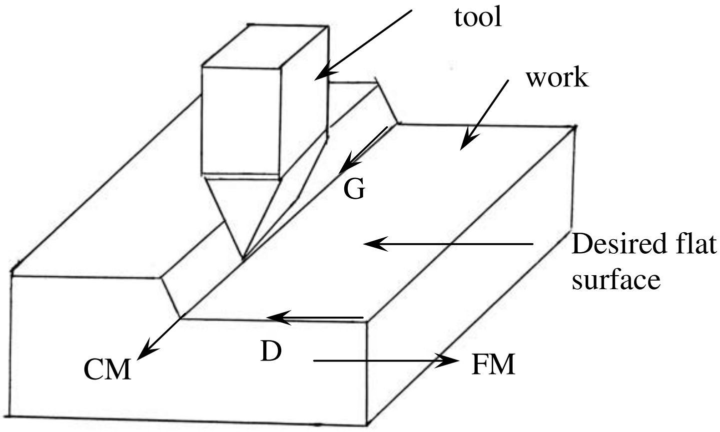


Fig. 2.4 *Principle of producing flat surface in shaping machine*

In case of making flat surface in a shaping machine as shown in Fig. 2.4 the connections are:

G – CM – T
D – FM – W

which indicates that in shaping flat surfaces the Generatrix is provided by the cutting motion imparted to the cutting tool and the Directrix is provided by the feed motion of the work.

Flat surfaces are also produced by planning machines, mainly for large jobs, where the cutting motion is imparted to the work and feed motion to the tool and the connections will be: G – CM – Work
D – FM – Tool

The Genratrix and Directrix can be obtained in four ways:

- Tracing (Tr) – where the continuous line is attained as a trace of path of a moving point as shown in Fig. 2.3 and Fig. 2.4.

- Forming (F) – where the Generatrix is simply the profile of the cutting edge as indicated in Fig. 2.2 (c and d)
- Tangent Tracing (TTr) – where the Directrix is taken as the tangent to the series of paths traced by the cutting edges as indicated in Fig. 2.5.
- Generation (G): Here the G or D is obtained as an envelope being tangent to the instantaneous positions of a line or surface which is rolling on another surface. Gear teeth generation by hobbing or gear shaping is the example as can be seen in Fig. 2.6.

Fig. 2.5 typically shows the tool-work motions and the corresponding Generatrix (G) and Directrix (D) while producing flat surface by a plain or slab milling cutter in a conventional horizontal arbour type milling machine. The G
and D are connected here with the tool work motions as

G – x – T – F
D – FM – W – T.Tr
CM – T

Here G and D are independent of the cutting motion and the G is the line of contact between the milling cutter and the flat work surface. The present cutter being of roller shape, G has been a straight line and the surface produced has also been flat. Form milling cutters will produce similar formed surfaces as shown in Fig. 2.7 where the 'G' is the tool-form.

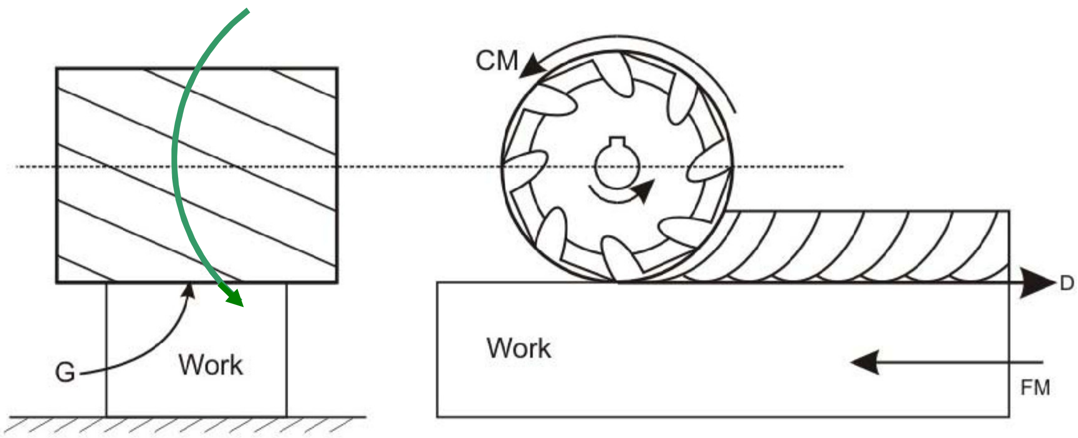
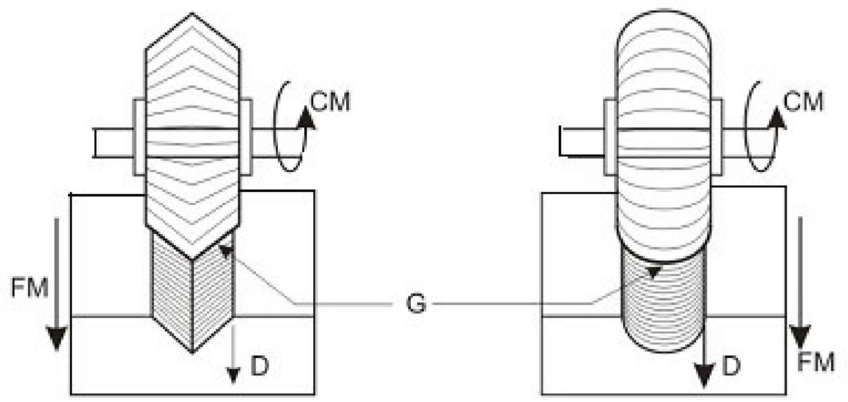


***Fig. 2.5** Directrix formed by tangent tracing in plain milling*



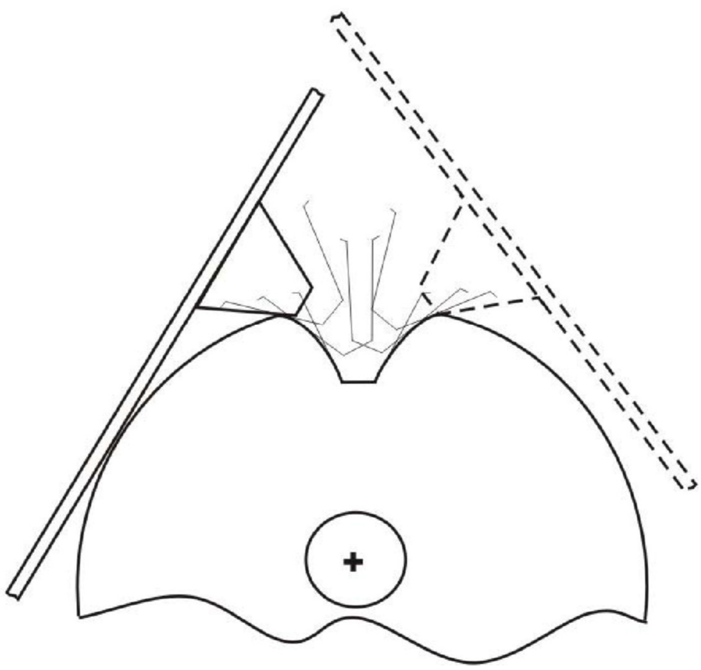


Fig. 2.6 *Generatrix (or Directrix) in gear teeth cutting by generation.*

Fig. 2.7 *Tool-work motions and G & D in form milling*

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For making holes in drilling machines both the cutting motion and the feed motion are imparted to the cutting tool i.e., the drill bit whereas the workpiece remains stationary. This is shown in Fig. 2.8. The G and D are linked with the tool-work in the way:

G – CM – T – Tr
D – FM – W – Tr

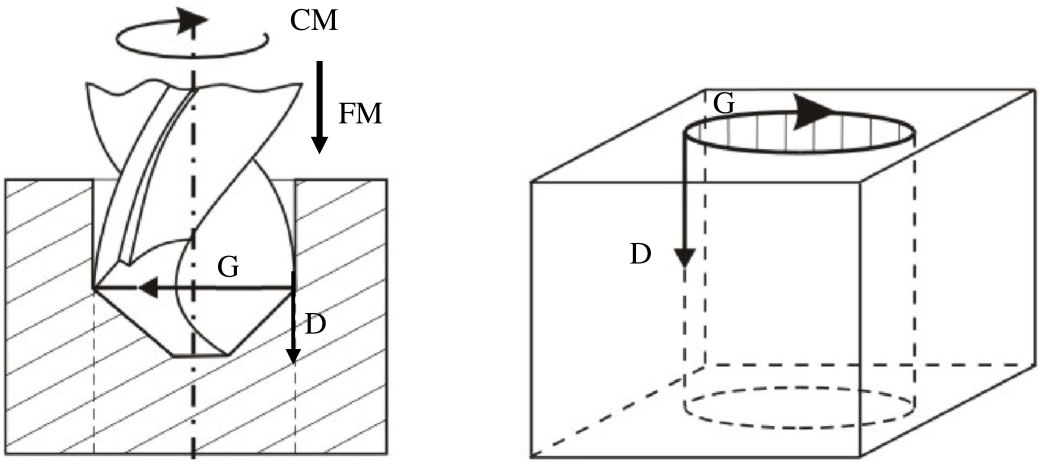

Fig. 2.8 *Tool-work motions and G & D in drilling.*

Boring machines are mostly used for enlargement and finishing of existing cylindrical holes. Boring machines are of two types:

- Vertical boring machine – low or medium duty and high precision, e.g., Jig boring machine
- Horizontal axis boring machine – medium or heavy duty.

In respect of tool-work motions and G and D, vertical boring and drilling are same. In horizontal boring machine the feed motion is imparted to the work to provide the Directrix by Tracing.

Parameters defining working motions of machine tool

Module 1

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

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

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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

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

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

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

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

$$s_m = s \cdot n \quad (1.5)$$

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:

$$s = s_z \cdot Z \quad (1.6)$$

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:

$$T_m = \frac{L}{s_m} \text{ min} \quad (1.7)$$

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

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

Elementary transmissions that transfer rotation

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

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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.

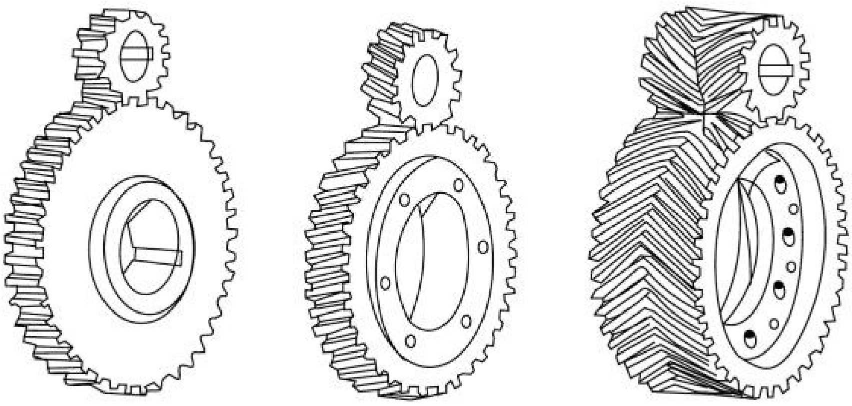
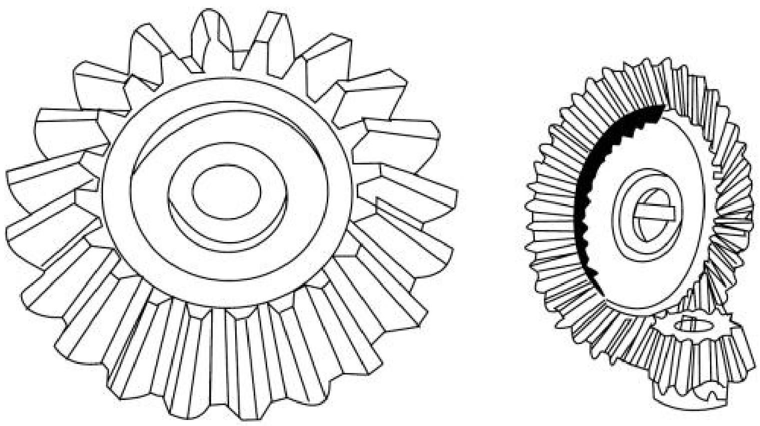

(a) (b) (c)

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.

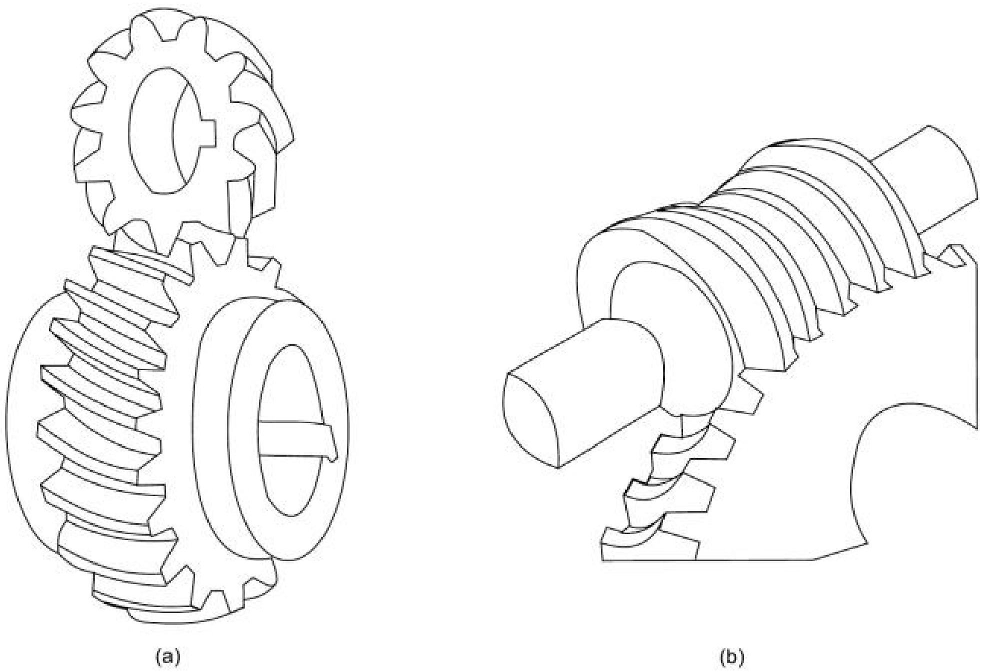
Transmission of rotation between inclined intersecting axes is done with the help of bevel gears. A bevel gear is shown in Fig. 1.35a. The angle between the inclined axes is generally 90° and the bevel-gear transmission (Fig. 1.35b) is commonly employed for transmitting rotation between perpendicular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

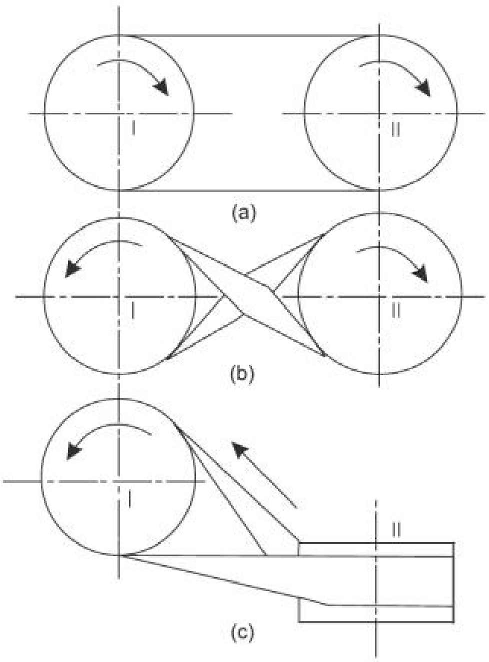


Fig. 1.37 *Belt drives: (a) Open-belt arrangement (b) Cross-belt arrangement (c) Quarter-turned arrangement*

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.

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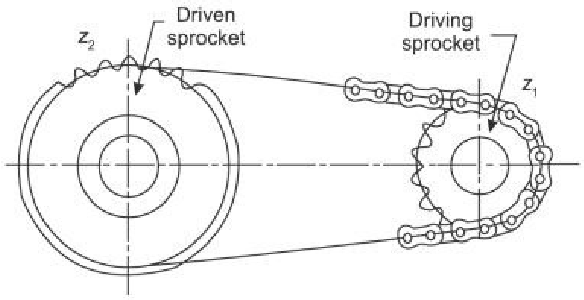


Fig. 1.38 *Chain transmission*

Elementary transmissions that transfer rotary motion into translatory motion

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.

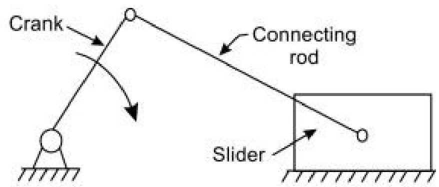


Fig. 1.39 *Slider crank mechanism*

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

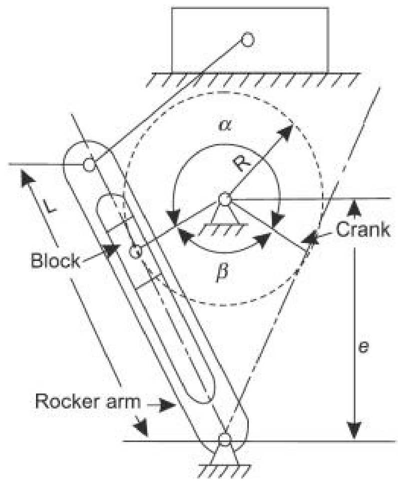


Fig. 1.40 *Crank-and-rocker mechanism*

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).

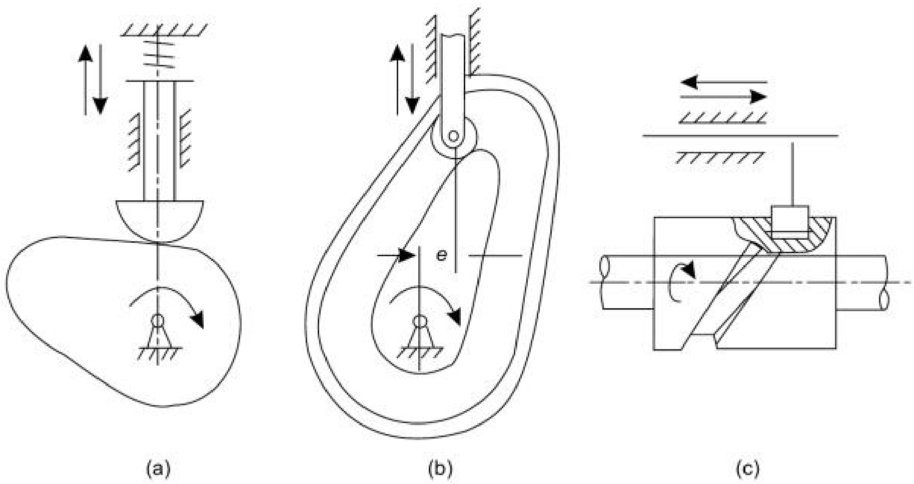


Fig. 1.41 *Cam mechanism: (a) Disc type (b) Face type (c) Drum type*

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

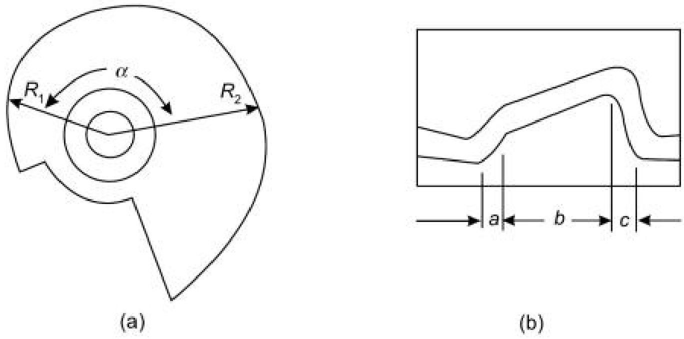


Fig. 1.42 *(a) Profile of a disc-type cam (b) Development of the profile of a drum-type cam*

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,

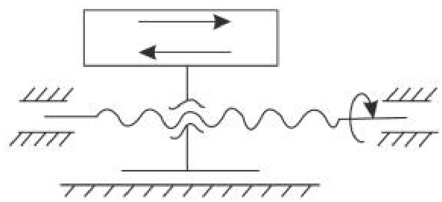


Fig. 1.43 *Schematic diagram of a nut-and-screw transmission*

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

where 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.

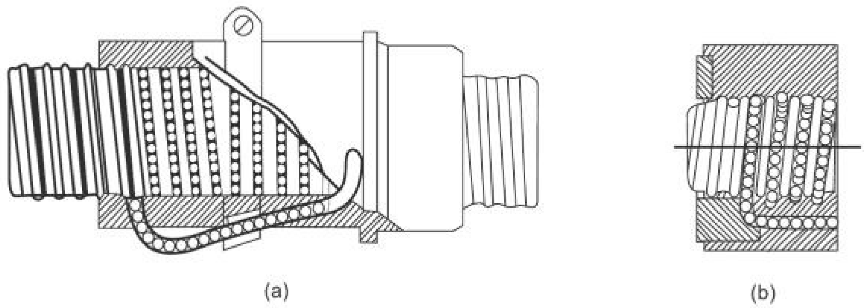


Fig. 1.44 *Schematic diagram of anti-friction nut-and-screw transmission*

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 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

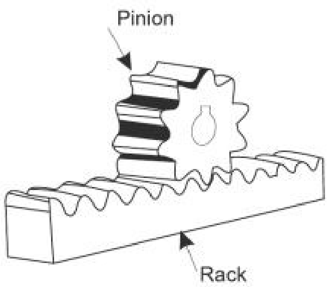


Fig. 1.45 *Rack-and-pinion transmission*

Module 1

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.

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.

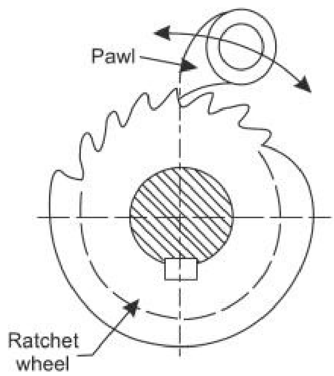


Fig. 1.46 *Pawl-and-ratchet mechanism*

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.

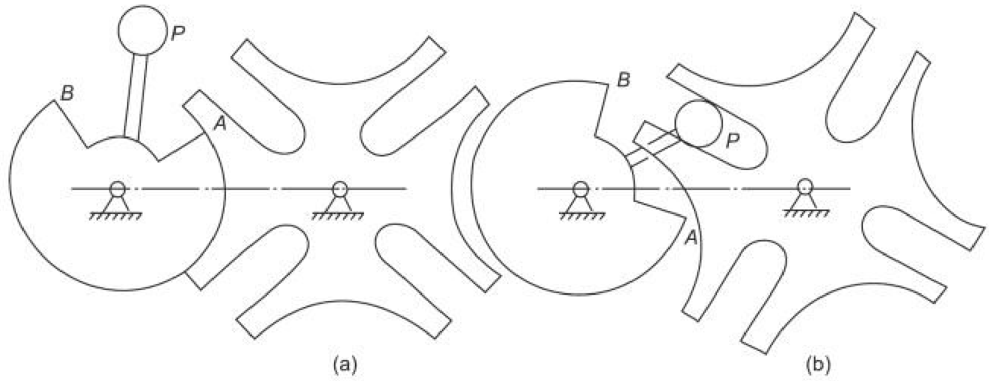


Fig. 1.47 *Geneva mechanism*

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.

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.

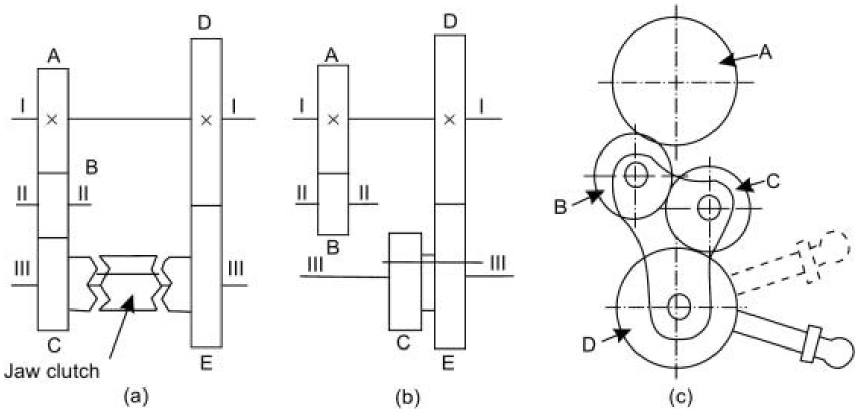


Fig. 1.48 *Reversing mechanisms: (a) using spur gears (b) using helical gears*

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.

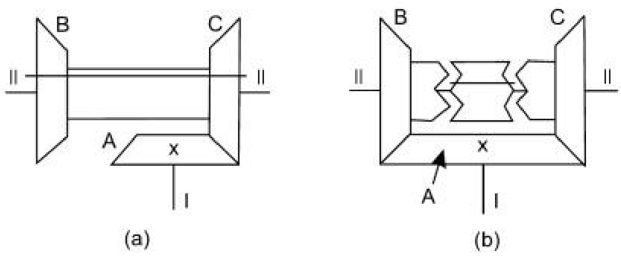


Fig. 1.49 *Reversing mechanisms using bevel gears*

Module 1

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:

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

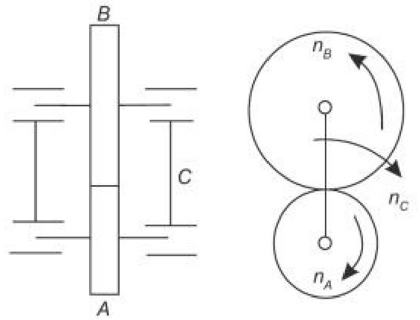


Fig. 1.50 *Differential mechanism using spur or helical gears*

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

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

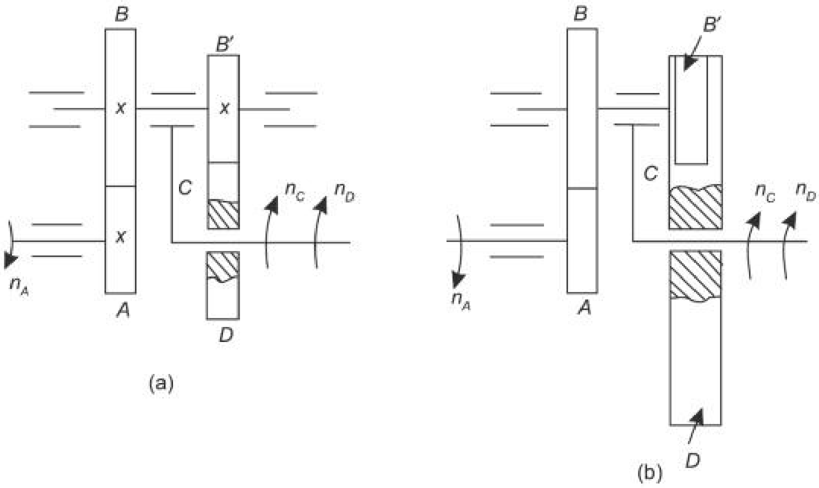


Fig. 1.51 *Differential mechanisms using double-cluster planetary gears*

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.

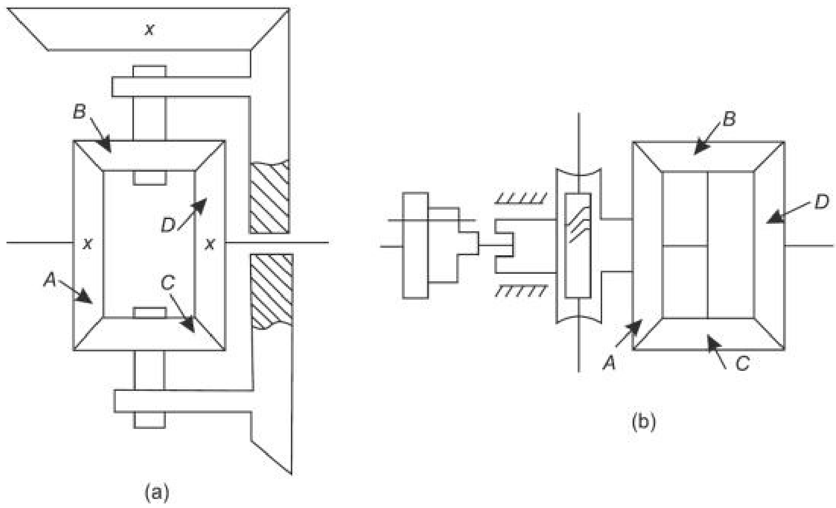


Fig. 1.52 *Differential mechanisms: (a) used in automobiles (b) used in machine tools*

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., $n_D = 2n_B$.

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.

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.

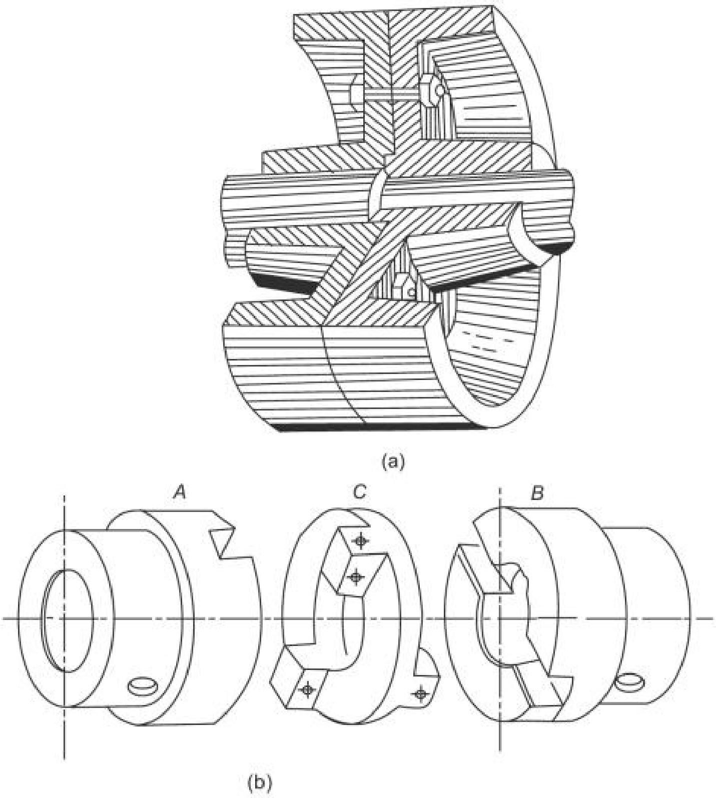


Fig. 1.53 *(a) Rigid coupling (b) Flexible coupling*

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.

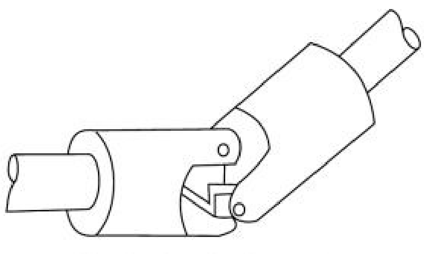


Fig. 1.54 *Elastic coupling*

Module 1

MACHINE TOOL DRIVES

To obtain a machined part by a machine tool, coordinated motions must be imparted to its working members. These motions are either primary (cutting and feed) movements, which removes the chips from the WP or auxiliary motions that are required to prepare for machining and ensure the successive machining of several surfaces of one WP or a similar surface of different WPs. Principal motions may be either rotating or straight reciprocating. In some machine tools, this motion is a combination of rotating and reciprocating motions. Feed movement may be continuous (lathes, milling machine, drilling machine) or intermittent (shapers, planers). As shown in Figure1. Stepped motions are obtained using belting or gearing. Step less speeds are achieved by mechanical, hydraulic, and electrical methods.

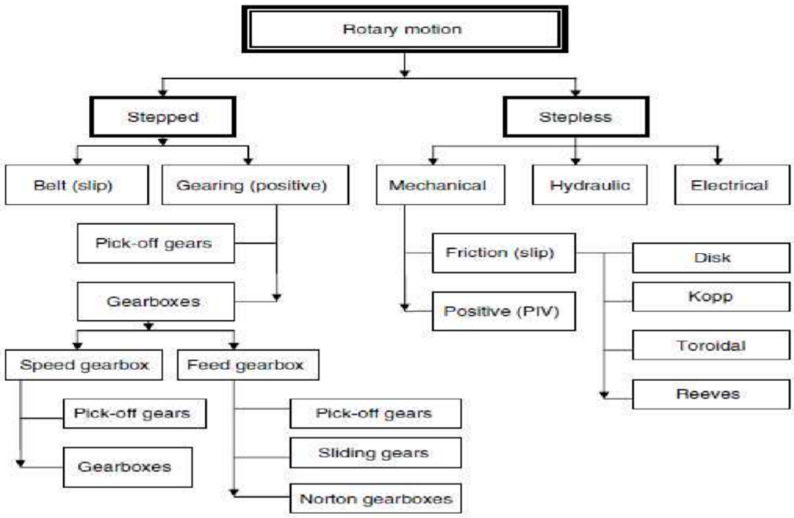


Fig 1.Classification of transmission of rotary motion.

STEPPED SPEED DRIVES:

Belting:

The belting system, shown in Figure 2. is used to produce four running rotational speeds n_1, n_2, n_3,and n_4. It is cheap and absorbs vibrations. It has the limitation of the low-speed changing, slip, and the need for more space. Based on the driver speed n_1, the following speeds

Module 1

Initial information required for designing a speed box

The following information is essentially required before we can start designing a stepped drive:

1. The highest output rpm, n_{max}
2. The lowest output rpm, n_{min}
3. The number of steps z into which the range between n_{max} and n_{min} is divided.
4. The number of stages in which the required number of speed steps are to be achieved.

An important parameter in designing speed boxes is the range ratio R_n given by

$$R_n = \frac{n_{max}}{n_{min}} = \frac{v_{max} \cdot d_{max}}{v_{min} \cdot d_{min}} = R_v \cdot R_d$$

Here R_v represents the range of cutting speeds employed on the machine tool and R_d the range of work-piece diameters machined.

There is a tendency of assigning higher cutting speeds for machining operations as new tool materials permitting higher cutting speeds are developed. While selecting the lowest and highest speed limits for a new machine tool, we must take into account its exploitation in actual production conditions. Figure 2.1[1] shows the probability distribution of using the various available rpm values for a high-speed milling machine for machining of aluminium and non-ferrous alloys. The speeds falling in the hatched portion can be ignored. A decision about the speed limits for the machine tool to be designed should be taken only after plotting such a probability-distribution curve on the basis of statistical data collected by observing the performance of machine tools doing similar operations. The rpm range which is rarely used is rejected. Both the upper and lower speed limits are selected from these considerations. However, in the case of the lower limit, exceptions are made in those cases in which the low cutting speeds, though rarely used, are essential for certain machining operations, e.g., thread cutting on a lathe. The lower cutting speed limit is often determined by the consideration that below a certain cutting speed, tool life begins to decrease, necessitating frequent regrinding of the cutting tool and resulting in lower productivity.

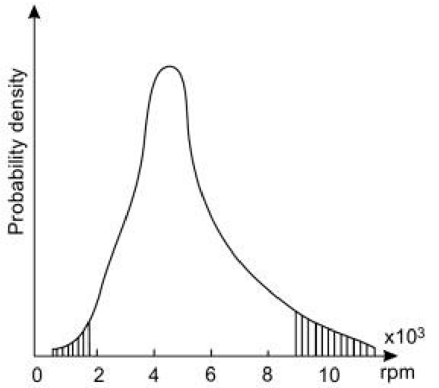


Fig. 2.1 *Probability distribution of utilising different rpm values of a high-speed milling machine*

The fixing of a higher speed limit certainly involves productivity loss in some machining operations in which a higher cutting speed is permissible, but the corresponding rpm value is not available on the machine tool. However, a very wide speed range is generally neither practicable nor economically feasible. The value of R_v should, therefore, be kept within reasonable limits.

The range of diameters should also be selected on the basis of the statistical study of the working of similar machine tools. Investigations conducted by ENIMS (Machine Tool Research Institute, Moscow, Russia) reveal that a ratio of $R_d = d_{max}/d_{min} = 4$ covers more than 85% of the workpieces, while $R_d = 6$ covers 92% of the workpieces.

can be obtained in a decreasing order:

$$n_1 = n\frac{d_1}{d_5}$$

$$n_2 = n\frac{d_2}{d_6} \qquad n_3 = n\frac{d_3}{d_7}$$

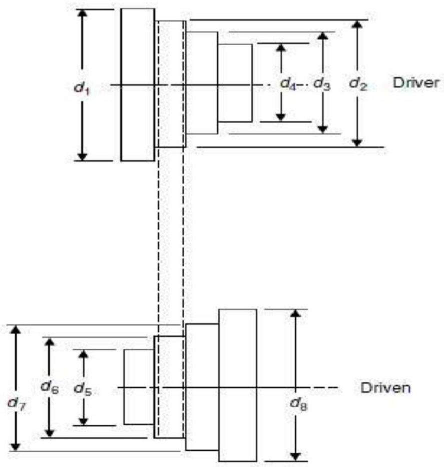


FIG 2. Belting transmission This type is commonly used for grinding and bench-type drilling machines.

Pick-Off Gears

Pick-off gears are used for machine tools of mass and batch production (automatic and semiautomatic machines, special-purpose machines, and so on) when the changeover from job to job is comparatively rare. Pick-off gears may be used in speed or feed gearboxes. As shown in Figure 3, the change of speed is achieved by setting gears A and B on the adjacent shafts. As the center distance is constant, correct gear meshing occurs if the sum of teeth of gears A and B is constant.

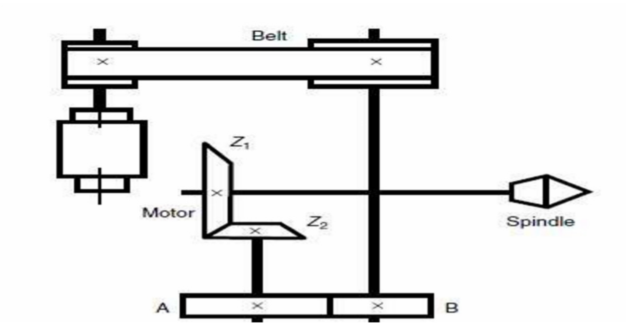


Fig 3.Pick-Off Gears

Gearboxes

Machine tools are characterized by their large number of spindle speeds and feeds to cope with the requirements of machining parts of different materials and dimensions using different types of cutting tool materials and geometries. The cutting speed is determined on the bases of the cutting ability of the tool used, surface finish required, and economic considerations.

A wide variety of gearboxes utilize sliding gears or friction or jaw coupling. The selection of a particular mechanism depends on the purpose of the machine tool, the frequency of speed change, and the duration of the working movement. The advantage of a sliding gear transmission is that it is capable of transmitting higher torque and is small in radial dimensions. Among the disadvantages of these gearboxes is the impossibility of changing speeds during running. Clutch-type gearboxes require small axial displacement needed for speed changing, less engagement force compared with sliding gear mechanisms, and therefore can employ helical gears. The extreme spindle speeds of a machine tool main gearbox n_{max} and n_{min} can be determined by

$$n_{\max} = \frac{1000V_{\max}}{\pi d_{\min}} \qquad n_{\min} = \frac{1000V_{\min}}{\pi d_{\max}}$$

Where

V_{max} = maximum cutting speed (m/min) used for machining the most soft and machinable material with a cutting tool of the best cutting property

V_{min} = minimum cutting speed (m/min) used for machining the hardest material using a cutting tool of the lowest cutting property or the necessary speed for thread cutting d_{max}, d_{min} = maximum and minimum diameters (mm) of WP to be machined The speed range *Rn* becomes

$$R_n = \frac{n_{\max}}{n_{\min}} = \frac{V_{\max}}{V_{\min}} \cdot \frac{d_{\max}}{d_{\min}} = R_v \cdot R_d$$

Where

R_v = cutting speed range

Rd = diameter range

In case of machine tools having rectilinear main motion (planers and shapers), the speed range Rn is dependent only on Rv. For other machine tools, Rn is a function of Rv and Rd, large cutting speeds and diameter ranges are required. Generally, when selecting a machine tool, the speed range Rn is increased by 25% for future developments in the cutting tool materials. Table 1 shows the maximum speed ranges in modern machine tools. Table 1

Speed Range for Different Machine Tools

Machine	Range
Numerically controlled lathes	250
Boring	100
Milling	50
Drilling	10
Surface grinding	4

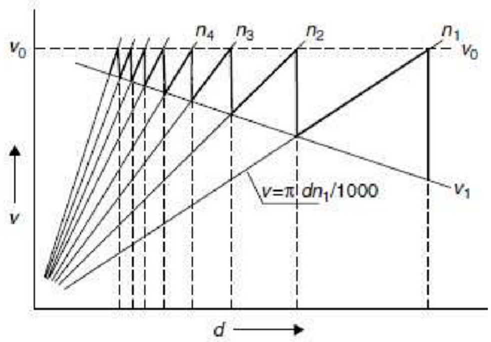
Module 1

Stepping of Speeds According to Arithmetic Progression:

Let $n_1, n_2, \dots, n_z$ be arranged according to arithmetic progression. Then $\mathbf{n_1 - n_2 = n_3 - n_2 = constant}$

The saw tooth diagram in such a case is shown in Figure 4. Accordingly, for an economical Cutting speed v_0, the lowest speed v_l is not constant; it decreases with increasing diameter. Therefore, the arithmetic progression does not permit economical machining at large diameter ranges

The main disadvantage of such an arrangement is that the percentage drop from step to step δn decreases as the speed increases. Thus the speeds are not evenly distributed and more concentrated and closely stepped, in the small diameter range than in the large one. Stepping speeds according to arithmetic progression are used in Norton gearboxes or gearboxes with a sliding key when the number of shafts is only two.



Speed stepping according to arithmetic progression.

Stepping of Speeds According to Geometric Progression

As shown in Figure, the percentage drop from one step to the other is constant, and the absolute loss of economically expedient cutting speed Δv is constant all over the whole diameter range. The relative loss of cutting speed $\Delta v_{max}/v0$ is also constant. Geometric progression, therefore, allows machining to take place between limits v_0 and v_u independent of the WP diameter, where v_0 is the economical cutting speed and v_u is the allowable minimum cutting speed. Now suppose that n_1, n_2,

Module 1

n_3, …, nz are the spindle speeds. According to the geometric Progression,

$$\frac{n_2}{n_1} = \frac{n_3}{n_2} = \varphi$$

Where φ is the progression ratio. The spindle speeds can be expressed in terms of the minimal speed n_1 and progression ratio φ.

n_1	n_2	n_3	n_4	n_z
n_1	$n_1\varphi$	$n_1\varphi^2$	$n_1\varphi^3$	$n_1\varphi^{z-1}$

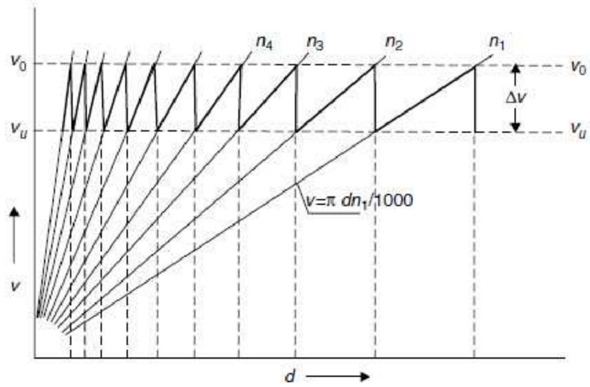
$$n_z = n_1\varphi^{z-1}$$

Hence, the maximum spindle speed n_z is given by
Where z is the number of spindle speeds, therefore,

$$\varphi = \sqrt[z-1]{\frac{n_z}{n_1}} = \sqrt[z-1]{R_n} = (R_n)^{1/(z-1)}$$

$$z = \frac{\log R_n}{\log \varphi} + 1$$

From which



Speed stepping according to geometric progression.

ISO Standard values of progression ratios φ (1.06, 1.12, 1.26, 1.4, 1.6, 1.78,2.0)

Breakup Speed steps:

Z: No of speed steps φ: No of transmissions

$z= P_1*P_2*P_3*\ldots\ldots\ldots\ldots*P_u$
P: no of speeds in each transmission
To obtain the Z speeds with minimum no of gears $Z=P^{u}$.
When

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$P=Z^{1/u}$ is not a whole no divide the speed steps in such a way that
$Z=2^{E1}*3^{E2}$...........(1)

Eqn 1 is satisfied by the numbers Z= 2,3,4,6,8,9,12,16,18,24,27,32,36, etc

Structural formulae & Structural Diagrams:

Suppose a speed on one shaft yields two speed values on the next shaft. The noo of speed steps of the particular transmission group is p=2.if the transmission is through gears, the transmission ratios that provide the two new speed values must lie in the following range:

$$n1/nz= i_{max}=2 \text{ (maximum increase in speed)}$$

$$n2/nz=i_{min}=1/4 \text{(max reduction in speed)}$$

Transmission range for any stage: $i_g= i_{max}/ i_{min}=8$

For any stage: if there are Z^1 steps: $n_1/n_2= n2/n3=\ldots\ldots\ldots n_{z^1-1}/n_{z1}=K$ Since speeds are to be in GP $K=\varphi^x$.

X characteristics of the transmission stage .

$Z=P_1*P_2*\ldots\ldots\ldots..P_u$

$$Z \quad =P_1(x_1)*P2(x_2)*\ldots.P_u(x_u)\ldots.. \textbf{ Structural Formula}$$

P_1 speed steps in stage. x_1 characteristic of stage .

Where $X_1=1, X_2=P_1, X_3=P_1*P_2,\ldots\ldots\ldots X_n=P_1*P_2*P_{n-1}$.

Ex: Z=12, u=3

Z=2*2*3
=2*3*2
=3*2*2

Structural formula:

$X_1=1$, $X_2=P_1$,$X_3=P1*P2$.

Z=2(1)*2(2)*3(4)
=2(1)*3(2)*2(6)
=3(1)*2(3)*2(6)

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GUIDELINES FOR SELECTING BEST STRUCTURAL FORMULA:

1. Transmission ratio $i_{max}=2, i_{min}=1/4, \quad i_g= i_{max}/ i_{min}=8.$
2. Minimum total shaft size:

The torque transmitted by a shaft is given by

T α 1/N;

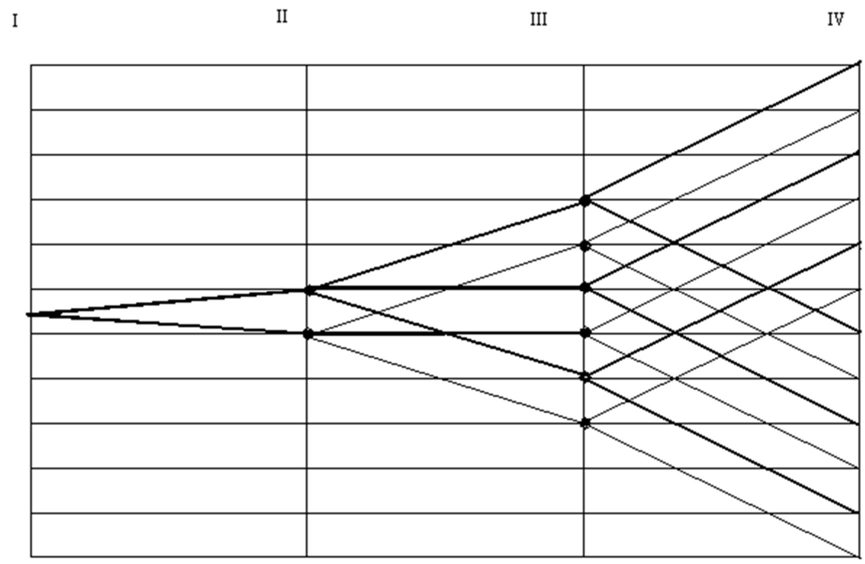
From the strength consideration : $(d_1/d_2)= (N_2/N_1)^{1/3}$

3. For least radial dimensions of gear box $i_{max} * i_{min}=1$.
4. No of gears on last shaft should be minimum.
5. No of gears on any shaft should be limited to three.

Limiting values of transmission intervals for different φ:

Type of transmission φ=	1.06	1.12	1.26	1.41	1.58	1.78	2.0
Speed reduction i<1	24	12	6	4	3	2	2
Speed increases i>1	12	6	3	2	1	1	1

Ex: draw the structural diagram for **2(1)3(2)2(6).**



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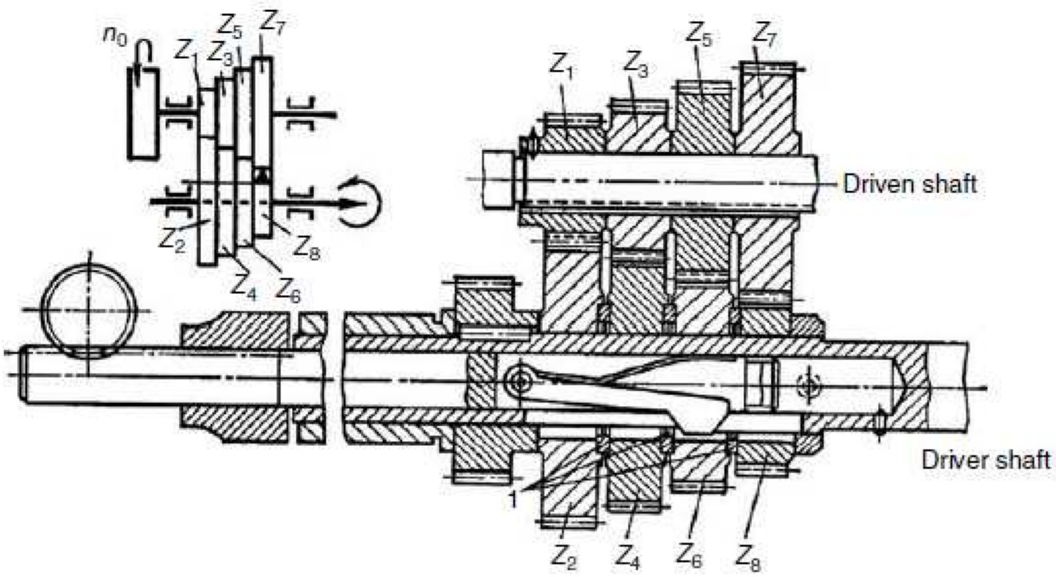
Feed Gearboxes:

Feed gearboxes are designed to provide the feed rates required for the machining operation. The values of feed rates are determined by the specifi ed surface fi nish, tool life, and the rate of material removal.

The classification of feed gearboxes according to the type of mechanism used to change the rate of feed is as follows:

1. ***Feed gearboxes with pick-off gears.*** Used in batch-production machine tools with infrequent Change over from job to job, such as automatic, semiautomatic, single-purpose, and specialpurpose machine tools. These gearboxes are simple in design and are similar to those used for speed changing

2. ***Feed gearboxes with sliding gears***. These gearboxes are widely used in general-purpose machine tools, transmit high torques, and operate at high speeds. Figure shows a typical gearbox that provides four different ratios. Accordingly, gears Z2, Z4, Z6, and Z8 are keyed to the drive shaft and mesh, respectively, with gears Z1, Z3, Z5, and Z7, which are mounted freely on the driven key shaft. The sliding key engages any gear on the driven shaft. The engaged gear transmits the motion to the driven shaft while the rest of the gears remain idle. The main drawbacks of such feed boxes are the power loss and wear occurring due to the rotation of idle gears and insufficient rigidity of the sliding key shaft. Feed boxes with sliding gears are used in small- and medium-size drilling machines and turret lathes.



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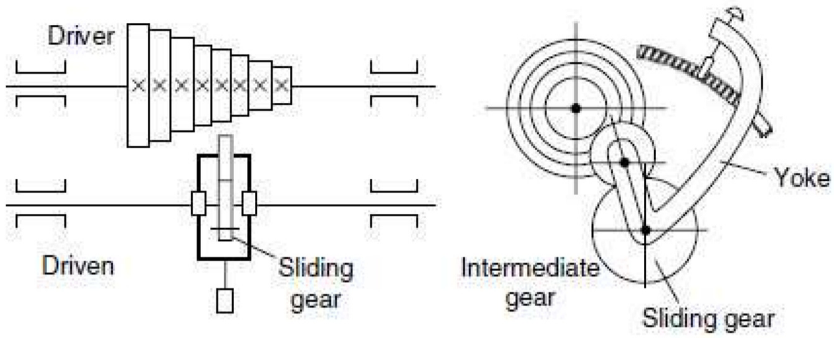
3. ***Norton gearboxes.*** These gearboxes provide an arithmetic series of feed steps that is suitable for cutting threads and so are widely used in engine lathe feed gearboxes as shown in FIG



STEPLESS SPEED DRIVES:

Stepless speed drives may be mechanical, hydraulic, or electric. The selection of the suitable drive depends on the purpose of the machine tool, power requirements, speed range ratio, mechanical characteristics of the machining operation, and cost of the variable speed unit. In most stepless drives, the torque transmission is not positive. Their operation involves friction and slip losses. However, they are more compact, less expensive, and quieter in operation than the stepped speed control elements

Mechanical Stepless Drives:

Infinitely variable speed (stepless) drives provide output speeds, forming infinitely variable ratios to the input ones. Such units are used for main as well as feed drives to provide the most suitable speed or feed for each job, thereby reducing the machining time. They also enable machining to be achieved at a constant cutting speed, which leads to an increased tool life and ensures uniform surface finish.

Mechanical stepless drives are 4 types:

- Friction Stepless Drive
- Kopp Variator
- Toroidal and Reeves Mechanisms
- Positive Infinitely Variable Drive

Friction Stepless Drive

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The disk-type friction stepless mechanism. Accordingly, the drive shaft rotates at a constant speed n_1 as well as the friction roller of diameter d. The output speed of the driven shaft rotates at a variable speed n_2 that depends on the instantaneous diameter D. Because

$$n_1 d = n_2 D \quad \text{Hence} \quad n_2 = n1 * d/D.$$

The diameter ratio d/D can be varied in infinitely small steps by the axial displacement of the friction roller. If the friction force between the friction roller and the disk is F,

$$F = \frac{\text{input torque (T1)}}{\text{input radius (d/2)}} = \frac{\text{output torque (T2)}}{\text{output radius (D/2)}}$$

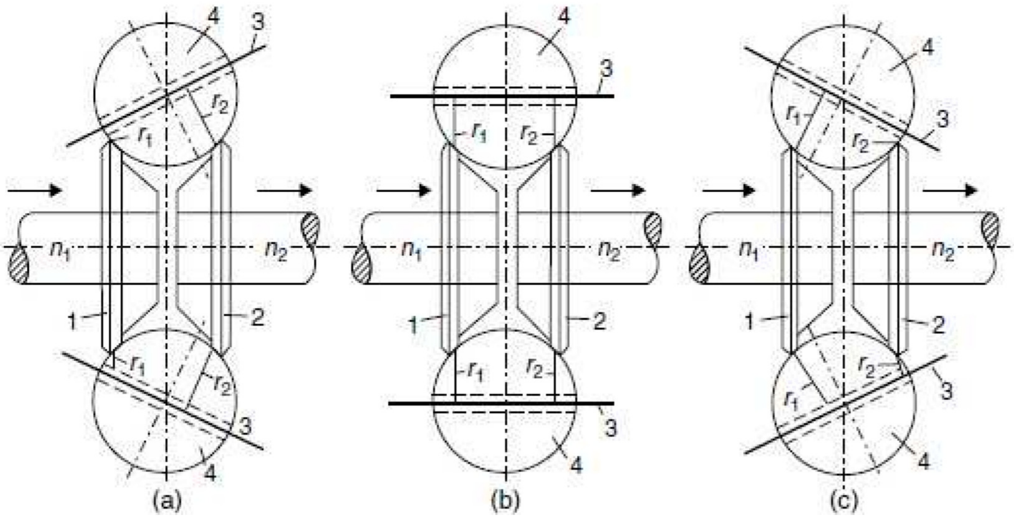
If the power, contact pressure, transmission force, and effi ciency are constant, the output torque T2 is inversely proportional to the speed of the output shaft n2. T2 α T1n1 /n 2

Due to the small contact area, a certain amount of slip occurs, which makes this arrangement suitable for transmitting small torques and is limited to reduction ratios not more than 1:4.

Kopp Variators

The drive balls (4) mounted on inclinable axes (3) run in contact with identical, effective radii r1 = r2, and drive cones (1 and 2) are fi xed on coaxial input and output shafts. When the axes of the drive balls (3) are parallel to the drive shaft axes, the input and output speeds are the same. When they are tilted, r1 and r2 change, which leads to the increase or decrease of the speed.

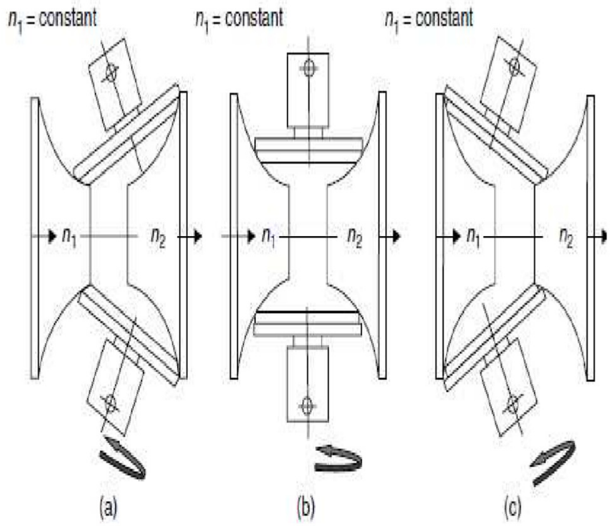
Using Kopp mechanism, a speed range of 9:1, effi ciency of higher than 80% and 0.25–12 hp capacity are obtainable.

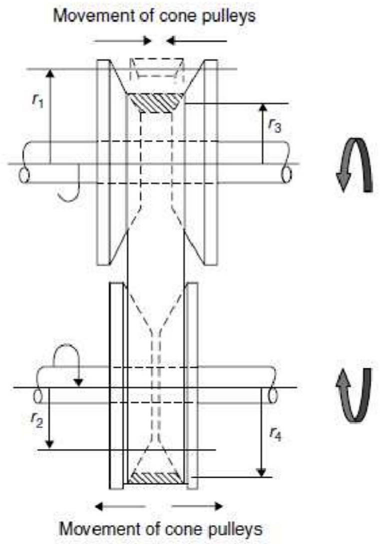


Kopp stepless speed mechanism: (a) n2 < n1, (b) n2 = n1, and (c) n2 > n1.

Toroidal and Reeves Mechanisms

The principle of Toroidal step less speed transmission. Figure shows the Reeves variable speed transmission, which consists of a pair of pulleys connected by a V-shaped belt; each pulley is made up of two conical disks. These disks slide equally and simultaneously along the shaft and rotate with it. To adjust the diameter of the pulley, the two disks on the shaft are made to approach each other so that the diameter is increased or decreased. The ratio of the driving diameter to the driven one can be easily changed and, therefore, any desired speed can be obtained without stopping the machine. Drives of this type are available with up to 8:1 speed range and 10 hp capacity.



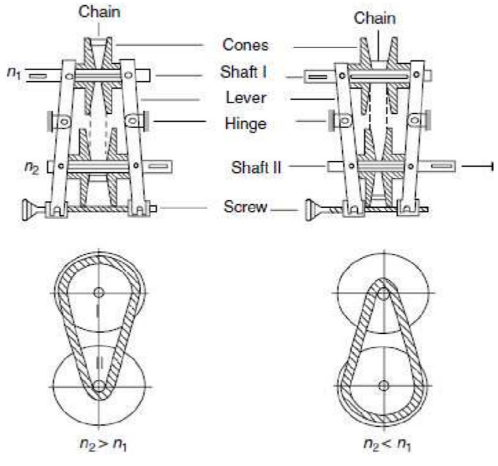


Reeves variable speed transmission.

Toroidal stepless speed transmission (a) n2 < n1, (b) n2 = n1, and (c) n2 > n1.

Positive Infinitely Variable Drive

Positive torque transmission arrangement that consists of two chain wheels, each of which consists of a pair of cones that are movable along the shafts in the axial direction. The teeth of the chain wheels are connected by a special chain. By rotating the screw, the levers get moved thus changing the location of the chain pulleys, and hence the speed of rotation provides a speed ratio of up to 6 and is available with power rating up to 50 hp. The use of infinite variable speed units in machine tool drives and feed units is limited by their higher cost and lower efficiency or speed range.



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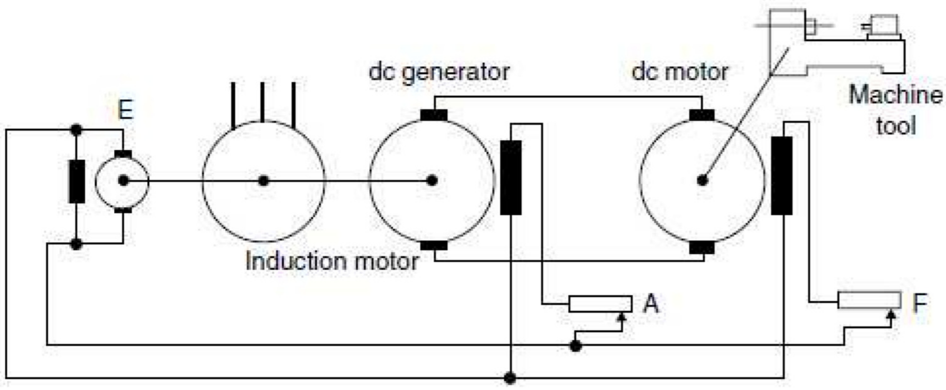
Positive Infinitely Variable Drive

Electrical Stepless Speed Drive

Figure shows the Leonard set, which consists of an induction motor that drives the direct current generator and an exciter (E). The dc generator provides the armature current for the dc motor, and the exciter provides the field current; both are necessary for the dc motors that drive the machine tool.



Leonard set (electrical stepless speed drive).

The speed control of the dc motor takes place by adjusting both the armature and the field voltages by means of the variable resistances A and F, respectively. By varying the resistance A, the terminal voltage of the dc generator and hence the rotor voltage of the dc motor can be adjusted between zero and a maximum value. The Leonard set has a limited efficiency: it is large, expensive, and noisy. Nowadays, dc motors and thyrestors that permit direct supply to the dc motors from alternating current (ac) mains are available and, therefore, the Leonard set can be completely eliminated. Thyrestor feed drives can be regulated such that the system offers infinitely variable speed control.

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 value 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.

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Module 1

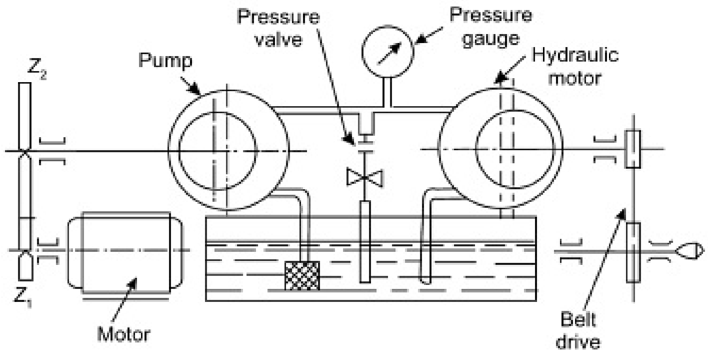

Fig. 1.20 *Schematic diagram of a rotary hydraulic drive*

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.

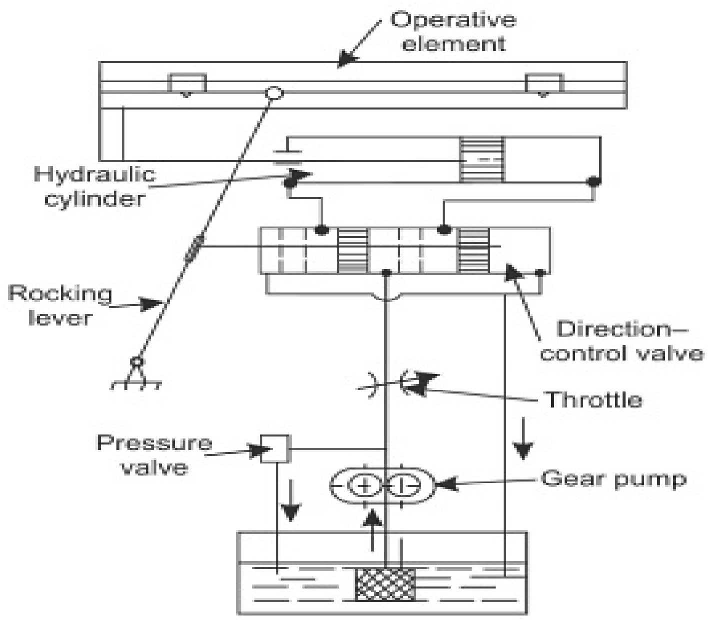

Fig. 1.21 *Schematic diagram of a translatory hydraulic drive*

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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.

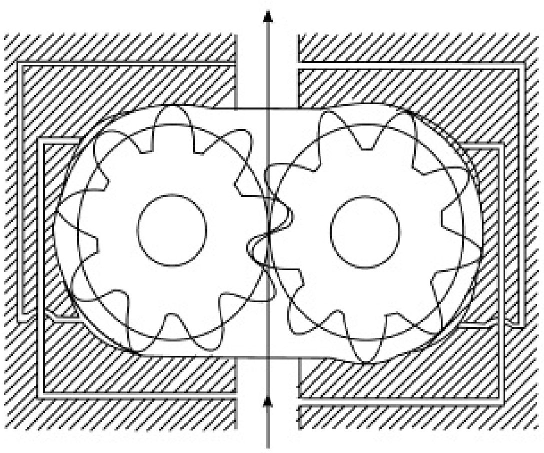

Fig. 1.22 *Schematic diagram of a gear pump*

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

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

where d_0 = pitch circle diameter of the gears, mm
d_e = addendum circle diameter of the gears, mm

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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.

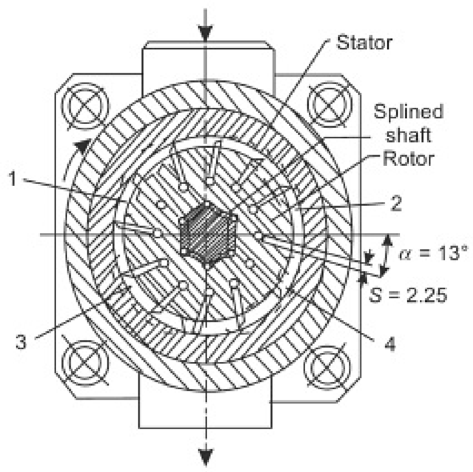


Fig. 1.23 *Schematic diagram of a constant-delivery vane pump*

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

$$Q = \frac{2Bn}{10^9}\left[\pi(r_2^2 - r_1^2) - \frac{(r_2 - r_1)t \cdot z}{\cos\alpha}\right] \text{m}^3/\text{min} \tag{1.15}$$

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 $\alpha = 13°$

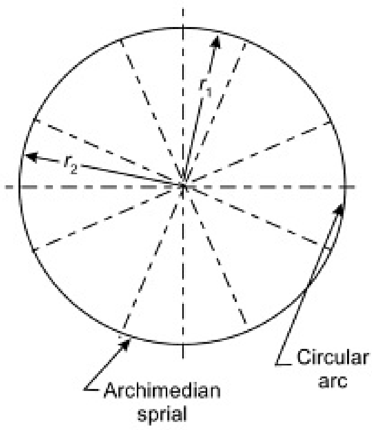


Fig. 1.24 *Profile of the stator of a constant-delivery vane pump*

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,

$$Q = \frac{2en}{10^9}[B(\pi D - tz) + 4\pi bd] \text{ m}^3/\text{min} \tag{1.16}$$

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

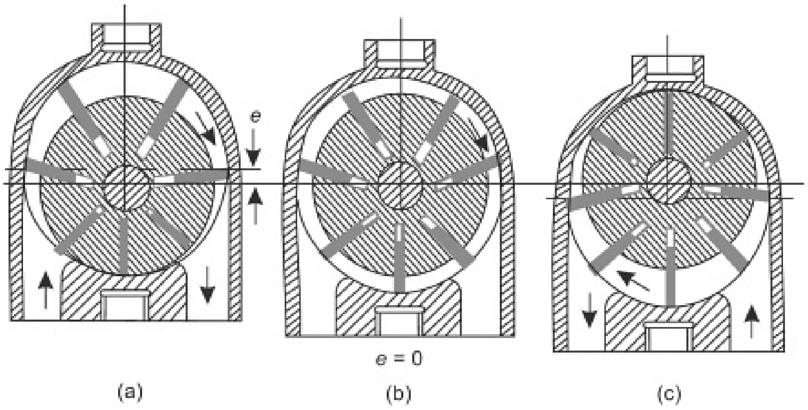


Fig. 1.25 *Schematic diagram of a variable-delivery vane pump*

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,

$$Q = \frac{\pi d^2 ezn}{2 \times 10^9} \text{ m}^3/\text{min} \tag{1.17}$$

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,

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

where P = resisting force, kgf
A = effective area of cross section of the piston, cm^2

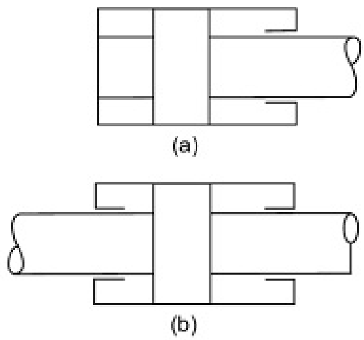


Fig. 1.26 *Cylinders: (a) Single-piston rod type (b) Double-end rod type*

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,

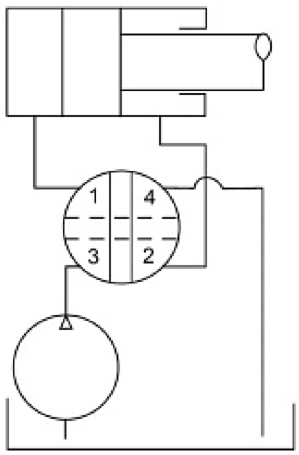


Fig. 1.27 *Schematic diagram of rotary, spool-type direction-control valve*

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.

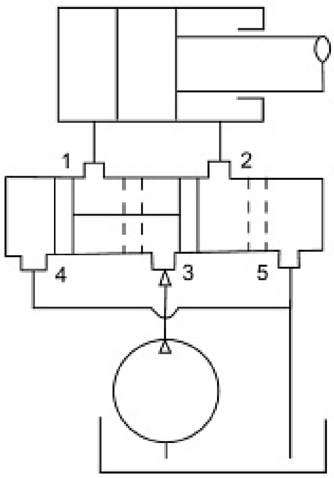


Fig. 1.28 *Schematic diagram of a four-way, two-position, piston-type direction-control valve*

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.

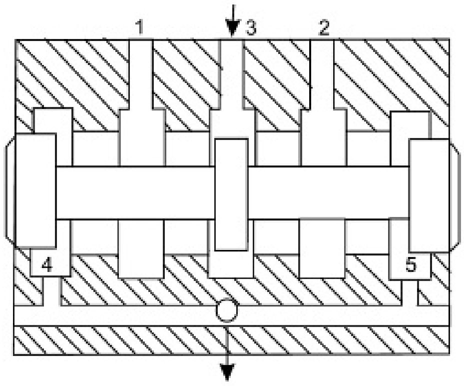


Fig. 1.29 *Schematic diagram of a four-way, three-position, piston-type direction-control valve*

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

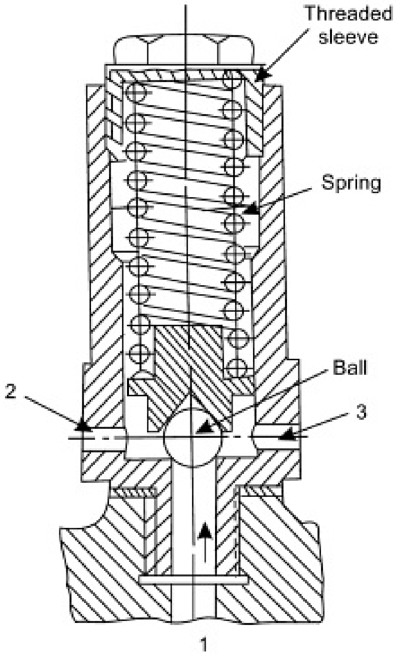


Fig. 1.30 *Schematic diagram of a ball-type pressure valve*

When due to increase in the pressure, force $P + F$ 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.

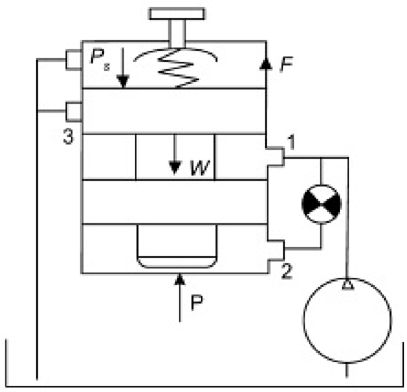


Fig. 1.31 *Schematic diagram of a spool-type pressure valve*

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.

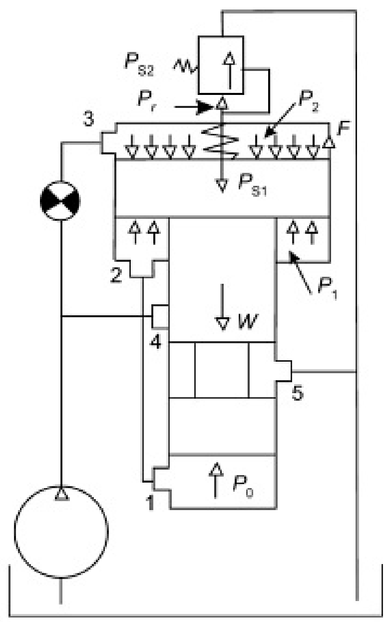


Fig. 1.32 *Schematic diagram of a compound-relief valve*

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.

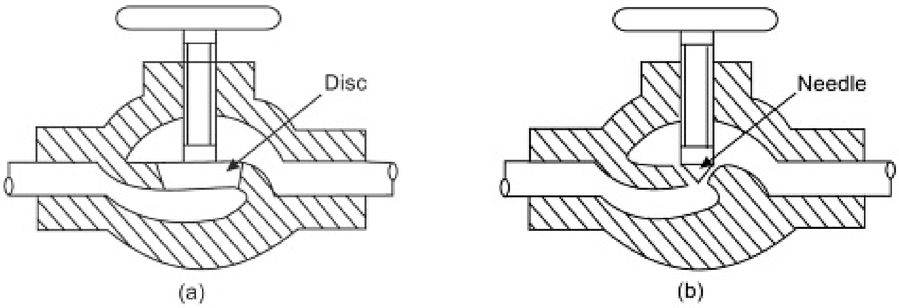


Fig. 1.33 *Throttle valves: (a) Globe valve (b) Needle valve*

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.

MODULE 2

DESIGN OF MACHINE STRUCTURES

1. INTRODUCTION

Machine tool parts such as beds, bases, columns, box-type housings, over arms, carriages, tables etc. are known as structures. These are the base of machine tool on which the guide ways, and spindle, carriage, etc. are mounted. These elements must able to withstand at higher permissible load. The structures depending upon their function may be broadly divided into the following three groups:

Group 1: beds and bases, upon which the various subassemblies are mounted.
Group 2: Box-type housings in which individual units are assembled, e.g., speed box housing, spindle head etc
Group 3: Parts that serve for supporting and moving the workpiece and cutting tool, e.g., table carriage, knee, tail stock, etc.

Machine tool structures must satisfy the following requirements:

1. All important mating surface of the structures should be machined with a high degree of accuracy to provide the desired geometrical accuracy.
2. The initial geometrical accuracy of the structures should be maintained during the whole service life of the machine tool.
3. The shapes and sizes of the structures should not only provide safe operation and maintenance of the machine tool but also ensure that working stresses and deformations are due to mechanical as well as thermal loading.

OBJECTIVES

After studying this unit, you should be able to understand

- functions of machine tool structure and the design criteria for selection of material for sideways
- the design of bed,
- the design of column, and
- The design of housing.

MODULE 2

1. FUNCTIONS OF MACHINE TOOL STRUCTURE

Machine tool structure consists of bed, base, columns, box type housings, over arms, carriages, tables etc.

The structures are divided into three categories according to their functions:

Category 1

An element, upon which various subassemblies are mounted, falls under this category. Example: bed and base.

Category 2

Elements consist of box type housings in which individual parts are assembled fall under this category. Example: Speed box housing, spindle head, etc.

Category 3

Elements consist of parts that are used for supporting and moving the work piece and cutting tool fall under this category. Example: Table, carriage, knee, tailstock etc.

Machine tool structure must satisfy the following requirement:

(a) The initial geometrical accuracy of the structure should be maintained for the whole life of the machine tool.

(b) All mating surfaces of the structure should be machined with a high degree of accuracy to provide the desired geometrical accuracy.

(c) The shape and size of structure should not only provide safe operation and maintenance of the machine tool but also ensure that working stresses and deformation should not exceed specific limits.

(d) The selection of material and high static and dynamic stiffness are the fundamental requirement to fulfil above-mentioned requirement.

SAQ 1:- What are functions and requirements of machine tool structure?

MODULE 2

2. DESIGN CRITERIA FOR MACHINE TOOL STRUCTURE

The simple machine tool bed with two-side wall is represented as a simply supported beam. Figure 1 depicts a simply supported beam. Point load *F* acts at its centre. The Maximum normal stress acting on the beam is given by

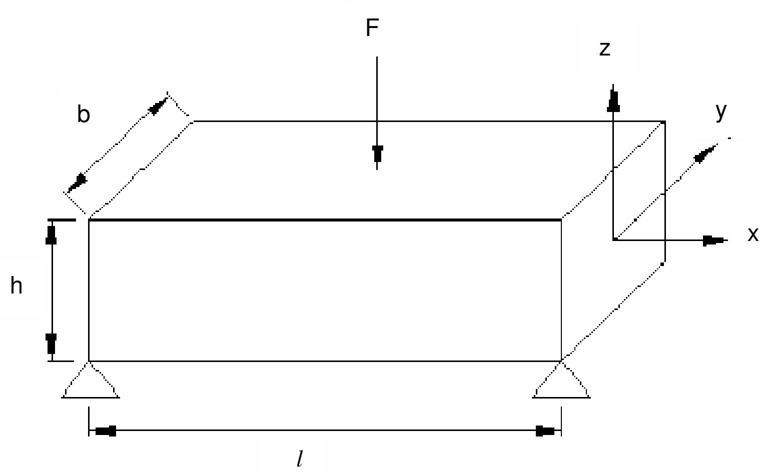


Figure 1 : Simply Supported Beam

$$\sigma_{max} = \frac{B_{max} * D_{max}}{I_n} \qquad \ldots (1)$$

where, B_{max} B= Maximum bending moment $= \frac{Fl}{4}$

D_{max} = distance of outermost fiber from the neutral axis $= \frac{h}{2}$

I_n = Moment of inertia of the beam section about the neutral axis $= \frac{bh^3}{12}$

On substituting these values in Eq. (1), σ_{nmax} changes to

$$\sigma_{max} = \frac{\frac{Fl}{4} * \frac{h}{2}}{\frac{bh^2}{12}} = \frac{3}{2}\frac{Fl}{bh^2} \qquad \ldots\ldots (2)$$

The permissible normal stress under tension for the beam material is given by

$$\sigma_{per} = \frac{3}{2}\frac{Fl}{bh^2} \qquad \ldots\ldots (3)$$

Or minimum volume of material (V_{min}) required to make sure that beam has sufficient strength is given by

$$V_{min} = b \times h \times l$$

$$= \frac{3}{2} * \frac{Fl}{\sigma_{per}} * \frac{l}{h} \qquad \ldots\ldots (4)$$

The maximum deflection of simply supported beam is given by the following expression :

$$d_{max} = \frac{FI^3}{48EI_n} \qquad ..... (5)$$

Where E is young modulus of beam material

If the deflection of the beam d_{per} is not to exceed a permissible value, then

$$d_{max} = \frac{FI^3}{48EI_n} = \frac{FI^3}{48E\frac{bh^3}{12}} \qquad ..... (6)$$

or $$V_{max} = b * h * l$$

$$= \frac{F}{4Ed_{per}}.(\frac{I^2}{h})^2 \qquad .....(7)$$

Where V_{min} = minimum volume of metal required to make sure that deflection of the beam under load does not exceed the permissible value.

The condition of optimum design is given by

$$V\text{max} = V\text{min}$$

$$= \frac{3}{2} * \frac{F}{\sigma_{per}} * \frac{I^2}{h} = \frac{F}{4Ed_{per}}(\frac{I^2}{h})^2 \qquad .....(8)$$

Hence Eq. (8) indicates that for every structure, there exists an optimum ratio $\frac{I}{h}$ and the ratio $\frac{I}{h}$ depends upon:

(a) Operation constraint i.e. d_{per}.

(b) The material of the structure i.e. E and σ_{per}.

3.1 Materials for Machine Tool Structure

The commonly used material for machine tool structures are cast iron and steel. Earlier cast iron structures were widely used but due to advances in welding technology, welded steels are widely used now days.

The selection of material for machine tool structure depends upon following factors:

Material properties

(a) Cast iron has higher damping properties than steel. Welded steel also shows good damping properties.

(b) Cast iron has better sliding properties.

(c) Steel has higher strength under static and dynamic loading.

(d) The unit rigidity of steel under tensile, torsional and bending loads is higher than cast iron.

Manufacturing Problems

Welded structures of steel have much thinner wall thickness as compared to cast structure. Walls of different thickness can be welded more easily than casting it.

MODULE 2

Machining allowances for cast structures are generally greater than for weld steel structures. Machining allowance is necessary in casting to remove defects such as inclusions, scales, etc. Welded structure can be easily repaired as compared to cast structure.

Economy

The selection of material for structure will also depend upon its cost. The weight of steel is lesser and but actual metal consumption is higher than that of cast iron. Hence in such cases the cost increases. Holes are obtained with the help of core in the casting structure but holes are made in welded steel structure by machining. These will not only increase the material cost but also increases labour cost. Cost of patterns, welding fixtures, and cost of machining are considered while selecting material for structure.

On considering above factors, the cast iron and steel may be used for following application :

(a) Cast iron should be used for complex structure subjected to normal loading which are to be produced in large number.

(b) Steel should be used for simple and heavy loaded structures which are to be produced in small number.

(c) Combined welded steel and cast iron should be used where steel structure is economically suitable. Example: Cast bearing housings that are welded into the feed box.

SAQ 2:- 1. Derive expression for design of machine tool structure

2. Explain the design criteria for selection of material for machine tool structure.

3. DESIGN OF BEDS

The machine tool beds consist of partially or fully closed box sections with ribs, partitions, etc. Beds are usually used in machine tools with wall arrangements and are evaluated as bars subjected to bending and torsion. This arrangement is shown in Table 1.

Table 1 : Bed Sections and Wall Arrangements with Their Application

Sl. No.	Wall Arrangement	Application
1	Covered top closed profile bed	These are used in boring, Plano-milling and slotting machines.
2	Open top closed profile bed.	These are used in grinding machines. They are also used when the bed is also required to serve as an oil reservoir.
3	Beds on legs: without stiffening diagonal wall.	These are used in lathes, turrets etc.
4	Beds on legs: without stiffening wall with 30-40% higher stiffness than (3).	These are used in multiple tool and high production lathes.

5	With stiffening wall and provision of chip disposal through opening in rear wall	These are employed in large sized lathes and turrets.
6	With stiffening wall	These are used in large size lathes and turrets.

The deflection of bar depends upon the product of young's modulus of material (E) and moment of inertia about neutral axis (I_n) and angle of twist depends upon the product of modulus of rigidity of material (G) and torsional moment of inertia I_t for a given compound loading. The deflection and twist is resisted by G and E. If the values of GI_t and EI_n are larger, the deflection and twist of the bar will be smaller.

The beds have perpendicular or diagonal stiffeners in every arrangement mentioned in Table 1. The reduced bending rigidity of a bed with diagonal stiffeners is given by the equation:

$$EI_r = p_2 \,.\, EL^2 A_c \qquad \ldots (9)$$

The reduced bending rigidity of a bed for bending in horizontal plane, which have two walls and perpendicular stiffeners is given by the equation

$$E\, I_r = p_1 \,.\, EI_{\min} \qquad \ldots (10)$$

Where E = young's modulus of the bed material, kgf/cm^2,

L = length of the bed that undergoes deformation,

A_c = area of cross section of the wall, cm^2,

$I_{\min}$ = moment of inertia of the wall cross section in the plane of minimum rigidity against bending, cm^4,

$E \,.\, I_r$ = reduced bending rigidity of the bed, kgf.cm^2, and

p_1, p_2 = coefficients that depend upon the arrangement of stiffeners. The value of p_1 and p_2 are given in Table 2 .

Constant, $\square_1 = 1 + \frac{36\xi}{\varphi^2}$

Constant, $\square_2 = \frac{3+4\eta}{3+\eta} + \frac{36\xi}{\varphi^2}\left[1 + \frac{9\varphi\mu}{(3+\eta)^2}\right]$

Constant, $\square = \frac{1}{\varphi}\left[\frac{I_{min}}{I_{smin}} + 36\xi\mu\right]$

Constant, $\square = \frac{A_c}{A_s}$

Constant, $\square = \frac{L}{B(n+1)}$

Constant, $\square = \frac{I_{min}}{2\ Ac\ \ B}$

MODULE 2

θ = half of the angle between diagonal stiffeners

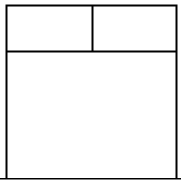
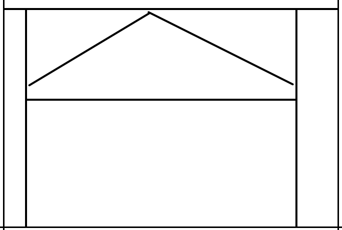
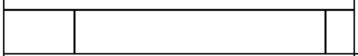
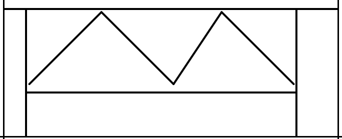
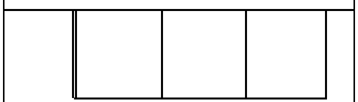
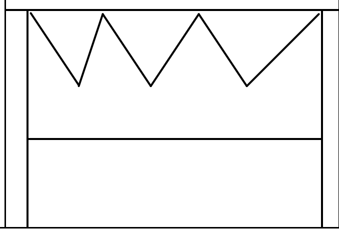
Beds with Diagonal Stiffeners			Beds with Vertical Stiffeners		
Stiffener Arrangement	**n**	p_1	**Stiffener Arrangement**	**n**	p_2
	1	8/λ_1		2	2 sin □ cos □ 3 12[□+ sin □]
	2	4(4□1 +□2)		4	2 sin □ cos □
	3	32 □2		6	2 sin □ cos □ 3 12[□+ 6.3 sin □]

Table 2:- Coefficient of p_1 and p_2 for different Stiffeners Arrangement. Coefficient p_2 is evaluated as the arithmetic mean of values of p_2, if the number of stiffeners is different from tabulated value. These values are corresponding to the nearest available larger and smaller values of n available in the above table.

The following points should be considered for selecting stiffeners:

(a) The distance between two adjacent stiffeners in perpendicular stiffeners should be approximately equal to the width of the bed, i.e. the distance between the parallel walls.

(b) The angle between two adjacent stiffeners in diagonal stiffeners should lie in between 60° to 100°.

The reduced rigidity of the bed is determined by walls for bending in vertical plane. Perpendicular and diagonal stiffeners have no effect upon the overall bed stiffness. The reduced rigidity is evaluated by multiplying the analytical rigidity with coefficient p_3.

P_3 is given by the equation:

MODULE 2

$$P_3 = \frac{1}{1+\frac{30I_{max}}{L^2 A_v'}} \qquad \text{..... (11)}$$

Where I_{max} = moment of inertia of the cross section in the plane of maximum rigidity against bending, cm^4, and

A_v' = area of vertical portions of the all, cm^2.

The reduced torsional rigidity of a bed with perpendicular stiffeners is given by the expression,

$$GI_t = \frac{B^2 EI_{max}}{\theta_1 kL^2 + \theta_2 \frac{EI_{max}}{GA_v'}} \qquad \text{...... (12)}$$

where B = width of bed, cm,

k = coefficient which depends upon the number of stiffeners, and

θ_1, θ_2 = coefficients which depends upon the bed profile.

For a bed consisting of two parallel walls and perpendicular stiffeners

$\theta_1 = 1/6$ and $\theta_2 = 2$

Hence, reduced torsional rigidity of such bed is given by

$$GI_t = \frac{B^2 EI_{max}}{\frac{kL^2}{6} + \frac{2EI_{max}}{GA_v'}} \qquad \text{. . . (13)}$$

For a bed consisting of two vertical and one horizontal wall with perpendicular stiffeners,

$$\theta_1 = \frac{(\delta+6)}{12(2\delta+3)^2} \qquad \text{. . . (14)}$$

$$\theta_2 = \frac{3(3\delta^3+16\delta^2+48\delta+36)}{5(2\delta+3)^2} \qquad \text{. . . (15)}$$

where $\delta = B/h_v$, and

h_v = the height of the vertical walls.

For a bed consisting of two vertical walls and an inclined wall with perpendicular stiffeners :

$$\theta_1 = \frac{\delta+2}{12(2\delta+1)} \qquad \text{. . . (16)}$$

$$\theta_2 = \frac{3(5\delta_1^3+16\delta_1^2+14\delta_1+4)}{5(2\delta_1+1)^2} \qquad \text{. . . (17)}$$

where $\square_1 = \sqrt{1 + \frac{B^2}{h_v^2}}$

For Eqs. (12) and (13), the value of coefficient $k = 1$ for beds without partitions. In beds having one or more partitions perpendicular (particularly stiffeners), i.e. $n > 1$, the value of k is taken from design data book as a function of □, $\square_1$ and □. The coefficient □ is obtained from following relationship

$$\frac{1}{\rho} = \frac{L.G.I_{st}}{(n+1)BEI_{max}} \qquad \ldots (18)$$

where I_{st} is the torsional moment of inertia of the stiffener.

For beds having T-shaped walls, I_{st} is given by

$$I_st = \frac{1}{3}\beta_S^3\ (b_s\ +\ h_{\ s}) \qquad \ldots (19)$$

Where b_s is width of stiffeners, cm,

h_s is height of stiffeners, cm, and

β_s is thickness of stiffeners, cm.

If the wall has square shaped section, then

$$I_{st} = \frac{0.21\beta_S h_S^3}{\frac{B^2(\delta_S+6)}{12b_S^2(2\delta_S+3)} + \frac{3\varepsilon_S^3+16\delta_S^2+42\delta_S+42}{8\delta_S^2(2\delta_S+3)^2}} \qquad \ldots (20)$$

where $\delta_s = \frac{b_s}{h_s}$

The reduced torsional rigidity of beds with diagonal stiffeners can be calculated from the expression

$$GI_t = q_1 EI_{max}\frac{B^2}{L^2} \qquad \ldots (21)$$

where q_1 is coefficient that depends upon the shape and number of stiffeners.

It may be calculated from following expressions :

$$\frac{1}{q_1} = \frac{1}{12n^2}[2 + 3\beta_1 + 6\gamma_1]$$

$$- \frac{(6\square 1 - 3\square_1 - 1)^2[(2n-1)(n-1) + n(6\square_1 + 6\square_2 + 2\square_1 - 1) - (6\square_1 + 3\square_1 - 1)]}{n(2n-1)(n-1)(6\square_1 + 6\square_2 + 2\square_1 + 2) + (6\square_1 + 3\square_1 - 1 + 3n)[n(6\square_1 + 6\square_2 + 2\square_1 - 1) - (6\square_1 + 3\square_1 - 1)]}$$

where $\square 1 = \frac{n^3 d^3 EI_{max}}{L^3 EI_{smax}}$

$$v_1 = \frac{n^2 EI_{wmax}}{L^2 GA'_w} \qquad \text{. . . (22)}$$

$$v_2 = \frac{dn^3 EI_{wmax}}{L^3 GA'_s} \qquad \text{. . . (23)}$$

where I_{smax} = moment of inertia of the stiffener in the plane of maximum rigidity against bending, cm^4,

d= length of the diagonal stiffener , cm, and

A'_s= area of the vertical position of the stiffener, cm^2

The reduced torsional rigidity of a rectangular box type section is given by the relation:

$$GI_t = G.\ 4A_0^2 \sum_{i=1}^{n} \frac{\delta_i}{s_i} \qquad \text{. . .(24)}$$

SAQ 3

(a) Give details of bed section and wall arrangement with their application.

(b) Explain design of beds?

(c) What are the various considerations while selecting stiffeners?

4. DESIGN OF COLUMNS

The spindle head is mounted on the column in machine tool with fixed bed. The spindle head and knee table unit is mounted on the column in the knee type machine tool. The forces are acted on the columns in the symmetrical plane, e.g. drilling machine. The forces are also acted arbitrarily in space, e.g. milling and boring machine, vertical lathe etc. The principle design requirements of columns are high static and dynamic stiffness. These properties are achieved by proper selection of the column material and its cross-section. Sections of machine tool columns are shown in Table 3.

Table 3 : Commonly used Column Sections and Their Application

Sl. No.	Section	Application
1	Square box type section with vertical ribs and	It is used for columns subjected to three dimensional loading, chief

	spaced horizontal stiffeners.	application is in boring and milling machines.
2	Rectangular box type section.	It is used when forces act only in the plane of symmetry. It is used in vertical drilling and unit built in machine tools. Optimum a/b = 2 to 3
3	Rectangular box type section with vertical ribs and spaced horizontal stiffeners.	It is used in gantry type machine tools. The recommended a/b ratios are: For vertical lathe a/b = ¾, for planning machine a/b = 2-3, for plano-mailing machine a/b ratio = 2-3.
4	Circular section with stiffeners.	It is used when the load is small and it is necessary to provide for rotation of the column, e.g. in radial drilling machine and in bench.

The columns are generally made with thin walled circular or box sections. The section is made stronger by providing stiffeners. Number and size of holes and opening is kept as small as possible. Warping may takes place, if the height of the column is greater than its cross sectional dimensions under three dimensional loading. Hence warping is avoided by providing torsional stiffeners. These stiffeners are not employed in the absence of warping as they marginally improve overall section stiffness. The area of column gradually increases from top to the base where the bending moment is high. The section should be progressively changed from thin walled rectangle to a thin walled square.

The machine tool columns are experienced bending in two perpendicular planes, shearing in two perpendicular planes and torsion. The deflection in bending of the column is calculated by analyzing the column as a cantilever fixed at the base, and plotting bending moment diagram in both the directions.

The deflection due to shearing of the column is determined for following expression :

$$d = \omega \frac{F_S.I}{G.A} \qquad \ldots (25)$$

where F_s = shearing force, N,

l = distance from the base of the section in which shearing deformation is being determined, mm,

A = area of cross-section, mm^2,

G = Shear modulus of column material

□ = coefficient of distribution of shearing displacement

The value of coefficient □ can be calculated from Figure 2.

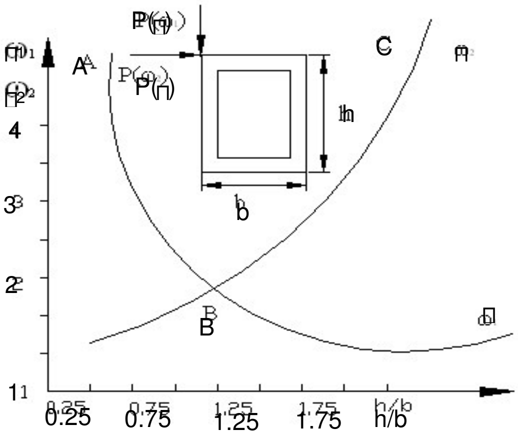


Figure 2 : Variation of Coefficient of Distribution of Shearing Displacement

The displacement of the guideway (point A) of a rectangular box type section (as shown in Figure 14.3) due to torsion may be calculated from the following equations :

In direction *X*-*X*, deflection $\square X = \bar{y}.\ \square$

In direction *Y*-*Y*, deflection $\square Y = \frac{b}{2}.\ \square.$

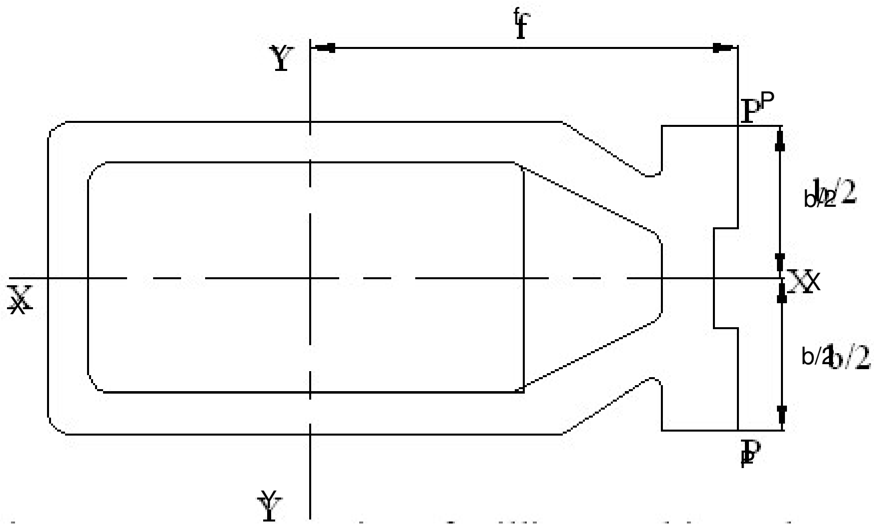


Figure 3: Cross-section of Milling Machine Column

The deflection of the base to which the column is bolted should also be taken into consideration. The base is used as a hollow box section for this purpose which is simply supported at the ends. The deflection at the top of the column due to bending of the base is obtained by multiplying the column height with the angle of slope of the deflected base in that section where it meets the column. The dimension of the column cross-section should be such that the total deflection of the Guideways at the top of the column doesn't exceed 3 to 5 µm/meter length in *Y*-*Y* direction and 10 to 25 µm/meter length in *X*-*X* direction.

MODULE 2

If holes are closed by cover plates, the reduction in torsional stiffness due to the openings can be neglected provided each force is tightened by the force, F_t.

$$F_t \geq \frac{T(b_a + l_a)}{A_b \mu n} \qquad \ldots (26)$$

Where F_t = tightening force of each bolt, N,

T = Torque acting on the column, N.mm,

μ = coefficient of friction,

n = number of bolts, and

A_b = area of the cross section of the one bolt, mm^2.

The effect of ribs and stiffeners on the bending stiffness of column is very small and hence can be neglected. The vertical ribs also don't have any appreciable effect on the torsional stiffness.

SAQ 4

(a) State and explain various column sections with their application.

(b) Explain design of columns.

5. DESIGN OF HOUSING

Housing is one of the important elements of machine tool structure. Housing may be split or solid. Solid housings are used in small and medium sized machine tools. Split housings are easier to assemble but stiffness is less as compared to solid one. Split housings are provided with a hinged cover to facilitate its opening for regulation of some mechanisms such as the speed box engine lathes with cone pulley drive. The stiffness of such housing is less than solid housing by 50 %. Housing type structures are designed for stiffness. The stiffness is determined by determining the displacement of point C. This displacement is due to a force acting normal to the wall at the same point. The housing type structure is shown in Figure 14.4.

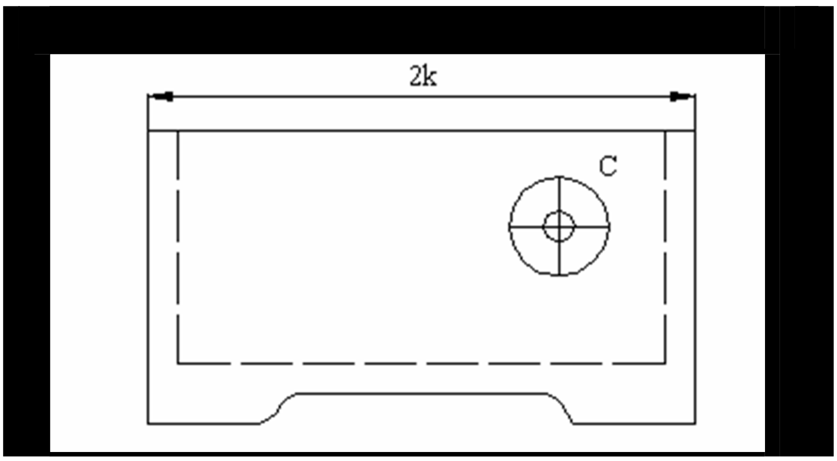


Figure 14.4 : Typical Housing Structure

This displacement is calculated from following empirical relation :

$$X = g_0 g_1 g_2 g_3 \cdot \frac{F.k^2(1-\mu)}{E.t^3} \qquad \ldots (27)$$

where g_0 = coefficient that accounts for the type of connection of the loaded wall with adjoining walls.

g_1 = coefficient that accounts for the effect of the bossing of the loaded hole.
g_2 = coefficient that accounts for the effect of unloaded holes and bossing.

g_3 = coefficient that accounts for the effect of the ribs and stiffeners.

F = force acting normal to the loaded wall, N.

t = thickness of loaded wall without bossing, mm.

$2k$ = the larger dimension of the rectangular loaded wall, mm.

μ = Poisson's ratio for the housing material.

E = Modulus of elasticity of the housing materials, N/mm^2.

The coefficient g_0 depends upon the ratio of 2k : 2m : 2n of the housing dimensions and point of application of the force F. Different values of g_0 is given in Table 4 for different values of k:m:n when force F is acting at the center of the loaded wall. The value of g_0 is high for the force acting at the center when the loaded wall is connected to adjoining walls on all four sides. The value of g_0 is two to three times smaller for loads acting at the corners. When the loaded wall is connected to adjoining walls only on three sides, the value of g_0 reduces as the point of application shifts towards the constrained corners.

Table 4 : Values of Coefficient g_0 for Housings

Ratio a : b : c	**All Four Edges of Loaded Wall 2k × 2m Connected to Adjoining Walls**	**Three Edges of Loaded Wall 2k × 2m Connected to Adjoining Walls, One Edge Free**
1 : 1 : 1	0.35	0.48
1 : 1 : 0.75	0.44	-
1 : 1 : 0.5	0.5	-
1 : 0.75 : 1	-	0.45
1 : 0.75 : 0.75	0.3	0.42
1 : 0.75 : 0.5	0.33	-

1 : 0.5 : 1	-	0.28
1 : 0.5 : 0.75	-	0.27

The value of g_0 is approximately 3 times less at two constrained corners. If the point of application of the force F is shifted towards the free corner, the maximum value of g_0 occurs at the middle of the free edge. The value of g_0 may be about 2.5 to 3 times greater than the tabulated values for load acting at the center.

Bosses are made to increase the wall thickness locally where the stiffness has suffered due to hole. They are usually located on the internal surface of wall. Bossing dimensions are generally limited to

$$D = (1.4 \text{ to } 1.6)\, d$$

$$H = (2.5 \text{ to } 3)\, t$$

This is because any further increase in dimension doesn't improve the stiffness. g_1 depends upon the ratios D/d, H/t, r/k. It decreases sharply with increase in the value of H/t. The different values of g_1 are given in Table 5.

Table 5 : Values of Coefficient g_1 for Housings; r/k = 1, D/d = 1.6

Hb/t	**[$D^2/(2k * 2m)$]**							
	0	0.02	0.04	0.06	0.08	0.1	0.12	0.14
1.2	1	0.95	0.94	0.92	0.9	0.88	0.87	0.85
1.4	1	0.88	0.83	0.78	0.72	0.72	0.69	0.67
1.6	1	0.82	0.75	0.68	0.6	0.6	0.57	0.55
2.0	1	0.75	0.65	0.58	0.48	0.53	0.45	0.41
3.0	1	0.7	0.58	0.5	0.38	0.44	0.35	0.3

In Table 5, H_b represents the active height of the bossing. The ratio H_b/t is taken from design data book. There is not any significant effect on the value of coefficient g_1 due to ratios D/d and r/j. The values of g_1 are generally higher than that tabulated by
5 -10%. The effect of r/k is not uniform. The reduction of ratio r/k to 0.6 has no effect on the value of g_1 for average values of [H_b/t ($D^2/(2k * 2m)$)] = 0.8 and H_b/t = 1.5] and [($D^2/(2k * 2m)$)]. The coefficient g_1 increases by two to five percent for low value of [($D^2/(2k * 2m)$)] while it reduces by two to five percent for high values of [$D^2/(2k * 2m)$] as compared to tabulated values.

The coefficient $$g_2 = 1 + \sum\left(\frac{\Delta X_i}{X}\right) \qquad \ldots (27)$$

Where Δx_i shows the increment in deflection due to i^{th} unloaded hole. Its values depend upon the ratios of D/d, H_b/t.

The values of coefficient g_2 are given in Table 6.

Table 6 : Values of Coefficient g_2 for Housings; r/k = 0.5, D/d = 1.6

Hb/t	**($D^2/(2k * 2m)$**					
	0	0.02	0.04	0.06	0.08	0.1
1.2	1	0.99	0.99	0.98	0.98	0.98
1.4	1	0.96	0.94	0.92	0.90	0.84
1.6	1	0.93	0.90	0.88	0.86	0.84
2.0	1	0.92	0.88	0.85	0.82	0.80
3.0	1	0.9	0.85	0.80	0.76	0.74

If coefficient g_2 increases by 5 to 10 %, ratio of D/d decreases from 1.6 to 1.2. The effect of R/k' on the value of g_2 depends greatly upon [$(D^2/(2k * 2m)$] and H_b/t. Here a' represents the distance between end of wall to the point at which load F is acting. The value of g_2 increases by 15 % with increase in the value of R/k' for average values of [$(D^2/(2k * 2m)$] and H_b/t. The value of g_2 decreases by 15 to 20 % if R/k' reduced to 0.3. The coefficient g_3 = 0.8 to 0.9, if the stiffening ribs are cast in the surrounding area of the loaded hole to increase the strength of bossing and wall. If the ribs are provided for a general overall improvement of housing stiffness, then g_3 = 0.75 to 0.85. The larger values are for non-intersecting ribs while smaller values are for interconnected ribs.

Hence, it is clear from Eq. (27) that thickness housing wall is most important factor that calculate overall deformation x. Hence bossing of hole is most important step to provide sufficient stiffness to housing type structures.

SAQ 4

Explain design of housings.

6. SUMMARY

Machine tool structure consists of beds, bases, columns, box type housings, overarms, carriages, tables etc. The structures are used to hold subassemblies, to support and move the cutting tool and workpiece. The machine tool structures are designed for high wear resistance of guiding and guided and surface high static and dynamic stiffness. This unit particularly deals with various aspects in designing bed, column and housings.

7. KEY WORDS

Bed : It is the base of machine tool on which whole assembly is mounted.

Column : The spindle head is mounted on the column in machine tool with fixed bed.

MODULE 2

MODULE 3

DESIGN OF GUIDEWAYS AND SPINDLE

Structure

1 INTRODUCTION

Design of machine tool elements is critical in tool engineering. They must withstand against applied external load. The machine tool elements such as guideways, slideways and spindle unit are discussed in detail in the next section. Additionally, requirements, functions and types of guideways and spindle are also explained.

Objectives

After studying this unit, you should be able to understand

- various types and functions of guideways,
- the design of slideways,

MODULE 3

- the design of spindle, and
- Function of spindle.

MODULE 3

In machines, guideways help to guide the tool or workpiece along a predetermined path, usually either a straight line or a circle [6]. Guideways, lubrication and drive systems are discussed in the next section and form an important part of ultra-precision machines. There are basically two types of guideways—friction guideways and anti-friction or hydrostatic guideways. Friction guideways were initially used but are now replaced by hydrostatic guideways in precision and ultra-precision machine tools. A guideway should be highly accurate, durable and rigid. Machine tools require guideways for guiding the movement of the workpiece and for positional adjustment.

The designing of guideways for tables, saddles and cross-slides involves the following aspects [7]:

- Shapes of the guiding elements and arrangements of their combinations
- Effect of material and working conditions upon the guiding accuracy (wear)
- Friction conditions and load carrying capacity (roller bearings and lubrication)

According to Koenigsberger [7], a good guideway design is needed to satisfy the following requirements:

- Provision of an exact alignment of the guided parts in all positions and under the effect of the operational forces
- Provision of a means for compensating possible wear
- Ease of assembly and economy in manufacture (possibility of adjusting the alignment in order to allow for manufacturing tolerances)
- Freedom from restraint
- Necessary prevention of chip accumulation and ease of removal of any chips
- Effective lubrication must be possible

In order to achieve a good wear resistance, the pressure distribution must be uniform. The most commonly used guideway materials are cast iron and durobar steels. Different types of profiles may be employed for different applications. Guideways are also classified into two groups, one with external and the other with internal features. The most common is the prismatic symmetric guideway, which is well suited for obtaining a very accurate movement of parts (Figure 5.12). It has the characteristic of self-aligning during wear. The external prismatic guideway enables easy removal of chips, while the internal type offers a good lubricant retention. For uneven pressure distribution, the prismatic symmetric type can be modified into the prismatic unsymmetric (asymmetric) type for the same operating characteristics (Figure 5.13). The internal prismatic unsymmetric type is normally used for rotary applications. These guideways are capable of automatic adjustment because of the action of gravity, which keeps the surfaces in contact [8].

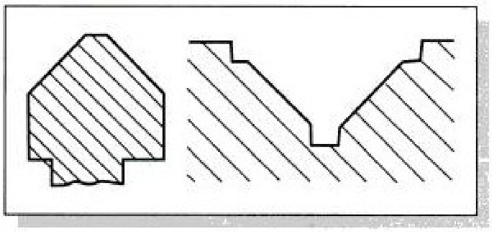


Fig. 5.12: External and internal prismatic symmetric guideways [6].

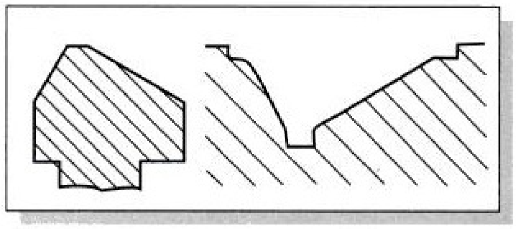


Fig. 5.13: External and internal prismatic unsymmetric guideways [6].

In conventional machines, internal and external flat guideways are suitable for normal accuracy requirements (Figure 5.14). The setting involves straight or tapered gibs. Generally, it requires good workmanship and proper protection from chips. On the other hand, the dovetail is used when there is a limitation on the height of the guideways (Figure 5.15). It is not suitable where forces tend to pull out the guides. Finally, the circular guideway is well suited for axial loading and is relatively easy to manufacture (Figure 5.16).

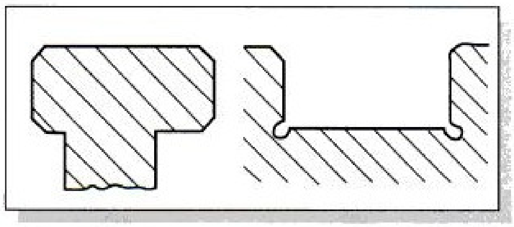


Fig. 5.14: External and internal flat guideways [6].

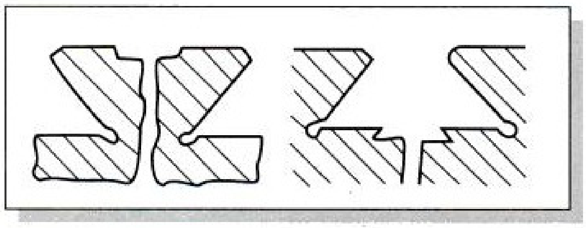


Fig. 5.15: External and internal dovetail guideways [6].

In general, internal guideways are chosen when the sliding velocity is high, and it is essential to provide a good retention of the lubricant at the interface. On the other hand, when the sliding velocity is not that high and it is necessary to prevent chip accumulation and to ensure its easy removal, the external guideway is preferred. In ultra-precision machines, it is common to use bellows for protection against chips and dirt.

MODULE 3

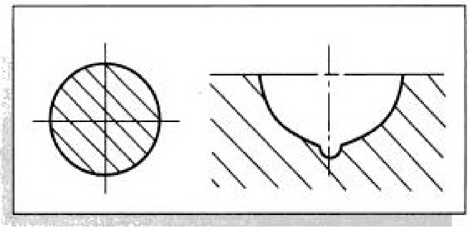


Fig. 5.16: External and internal circular guideways [6].

These guideways require proper lubrication to avoid a high coefficient of friction between the sliding surfaces. This can result in significant wear, reducing the life as well as badly affecting the machining accuracy. Proper lubrication can be provided either manually or by creating a groove along the longitude and the latitude of the guideway path for auto lubrication purposes. While grease and oil are employed in conventional machines, guideways with aerostatic, hydrostatic and even rolling elements are essential for precision applications. Guideways using rolling elements are not common in ultra-precision machines but are used very effectively in the Toshiba ultra-precision machine as shown in Figure 5.4 (b), particularly for grinding applications.

Table 5.1 discusses the suitability of different guideway systems for various guiding element properties. It has been observed that the hydrostatic system is the most suitable system for guiding elements as it has exceptional properties such as straightness, positional error, wear, load carrying capacity, static stiffness and dynamic behavior, whereas rolling elements are much more economical [3]. The straightness characteristic is defined by the use of differential equations of the first, second and third orders. The weakness of the rolling element guideway is thus demonstrated. Wear and

Table 5.1 Properties of different guideways for precision application [3]

Characteristics of guideways for high- and ultra-precise applications	Aerostatic guideway	Hydrostatic guideway	Rolling element guideway
1. and 2. order straightness	●	●	◕ - ●
3. order straightness	●	●	◑ - ◕
Position error (Step-response test)	●	●	●
wear	●	●	◑
load capacity	◕	●	●
static stiffness	◕	●	●
dynamic behaviour	◕	●	◑
price/cost	◑	◔	●

● ◕ ◑ ◔ ○

very good ⟷ poor

dynamic behaviour are other negative factors. Therefore, in ultra-precision machines, the hydrostatic guideway is often preferred.

Typical lathe machining operations shown in Figure 5.17 are a clear example of the application of guideways. Guideways are used to guide the carriage and tailstock to the required position along the pathway of the lathe machine. The type of the guideway used here is one that has a prismatic and one with a flat external shape. The advantages of this guideway combination are that it is easy to manufacture and has a greater accuracy of travel. There is another type of guideway in the cross slide-carriage application, which is of the dovetail type. This arrangement is used in this application because the height of the guideways is comparatively small due to carriage height limitation. The dovetail is preloadable resulting in a high stiffness in all directions. Furthermore, wear occurs usually symmetrically and does not affect the alignment of the carriage [9].

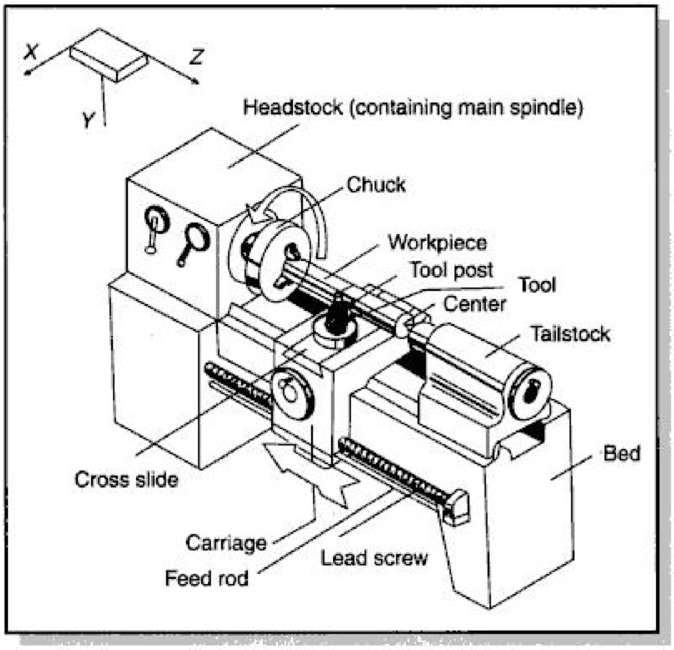


Fig. 5.17: The conventional lathe with an inverted prismatic and flat external guideway [9].

Figure 5.18 shows a typical open rectangular (T-shaped) configuration, which is commonly seen in machine tools. It provides a very high stiffness and has symmetrical wear. The open rectangular configuration is able to support machines with a 5–10 µm repeatability [9].

Figure 5.19 shows the possible combination of different types of guideways on the base of the machine tools. Figure 5.20 shows a lathe bed section, showing inverted prismatic symmetric and flat

MODULE 3

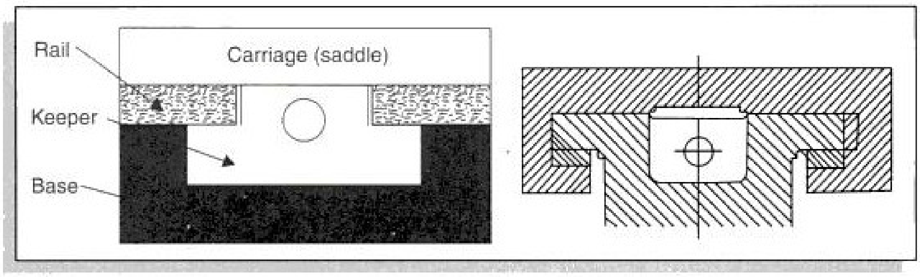


Fig. 5.18: An open rectangular or T-shaped configuration [9, 10].

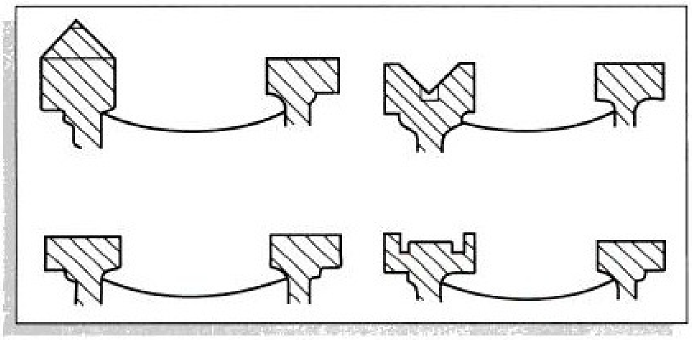


Fig. 5.19: Possible combinations of different types of guideways for conventional machines.

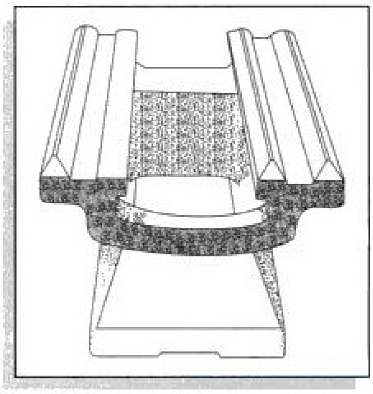


Fig. 5.20: A lathe bed section showing inverted prismatic and flat-type ways.

type guideway combinations, and Figure 5.21 shows an example of a guideway employing rolling elements to reduce friction. In addition, Figure 5.22 illustrates a precision lathe with a guideway containing foundry sand for enhanced stability.

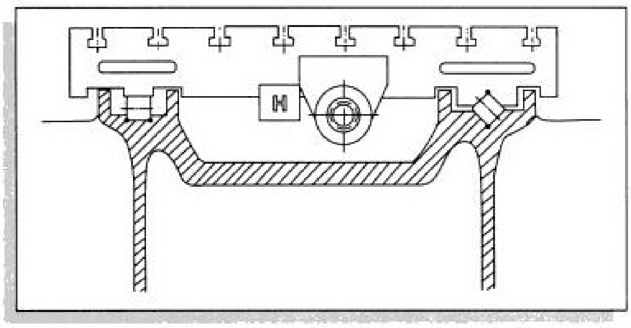


Fig. 5.21: Friction reduction in a conventional bedway is improved by using normal and cross roller bearings.

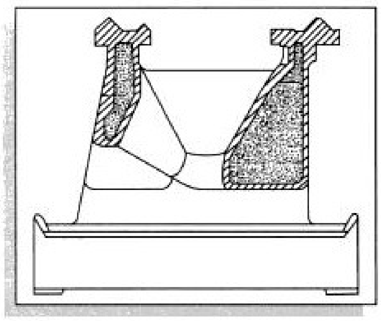


Fig. 5.22: A precision lathe with a bedway containing foundry sand for an enhanced stability.

In ultra-precision machines, the slideways utilize a fully constrained and preloaded hydrostatic oil bearing design to provide a high degree of stiffness, vibration, damping, smoothness of motion and geometrical accuracy. For the Precitech ultra-precision machine discussed earlier, the slideways are capable of a slide position feedback resolution of 8.6 nm, which is provided by an ultra-fine pitch low expansion glass scale. The slide has a horizontal straightness of between 0.2 μm and 0.3 μm. This extreme accuracy is only achievable through the use of hydrostatic oil bearings. The guideways employed in these machines are either box shaped or dovetail shaped as shown in Figure 5.23 and 5.24, respectively.

Fig. 5.23: A hydrostatic box type guideway for an ultra-precision machine [4].

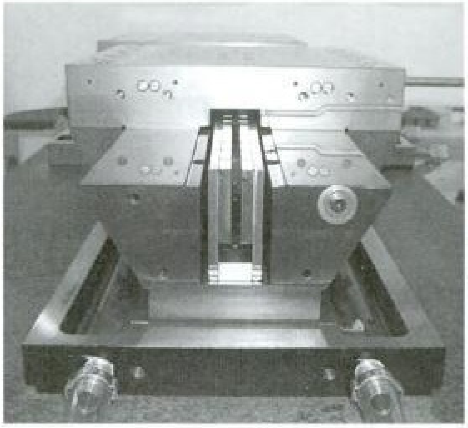


Fig. 5.24: An ultra-precision lathe with a hydrostatic dovetail guideway [4].

2 FUNCTIONS OF GUIDEWAYS

The Guideways is one of the important elements of machine tool. The main function of the Guideways is to make sure that the cutting tool or machine tool operative element moves along predetermined path. The machine tool operative element carries work piece along with it. The motion is generally circular for boring mills, vertical lathe, etc. while it is straight line for lathe, drilling, boring machines, etc.

Requirements of Guideways are:

(a) Guideways should have high rigidity.

(b) The surface of Guideways must have greater accuracy and surface finish.

(c) Guideways should have high accuracy of travel. It is possible only when the deviation of the actual path of travel of the operative element from the predetermined normal path is minimum.

(d) Guideways should be durable. The durability depends upon the ability of Guideways to retain the initial accuracy of manufacturing and travel.

(e) The frictional forces acting on the Guideways surface must be low to avoid wear.

(f) There should be minimum possible variation of coefficient of friction.

(g) Guideways should have good damping properties.

SAQ 1

(a) What are various functions of Guideways?

(b) State requirements of Guideways.

3 TYPES OF GUIDEWAYS

The Guideways are mainly classified according to the nature of friction between contacting surfaces of the operative element:

(a) Guideways with sliding friction

(b) Guideways with rolling friction

3.1 Guideways with Sliding Friction

The friction between the sliding surfaces is called as Guideways with sliding friction. These Guideways are also called as slideways. The slideways are further classified according to the lubrication at the interface of contacting surfaces. The friction between the sliding surfaces may be dry, semi-liquid, and liquid. When the lubrication is absent in between contacting surfaces, it is called as dry friction. Dry friction is rarely occurred in machine tools.

When two bodies slide with respect to each other having lubrication between them, the sliding body tends to rise or float due to hydrodynamic action of the lubricant film. The principle of slider is shown in Figure 1.

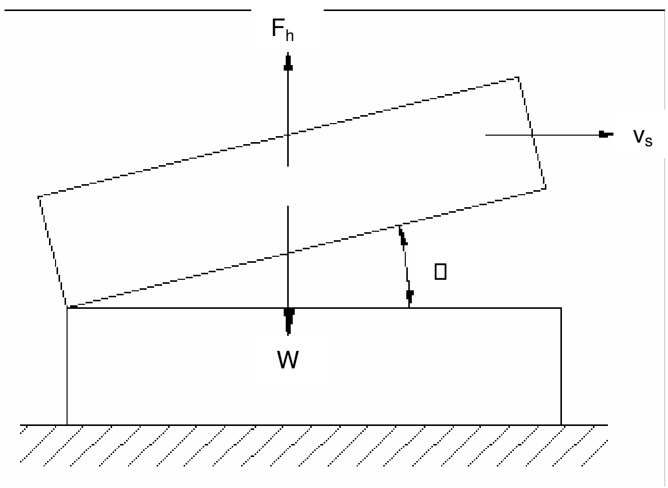


Figure 1 : Principle of a Slider

The hydrodynamic force,

$F_h = C * V_s$ . . . (1)

Where C is constant and depends upon wedge angle θ, the geometry of sliding surfaces, viscosity of the lubricant and parameter of lubricant film.

V_s is sliding velocity.

W is weight of the sliding body.

The resultant normal force acting on sliding body,

$$R = F_h - W$$

From Eq. (13.1), it is clear that the hydrodynamic force increases with increase in sliding velocity. The sliding body rests on the stationary body when hydrodynamic force is less than the weight of the sliding body. Here, there are semi-liquid type friction conditions and under these conditions the two bodies are partially separated by the lubricant film and partially have metal to metal contact. The resultant normal force on sliding body starts to act upwards and the body floats as hydrodynamic force is greater than the sliding weight of the body. The sliding surfaces are completely separated by the lubricant film and liquid friction occurs at their interface. The slideways in which the sliding surfaces are separated by the permanent lubricant layer are known as hydrodynamic slideways. This permanent lubrication layer is due to hydrodynamic action. A permanent lubricant layer between the sliding surfaces can be obtained by pumping the liquid into the interface under pressure at low sliding speed. The sliding body is lifted by this permanent lubricant layer. Such slideways are called as hydrostatic slideways.

3.2 Guideways with Rolling Friction

These are also called as anti friction ways. The anti-friction slideways may be classified according to the shape of the rolling element as :

(a) Roller type anti friction ways using cylindrical rollers.

(b) Ball type anti friction ways using spherical balls.

SAQ 2

(a) Explain various types of Guideways.

(b) Explain with figure the principle of slider.

4 DESIGN OF SLIDEWAYS

Slideways are designed for wear resistance and stiffness.

4.1 Design of Slideways for Wear Resistance

The wear resistance of slideways is mainly dependent upon maximum pressure acting on the mating surfaces. This condition may be given as

$$p_{max} \leq P_{mp} \qquad \ldots (2)$$

Where p_m = maximum pressure acting on the mating surface, and

p_{mp} = permissible value of the maximum pressure.

It is seen during the subsequent analysis that slideway designed for maximum pressure is quite complicated. Sometimes, this design is replaced by a simple procedure based upon the average pressure acting on the mating surfaces. The condition is that:

$$P_a \leq P_{ap} \qquad \ldots (3)$$

Where p_a = average pressure acting on the mating surface, and
p_{ap} = permissible value of the average pressure.

Hence from Eqs. (2) and (3), the design of slideways for wear resistance requires that

(a) p_m and p_a to be known,

(b) p_{mp} and p_{ap} to be known, and

(c) The values of p_{mp} and p_{ap} are given for different operating conditions of slideways on the basis of experience. For determining p_m and p_a, the first and foremost task is to determine the forces acting on the mating surfaces.

Forces acting on the mating surfaces in combination of *V* and flat slideways.

The combination of *V* and flat slideways is commonly used in lathe machines. The schematic diagram of slideways and the forces acting on the system for the case of orthogonal cutting are illustrated in Figure 2.

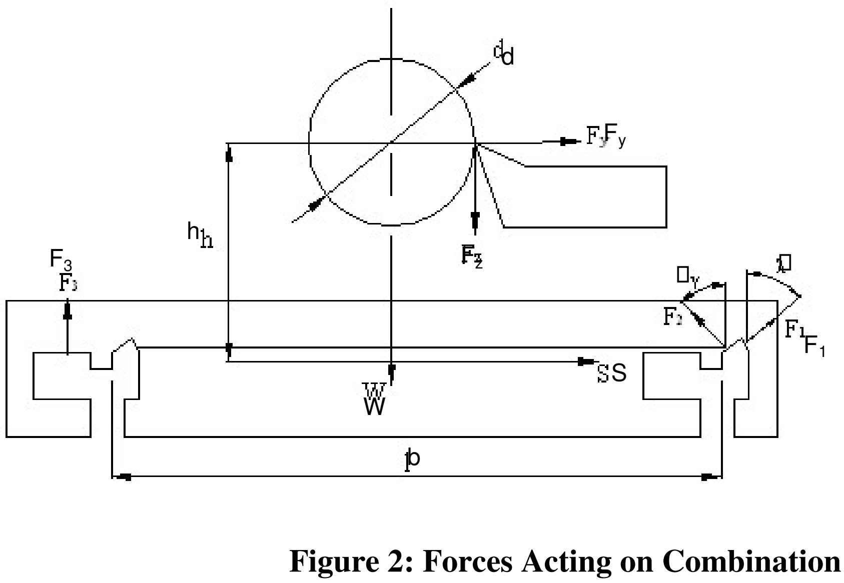


Figure 2: Forces Acting on Combination of *V* and Flat Sideways

MODULE 3

The forces acting on V and flat slideways are:

(a) Cutting force component F_z (in the direction of the velocity vector) and F_y (radial),

(b) Weight of carriage W, and

(c) Unknown forces F_1, F_2 and F_3 acting on the mating surfaces.

The unknown forces are calculated from following equilibrium conditions: Sum of components of forces acting along Y-axis = 0

i.e. $\sum Y = 0$

$$F_1 \sin \lambda - F_2 \sin \gamma + F_y = 0 \qquad \ldots (4)$$

Sum of components of forces acting along Z-axis = 0

i.e. $\sum Z = 0$

$$F_1 \cos \lambda + F_2 \cos \gamma + F_3 - W - F_z = 0 \qquad \ldots (5)$$

Moment of all forces about X-axis = 0

i.e. $\sum M_x = 0$

$b \quad b \quad d \quad b$

$$F_1 \cos . \frac{b}{2} + F_2 \cos \gamma . \frac{b}{2} - F_z . \frac{d}{2} - F_y . h - F_3 . \frac{b}{2} = 0$$

$$\therefore \quad F_3 = F_1 \cos \lambda + F_2 \cos \gamma - F_z . \frac{d}{b} - F_z \frac{d}{b} - F_y \frac{2h}{b} \qquad \ldots (6)$$

Substituting value of F_3 in Eq. (13.5), we get,

$$F_1 \cos \lambda + F_2 \cos \gamma + F_1 \cos \lambda + F_2 \cos \gamma - Fz \frac{d}{b} - Fy \frac{2b}{h} - Fz - W = 0$$

or $\quad 2 (F_1 \cos \lambda + F_2 \cos \gamma) = F_z \left(1 + \frac{d}{b} \right) + F_y \frac{2h}{b} + \frac{W}{2}$

$$\therefore \quad F_1 \cos \lambda + F_2 \cos \gamma = \frac{F_z(d+b)}{2b} + F_y \frac{h}{b} + \frac{W}{2} \qquad \ldots (7)$$

If the apex angle of the V is 90°, and assume that present angle γ may change to $\gamma = 90 - \lambda$, the solution of simultaneous algebraic Eqs. (4) and (7) gives :

$$F_1 = \frac{F_z(d+b)}{2b} \cos \lambda + F_y \frac{h}{b} \cos \lambda - F_y \sin \lambda + \frac{W}{2} \cos \lambda \qquad \ldots (8)$$

$$F_2 = \frac{F_z(d+b)}{2b} \cos \lambda + F_y \frac{h}{b} \cos \lambda + F_y \sin \lambda + \frac{W}{2} \cos \lambda \qquad \ldots (9)$$

Substituting the values of F_1 and F_2 in Eq. (6) we get,

$$F_3 = \frac{F_z(d-b)}{2b} - F_y\frac{h}{b} + \frac{W}{2} \qquad \ldots (10)$$

Eq. (10) represents the forces acting on the mating surfaces in combination of two flat sideways.

The schematic diagram of the slideways and the forces acting on the system under orthogonal cutting conditions are shown in Figure (3).

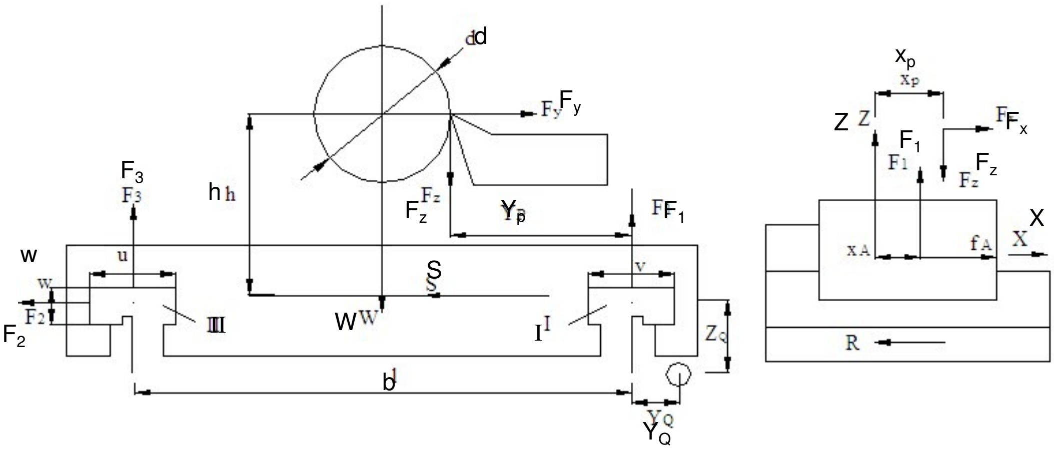


Figure 3 : Forces Acting on Combination of Two Flat Slideways

The forces acting on combination of two flat slideways are:

(a) Cutting force components, i.e. axial F_x, radial F_y, and F_z in the direction of velocity vector.

(b) Weight of carriage, W.

(c) Unknown forces F_1, F_2 and F_3 acting on the mating surfaces.

(d) Frictional forces μF_1, μF_2, μF_3, where μ is the coefficient of friction between the sliding surfaces.

The unknown forces F_1, F_2 and F_3 are calculated from following equilibrium conditions:

$$\sum X = 0$$

$$F_x + \square\,(F_1 + F_2 + F_3) - R = 0 \qquad \ldots (11)$$

$$\sum y = 0$$

$$F_2 - F_y = 0 \qquad \ldots (12)$$

$$F_2 = F_y$$

$$\sum z = 0$$

$$F1 + F3 - Fz - W = 0 \qquad \ldots (13)$$

$$\sum M_x = 0$$

$$W\frac{b}{2} + F_z y_p - Fy\ h - F_3 b = 0 \qquad \ldots (14)$$

From Eq. (14)

$$F_3 = \frac{F_z\, yp - Fy\, h}{b} + \frac{W}{2}$$

On substituting the value of F_3 in Eq. (13)

$$F1 = = F_x + \frac{W}{2} - \frac{F_z\, yp - Fy\, h}{b} \qquad \ldots (15)$$

The pulling force, R is calculated from Eq. (11) on substituting the values of F_1, F_2 and F_3.

$$R = F_x + \square\, (F_z + F_y + W) \qquad \ldots (16)$$

Determination of Average Pressure

The average pressure can be determined as :

$$pF1 = \frac{F_1}{vL}$$

$$pF2 = \frac{F_2}{wL}$$

$$pF3 = \frac{F_3}{uL}$$

where L = length of the carriage, and v, w, u = the width of slideway faces on which forces F_1, F_2 and F_3 are acting respectively.

Determination of Maximum Pressure

It is necessary to establish the points of action of the resultant normal forces F_1, F_2 and F_3 on the respective faces for calculating the maximum pressure. The distance between the point of action of normal force F_1 on the flat slideway I and the center of the carriage is denoted by x_A . This is shown in Figure 13.3. The distance between force F_2 acting on the vertical face of flat slideway II and the center of the carriage is denoted by x_B , and the distance between force F_3 acting on horizontal face of flat slideway II and the center of the carriage is denoted by x_c. For determining x_A , x_B and x_C , we have two equilibrium conditions :

$$\sum M_y = 0$$

$$F_x\, h + F_z\, x_p - F_1\, x_A - F_3\, x_c + R_z Q = 0 \qquad \ldots (17)$$

And $\sum M_z = 0$

$$F_x\, y_p + F_y\, x_p - R_{yQ} - F_2\, x_B + \square F_2 \frac{(l+u)}{2} + \square F_3\, l = 0 \qquad \ldots (18)$$

An additional equation may be written by assuming that the moment of reactive forces F_1 and F_3 about the Y-axis is proportional to the width of the slideway face, i.e.

$$\underline{F_1\, x_A} = \underline{v}$$

F_2

x_c u

On solving Eqs. (17), (18) and (19), we get, the values of x_A , x_B and x_c.

The ratio of x_A /L, x_B /L and x_c/L calculates the shape of the pressure distribution diagram and the maximum pressure on a particular face of the slideway. The procedure for determining the maximum pressure on flat slideway I is being subjected to the normal force F_1 which is described below. The most general case of pressure distribution along the length of contact L corresponds to a trapezoid as shown in Figure 4.

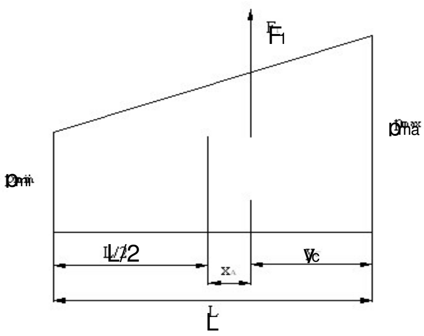


Figure 4 : Trapezoidal Pressure Distribution along Slideway Length

Force F_1 acts at the center of gravity of trapezoid. The distance y_c at the center of gravity from the larger arm of the trapezoid can be determined as :

$$\therefore \qquad y_c = \frac{p_{min} L . \frac{L}{2} + \frac{(p_{max} - p_{min})}{2} L . \frac{L}{3}}{p_{min} L + \frac{p_{max} - p_{min}}{2} L} \qquad \text{. . . (20)}$$

$$\therefore \qquad y_c = \frac{L}{3} \frac{p_{max} + 2p_{min}}{p_{max} + p_{min}}$$

As a result,

$$x_A = \frac{L}{2} - y_c$$

$$\therefore \qquad x_A = \frac{L}{6} \frac{p_{max} + p_{min}}{p_{max} + p_{min}} \qquad \text{. . . (21)}$$

Now $\qquad \frac{x_A}{L} < \frac{1}{6}$

In this case the pressure distribution diagram represents a trapezoid. This is explained with the help of following example.

Let $\qquad \frac{x_A}{L} < \frac{1}{10}$

From Eq. (13.18), we get,

$$p_{max} - p_{min} = \frac{6}{10} (p_{max} + p_{min})$$

$$p_{\min} = \frac{0.4}{1.6} p_{\max}$$

From above equation, it is clear that $p_{\min}$ is a positive, non-zero value. Hence pressure distribution diagram must be a trapezoid.

$$\therefore \quad p_{\max} + p_{\min} = 2\, p_{av} \qquad \ldots (22)$$

From Eq. (16), we get,

$$p_{\max} - p_{\min} = \frac{12\, x_A}{L} p_{av} \qquad \ldots (23)$$

On solving Eqs. (22) and (23), we get

$$p_{\max} = p_{av} \left(1 + \frac{6x_A}{L}\right)$$

4.2 Design of Slideways for Stiffness

Stiffness is one of the important parameters for designing slideways. The design of slideways for stiffness requires that the deflection of the cutting edge due to contact deformation of the slideway should not exceed certain permissible value. This value is specified from considerations of required machining accuracy. For example, in lathe machine, the vertical deflection of the tool or its horizontal deflection in the direction of feed motion has little effect on the dimensional accuracy of the machined workpiece. However, diametrical error of 2 C takes place due to a deflection of the cutting edge by C in the radial direction. Hence the lathe slideways are designed for stiffness with a view to restrict their radial deflection. The appropriate direction for stiffness design in various machine tools can be selected in each particular case from the above mentioned principle. In a bed using two flat slideways as shown in Figure 13.5, the radial deflection takes place due to the contact deformation C_B of the vertical face of the slideway and rotation B of the saddle due to unequal contact deformation C_A and C_C of the horizontal flat faces.

Hence radial deflection is given by

$$\frac{C_A - C_B}{C_c} . h$$

Hence total radial deflection is given by

$$C_{FF} = C_B + \frac{C_A - C_C}{b} . h \qquad \ldots (24)$$

In the lathe bed using a combination of flat and V slideways, the contact deformation of the flat slideways is shown by lowering C_C. The faces of the V-slideways suffer a deformation of C_A and C_B. This will result in : B

(a) Vertical lowering of the V-slideway by

$$C_v = C_B \sin \square + C_A \cos \square$$

(b) Horizontal displacement of the apex of the V-slideway by

$C_h = C_B \cos \lambda - C_A \sin \lambda$

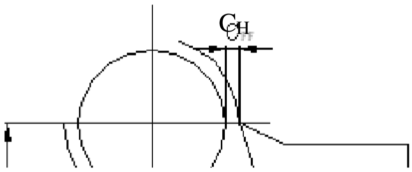


Figure 5 : Radial Displacement of Cutting Edge in Combination of Two Flat Slideways

This nature of deformation of flat and *V*-slideways results in :

(a) radial deflection of the cutting edge by C_H and

(b) rotation of the saddle due to unequal vertical lowering of the flat and *V* slideways which leads to radial deflection of the cutting edge by

$$\frac{C_A - C_B}{C_c} . h$$

The total radial deflection then becomes

$$C_{Fv} = C_h + \frac{C_v - C_C}{b} . h \qquad \ldots (25)$$

The contact deformation is assumed proportional to the average pressure for the purpose of stiffness design. The coefficient of proportionality d is known as contact compliance. This coefficient must be determined for each pair of slideway materials. However for approximate calculations, an average value of $d = 1 \times 10^{-6}$ mm^2/N may be used.

Eqs. (24) and (25) can be rewritten as follows :

$$C_{FF} = d_{pB} + \frac{P_A - P_C}{b} . h \qquad \ldots (26)$$

$$C_{Fv} = d\,(p_B \cos \lambda - p_A \sin \lambda) + \frac{dh}{b}(p_B \sin \lambda + p_A \cos \lambda - p_C) \qquad \ldots (27)$$

After calculating the total radial deflection of the cutting edge, the design for stiffness is carried out in accordance with $C_i = C_{ip}$.

SAQ 1

(a) What are principle parameters in designing slideways?

(b) Design the slideways for machine tool.

MODULE 3

Guideways operating under liquid friction conditions

Slideways operating under semi-liquid friction conditions are distinguished by a relatively high coefficient of friction μ. However, often, the major obstacle to the application of slideways in precision machine tools arises not from the high value of μ, but its variation. This variation occurs

1. with change of sliding speed, and
2. with passage of time.

The variation of μ as a function of sliding speed v is depicted in Fig. 4.19.[1] It is evident that in the range $v < v_0$ an increase of v is accompanied by a decrease of μ, while in the range $v > v_0$, an increase of v is accompanied by an increase of μ. The value of frictional resistance is thus not constant but depends upon the sliding velocity. Of particular interest is the fact that the initiation of a sliding movement is as a rule accompanied by a jerk due to the slip-stick effect. This phenomenon can be explained in the following manner. The coefficient of friction at zero sliding speed (static coefficient of friction) is greater than at a certain sliding speed v_0. At the start of a movement, the guided member must be strained to overcome frictional resistance corresponding to μ static. However, as soon as the guided member commences to slide, its frictional resistance drops due to a decrease in μ. The excessive force applied on the member when it was stationary is released giving rise to a 'jerk' in the movement of the guided member.

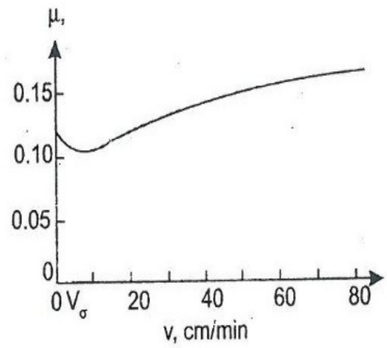


Fig. 4.19 *Variation of coefficient of friction with speed*

It was stated in Sec. 4.2 that slideways must be oiled from time to time to ensure their proper functioning. Under the weight of the guided member, the oil film between the sliding surfaces is gradually squeezed out, till it is replenished at the next oiling. It, therefore, transpires that the actual coefficient of friction between the sliding surfaces at a particular instant of time depends upon the time period elapsed since the last oiling. The variation of μ as a function of time is depicted in Fig. 4.20.[2] It may be concluded from Fig. 4.20 that the frictional resistance of the guided member does not remain constant.

It may be thus seen that due to variation of μ in slideways working under semi-liquid friction conditions, it is rather difficult, if not altogether impossible to employ them on machine tools which require a high accuracy of setting and working (feed or primary cutting) motions. In such machine tools, either hydrodynamic or hydrostatic slideways must be employed.

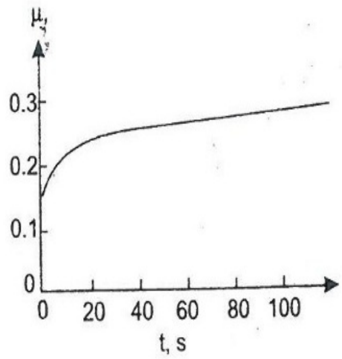


Fig. 4.20 *Variation of coefficient of friction with time*

MODULE 3

Design of Hydrodynamic Slideways

In hydrodynamic slideways, liquid friction conditions between sliding surfaces are achieved due to hydrodynamic action of the lubricant film. A sufficiently large hydrodynamic force which is capable of lifting the guided member is possible only at high sliding speeds. Hydrodynamic slideways are, therefore, used mainly where the sliding motion represents the primary cutting motion, e.g., vertical boring and turning mills and planing machines.

Hydrodynamic action is possible between sliding bodies only if they are inclined to each other, i.e., they form a wedge.

In a simple wedge (Fig. 4.21) of infinite width, the pressure distribution follows a parabolic law and the hydrodynamic force over a unit width is given by the expression,

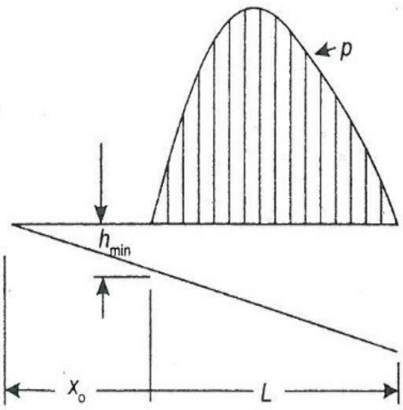


Fig. 4.21 *Distribution of hydrodynamic pressure in a wedge of infinite width*

$$Q = \frac{6\mu v L^2}{h_{\min}^2} \cdot K, \text{ kgf} \qquad (4.40)$$

For an actual wedge of finite width, the following expression holds good:

$$Q = \frac{5\mu v}{h_{\min}^2} \cdot \frac{L^2 b_0}{1+(L/b_0)^2} \cdot K \text{ kgf} \qquad (4.41)$$

In Eqs (4.40) and (4.41):

μ = coefficient of dynamic viscosity of the lubricant, $\frac{\text{kgf} \cdot \text{s}}{m^2}$

v = sliding velocity, m/s

L = length of slideway, m

b_0 = width of slideway, m

$h_{\min}$ = minimum thickness of the lubricant film, m; in ordinary machine tools $h_{\min}$ = 0.01–0.02 mm, for heavy-duty machine tools $h_{\min}$ = 0.06–0.1 mm,

K = coefficient that depends upon the wedge angle and its length,

$$K = m^2\left(l_n \frac{m+1}{m} - \frac{2}{2m+1}\right)$$

wherein $m = x_0/L$ (see Fig. 4.21).

The variation of K as a function of m is depicted in Fig. 4.22, wherefrom it can be seen that the maximum value of $K_{\max} = 0.0267$ occurs at $m = 0.7$.

If the slideway is provided with a single long wedge along its whole length, it will generally be found that $b_0 << L$. In such a case,

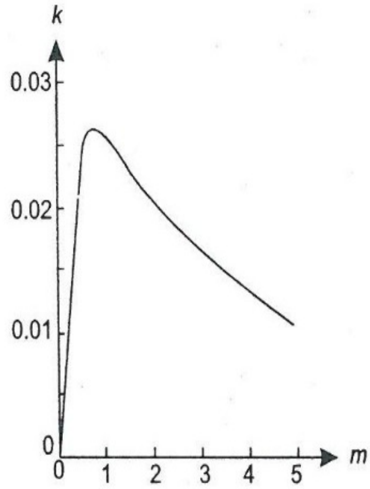


Fig. 4.22 *Design curve for computing K as a function of m*

$$\frac{L^2 b_0}{1+(L/b_0)^2} = \frac{L^2 b_0^3}{b_0^2 + L^2} = \frac{b_0^3}{1+(b_0/L)^2} \simeq b_0^3$$

and Eq. (4.41) can be written in the following modified form:

$$Q = \frac{5\mu v b_0^3}{h_{\min}^2} K, \text{ kgf} \tag{4.42}$$

Substituting $K_{\max} = 0.0267$, the expression for maximum load capacity can be written as

$$Q = \frac{0.133\mu v b_0^3}{h_{\min}^2} \text{kgf} \tag{4.43}$$

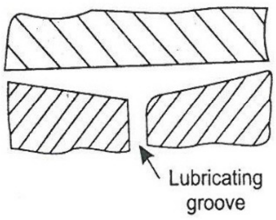


Fig. 4.23 *Elementary wedge*

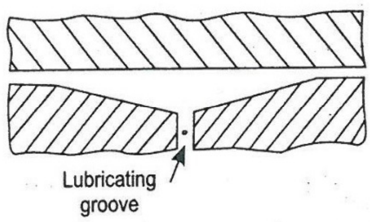


Fig. 4.24 *Elementary wedge alternating with a flat segment*

Often, instead of a single long wedge, machine tool slideways are provided with gently sloping elementary wedges on one of the mating surfaces as shown in Fig. 4.23. Each pair of oppositely inclined elementary wedges is located adjacent to a lubricating groove. If the total number of elementary wedges in the whole length of slideway is n, the total hydrodynamic force acting on the slideways can be found as

$$\Sigma Q = nQ$$

where Q is determined from Eq. (4.41).

A major drawback of hydrodynamic slideways is that liquid friction conditions are violated at starting and braking periods. This results in excessive wear of wedge corners. To overcome this difficulty, tapered wedges are alternated with small parallel segments (Fig. 4.24). An elementary length containing a flat segment and single wedge is known as an elementary composite slider bearing (Fig. 4.25) and its load capacity is given by the expression,

$$Q = \frac{5\mu v L^2 b_0}{6h_1^2(1 + (a^2L^2/b_0^2))} K_c \text{ kgf} \tag{4.43$'$}$$

The value of coefficient K_c has been plotted as a function of ratio h_2/h_1 for different values of a and shown in Fig. 4.26.

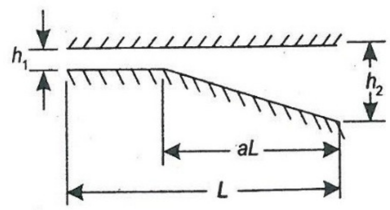


Fig. 4.25 *Elementary composite slider bearing*

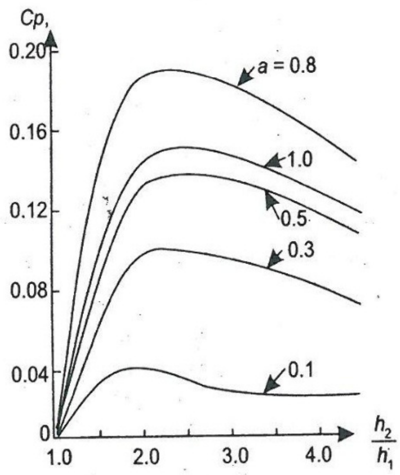


Fig. 4.26 *Design curve for computing K_c as a function of h_2/h_1*

MODULE 3

If n be the number of elementary composite sliders in the slideway length, the total hydrodynamic force can be found as

$$\Sigma Q = nQ = \frac{5\mu v L^2 b_0 \cdot n}{6h_1^2(1+(a^2L^2/b_0^2))}K_c$$

and substituting $K_c = 0.19$ at $a = 0.8$ and $h_2/h_1 = 2.2$, the load capacity of the slideway can be determined.

Equations (4.41) to (4.43) can be utilised for determining the load capacity, minimum required sliding speed and geometrical parameters, depending upon the particular design problem. They constitute the basic design equations for hydrodynamic slideways.

The experience of working with hydrodynamic slideways reveals that notwithstanding liquid-friction conditions, the mating surfaces get damaged due to seizure resulting from thermal and elastic deformations and also from accidental failures of the lubricating system. Hydrodynamic slideways are, therefore, made from friction pairs that have high resistance to seizure, such as non-ferrous alloy-cast iron, plastic-cast iron, etc.

Design of Hydrostatic Slideways

In hydrostatic slideways liquid-friction conditions at the interface of mating surfaces are achieved by supplying a lubricant under pressure, which is large enough to raise the sliding body and precludes metal-to-metal contact. Hydrostatic slideways are distinguished by

1. high load capacity at all sliding speeds, including zero speed,
2. no starting friction, extremely low running friction and consequently almost negligible wear,
3. high stiffness,
4. good damping, and
5. high uniformity and accuracy of feed and setting motions.

One of the drawbacks of hydrostatic slideways is the difficulty in fixing the moving member in a desired position. However, the single major factor which goes against these slideways is their high cost on account of an elaborate lubricating system. This restricts their application to sophisticated and expensive machine tools, such as grinding machines, heavy-duty horizontal boring machines, numerically controlled and copying machines, etc.

The principle of operation of hydrostatic slideways can be explained with the help of a pad bearing (Fig. 4.27). The oil is fed under pressure from the pump to the pocket. The required pressure depends upon load P acting on the pad. The greater the load P, the higher must be the pocket pressure p_0 to be able to lift the pad and thus provide an oil film in the gap. The oil is discharged through the sides of the pad and in the process its pressure falls from p_0 to atmospheric pressure.

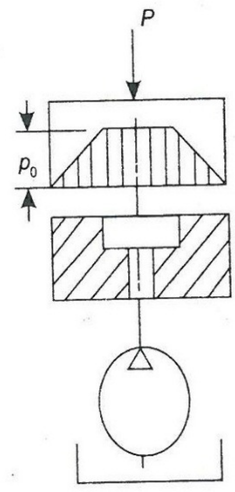


Fig. 4.27 *Schematic diagram of a hydrostatic pad bearing*

Consider now the case when a single pump is required to supply oil to more than one pocket simultaneously (Fig. 4.28). Suppose that loads P_1 and P_2 acting on pads I and II respectively are not equal, and that $P_1 < P_2$. Obviously, the required pocket pressure p_{01} will be less than p_{02}. As the oil is supplied in the gaps of pads I and II, the pressure inside the pockets begins to rise. When the pressure attains a value equal to p_{01}, pad I gets lifted. It now becomes impossible to build up the pressure to a value p_{02} required for raising pad II because all the excessive oil supplied by the pump is simply discharged through pocket I. It is, therefore, obvious that in this type of arrangement, a separate pump is required for each pad. Slideways normally have a large number of pockets, and therefore, providing an individual pump for each pocket makes them highly uneconomical.

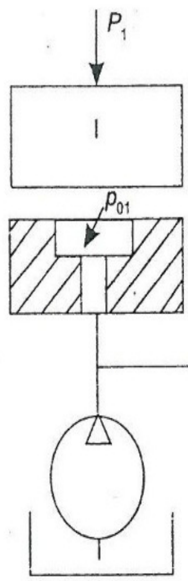


Fig. 4.28 *Schematic diagram depicting supply of lubricant to two pad bearings from a single pump*

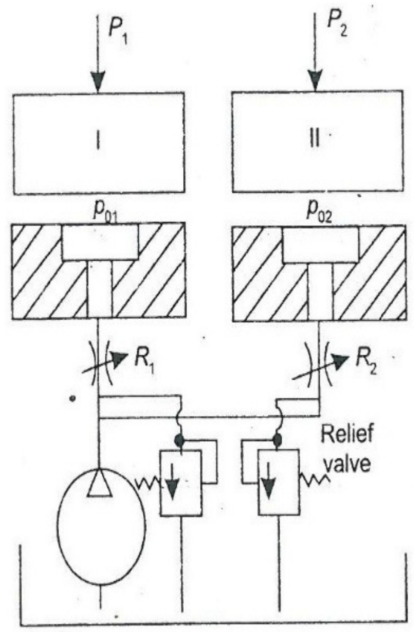


Fig. 4.29 *Schematic diagram depicting supply of lubricant to two pad bearings through restrictors*

The supply of oil to various pockets at different pressures from a single pump is possible if restrictors are introduced in the supply circuit (Fig. 4.29). The restrictor serves the function of restricting the flow of oil passing through it. Depending upon the pressure difference across its ends, a restrictor permits the flow of only a certain fixed volume of oil. If restrictor R_1 is set to provide a pocket pressure p_{01} in pad I, the excess pump supply will not be discharged through pad I, but will be diverted to the supply line of pad II.

Restrictor R_2 is preset to permit the passage of a larger volume of oil so that the pocket pressure in pad II can build up to the required level of p_{02}. The pressure in successive pads can thus be built up to the desired level as the discharge pressure is dictated by the relief-valve setting only.

Hydrostatic Slideway without Restrictor The load capacity of a single pad bearing which is directly connected to the pump (Fig. 4.30). is given by the expression,

$$P = p_0 A \cdot C_L = p_0 A_{eff} \text{ kgf} \tag{4.44}$$

where $A = BL$ = pad area m^2

C_L = load coefficient

$A_{eff} = C_L \cdot A$ = effective pad area

p_0 = pocket pressure, kgf/m^2

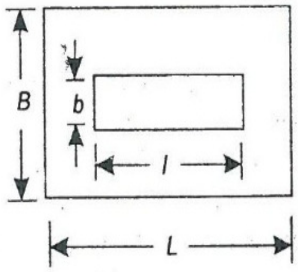


Fig. 4.30 *Rectangular pad bearing with a rectangular pocket*

MODULE 3

The value of coefficient C_L depends upon the shape of the pad and pocket. For a symmetrical rectangular profile of the pad and pocket,

$$C_L = \frac{1}{6LB}(2LB + lB + 2lb + Lb)$$

Generally, $C_L = 0.33–0.5$

The flow of oil through a directly connected pad bearing is,

$$Q = \frac{p_0}{R_0} \tag{4.45}$$

where R_0 represents the pocket resistance. It may be expressed as

$$R_0 = \frac{\mu}{h^3} C_F \tag{4.46}$$

Here h represents bearing clearance or gap in m and μ the absolute viscosity in kgf · s/m^2.

The quantity $1/C_F$ is known as flow coefficient and it depends upon the shape of the pad and pocket. For a rectangular pad and pocket,

$$C_F = 0.5 \times 10^{-9} \frac{(B-b)(L-l)}{l(L-l)+b(B-b)} \tag{4.47}$$

Keeping in mind Eqs (4.45) and (4.46), the load capacity of the pad bearing can be written as

$$P = \frac{Q \cdot \mu \cdot A \cdot C_L \cdot C_F}{h^3} = \frac{Q \cdot \mu \cdot A}{C_p h^3} \tag{4.48}$$

Coefficient $C_p = 1/C_F C_L$ is known as the power coefficient.

The stiffness K of a pad bearing is defined as the ratio of force increment to the change in bearing clearance, i.e.,

$$K = \frac{dP}{dh} \tag{4.49}$$

Differentiating Eq. (4.48) with respect to h, the stiffness of a pad bearing for constant Q, μ and A is obtained as

$$\frac{dP}{dh} = -\frac{3Q\,\mu A}{C_p h^4} = -\frac{3P}{h} \tag{4.50}$$

From the general expression of stiffness (Eq. (4.49)), it may be concluded that the pad bearing has infinite stiffness when $d_h = 0$, i.e., when gap h = constant for all load conditions. However, from Eq. (4.50), it ensues that the bearing stiffness actually depends upon the load and the film thickness in the bearing gap. For higher stiffness, it is necessary to achieve as small a film thickness as possible. However, the film thickness cannot be reduced below a certain level as otherwise metal-to-metal contact occurs, thus violating one of the fundamental functional requirement of hydrostatic slideways. In general, normal operating film thickness lies in the range 0.0025 to 0.025 mm. The higher value is selected for a large pad bearing working at high sliding speeds.

A comparison of load capacity Eqs (4.40) to (4.43) of hydrodynamic slideways with Eq. (4.48) of hydrostatic slideway reveals that in the former the film thickness varies as the square root of the load, whereas in the latter as a cube root. The hydrostatic slideway is, therefore, far more stiff than a hydrodynamic slideway.

Hydrostatic Slideway with a Restrictor The function of a restrictor in hydrostatic slideways has already been explained. The restrictor affects the flow, power and stiffness of the pad bearing. In most of the practical cases, the necessary slideway stiffness can be achieved by a simple capillary restrictor. This restrictor is a long tube of relatively small diameter ($l_c \geq 20\, d_c$). From considerations of equal fluid flow through the restrictor and pocket, the following relationship can be written:

$$\frac{p_0}{R_0} = \frac{p_p - p_0}{R_r} \tag{4.51}$$

where p_p = pump pressure
R_r = hydraulic resistance of the restrictor

The resistance of a capillary restrictor is given as

$$R_{rc} = \frac{128 \cdot \mu \cdot l_c}{\pi d_c^4} = \mu c_{rc}$$

where l_c and d_c are the length and diameter of the capillary respectively (both in m).

Substituting R_0 from Eq. (4.46) and R_{rc} from the above expression in Eq. (4.51) and equating flow through the pocket with the flow through the restrictor, we get

$$Q_c = \frac{p_0 h^3}{\mu C_F} = \frac{p_p - p_0}{\mu C_{rc}} \tag{4.52}$$

where from gap

$$h_c = \left[\frac{C_F}{C_{rc}} \frac{p_p - p_0}{p_0}\right]^{1/3} \tag{4.53}$$

Further, from Eq. (4.52) it is found that

$$p_0\left(\frac{h^3}{C_F} + \frac{1}{C_{rc}}\right) = \frac{p_p}{C_{rc}}$$

wherefrom,

$$p_0 = \frac{p_p C_F}{C_F + C_{rc} h^3}$$

Substituting this value of p_0 in Eq. (4.44), we get the load capacity of capillary compensated pad bearing as

$$P_c = \frac{p_p A C_F C_L}{C_F + C_{rc} h^3} \tag{4.54}$$

The stiffness of the pad bearing is found as

$$K_c = \frac{dP_c}{dh} = -p_p A C_F C_L \left[\frac{3C_{rc}h^2}{(C_F + C_{rc}h^3)^2}\right] = -3P_c C_{rc}\left(\frac{h^2}{C_F + C_{rc}h^3}\right) \tag{4.55}$$

Simplifying further

$$K_c = \frac{3P_c}{h}\left(\frac{h^3}{(C_F/C_{rc}) + h^3}\right) = -\frac{3P_c}{h}\left(\frac{1}{(C_F/C_{rc})h^3 + 1}\right)$$

From (4.52), we find that

$$\frac{C_F}{C_{rc}h^3} = \frac{p_0}{p_p - p_0}$$

Hence,

$$K_c = -\frac{3P_c}{h}\left(\frac{1}{(p_0/p_p - p_0) + 1}\right) = -\frac{3P_c}{h}\left(\frac{p_p - p_0}{p_p}\right) = -\frac{3P_c}{h}\left(1 - \frac{p_0}{p_p}\right)$$

The condition for maximum stiffness is $\frac{d^2P_c}{dh^2} = 0$; consequently

$$\frac{d^2P_c}{dh^2} = -3p_p A C_F C_L C_{rc}\left[\frac{(C_F + C_{rc}h^3)^2 2h - h^2 \cdot 2(C_F + C_{rc}h^3)(3C_{rc}h^2)}{(C_F + C_{rc}h^3)^4}\right] = 0$$

Therefore,

$$(C_F + C_{rc}h^3)h - 3C_{rc}h^4 = 0$$

whereform,

$$h^3 = \frac{C_F}{2C_{rc}}$$

Again, from Eq. (4.52)

$$\frac{C_F}{2C_{rc}} = \frac{h^3}{2}\left(\frac{p_0}{p_p - p_0}\right)$$

$$h^3 = \frac{h^3}{2}\left(\frac{p_0}{p_p - p_0}\right)$$

or $2(p_p - p_0) = p_0$, wherefrom $\frac{p_0}{p_p} = 2/3$

Thus, we find that maximum stiffness occurs at $\frac{p_0}{p_p} = 2/3$. Substituting this value in the expression for stiffness, we find

$$K_{c\max} = -3\frac{p_c}{h}\left(1 - \frac{p_0}{p_p}\right) = -3\frac{p_c}{h}\left(1 - \frac{2}{3}\right) = -\frac{p_c}{h}$$

MODULE 3

Knowing the expression for flow through an orifice restrictor, expressions similar to Eqs (4.52), (4.53), (4.54) and (4.55) can be derived for flow, gap, load capacity and stiffness of orifice compensated pad bearing. Flow through an orifice restrictor is given by the following expression:

$$Q_0 = C_0 A_0 \left[\frac{2(p_p - p_0)}{\rho}\right]^{1/2} = \frac{1}{C_{r0}}\left[\frac{p_p - p_0}{\rho}\right]^{1/2}$$

where C_0 = discharge coefficient of orifice

A_0 = area of the orifice

ρ = density of the fluid

Equating flow through the pocket and the orifice restrictor, we get

$$Q_0 = \frac{p_0 h^3}{\mu C_F} = \frac{1}{C_{r0}}\left[\frac{p_p - p_0}{\rho}\right]^{1/2} \tag{4.52$'$}$$

wherefrom,

$$h_0 = \left\{\frac{\mu C_F}{C_{r0}}\left[\frac{1}{\rho p_0}\left(\frac{p_p}{p_0} - 1\right)\right]^{1/2}\right\}^{1/3} \tag{4.53$'$}$$

Further, from Eq. (4.52′), we find

$$\frac{p_0^2 h^6}{\mu^2 C_F^2} = \frac{p_p - p_0}{\rho C_{r0}^2}$$

or

$$h^6 \rho C_{r0}^2 p_0^2 + \mu^2 C_F^2 p_0 - \mu^2 C_F^2 p_p = 0$$

Hence,

$$p_0 = -\frac{\mu^2 C_F^2 \pm \sqrt{\mu^2 C_F^4 + 4h^6 \rho C_{r0}^2 \mu^2 C_F^2 p_p}}{2h^6 \rho C_{r0}^2}$$

Neglecting the negative term under square root as infeasible, it is found that

$$p_0 = -\frac{\mu^2 C_F^2 + \mu C_F \sqrt{\mu^2 C_F^2 + 4h^6 \rho C_{r0}^2 p_p}}{2\rho C_{r0}^2 h^6}$$

Substituting this value of p_0 in Eq. (4.44), we get the following expression for load capacity of orifice compensated pad bearing:

$$P_0 = \left[\frac{-\mu^2 C_F^2 + \mu C_F \sqrt{\mu^2 C_F^2 + 4h^6 \rho C_{r0}^2 p_p}}{2\rho C_{r0}^2 h^6}\right] A \cdot C_L \tag{4.54$'$}$$

Let $$-\mu^2 C_F^2 + \mu C_F \sqrt{\mu^2 C_F^2 + 4h^6 \rho C_{r0}^2 p_p} = \text{Function (1)}$$

and $$1/2\ \rho C_{r0}^2 h^6 = \text{Function (2)}$$

The stiffness of the orifice compensated pad bearing is found as

$$\frac{dP_0}{dh} = \left[\frac{d}{dh}(\text{Function (1)}) \times \text{Function(2)} + \frac{d}{dh}(\text{Function (2)}) \times \text{Function (1)}\right] AC_L$$

$$\frac{d}{dh}(\text{Function (1)}) = \frac{\mu C_F \times 12 C_{r0}^2 \rho h^5 p_p}{\sqrt{\mu^2 C_F^2 + 4h^6 \rho C_{r0}^2 p_p}}$$

$$\frac{d}{dh}(\text{Function (2)}) = -3/\rho C_{r0}^2 h^7$$

Hence,

$$\frac{dP_0}{dh} = \left[\frac{12\mu C_F C_{r0}^2 \rho h^5 p_p}{\sqrt{\mu^2 C_F^2 + 4C_{r0}^2 \rho h^6 p_p}} \times \frac{1}{2\rho C_{r0}^2 h^6} + \left(\frac{-3}{\rho C_{r0}^2 h^7}\right) \times -\mu^2 C_F^2 + \mu C_F \sqrt{\mu^2 C_F^2 + 4C_F^2 \rho h^6 p_p}\right] AC_L$$

$$= \left[\frac{6\mu C_F p_p}{h\sqrt{\mu^2 C_F^2 + 4C_{r0}^2 \rho h^6 p_p}} - \frac{3}{\rho C_{r0}^2} \times \frac{-\mu^2 C_F^2 + \mu C_F \sqrt{\mu^2 C_F^2 + 4C_{r0}^2 \rho h^6 p_p}}{h^7}\right] AC_L$$

$$= \left[\frac{6\rho C_{r0}^2 \mu C_F p_p h^6 - 3\mu C_F\left(-\mu C_F + \sqrt{\mu^2 C_F^2 + 4C_{r0}^2 \rho h^6 p_p}\right)\left(\sqrt{\mu^2 C_F^2 + 4C_{r0}^2 \rho h^6 p_p}\right)}{h^7 \rho C_{r0}^2 \sqrt{\mu^2 C_F^2 + 4C_{r0}^2 \rho h^6 p_p}}\right] AC_L$$

$$= \left[\frac{6\rho C_{r0}^2 \mu C_F p_p h^6 - 3\mu C_F\left\{-\mu C_F \sqrt{\mu^2 C_F^2 + 4C_{r0}^2 \rho h^6 p_p} + (\mu^2 C_F^2 + 4C_{r0}^2 \rho h^6 p_p)\right\}}{h^7 \rho C_{r0}^2 \sqrt{\mu^2 C_F^2 + 4C_{r0}^2 \rho h^6 p_p}}\right] AC_L$$

$$= \left[\frac{6\rho C_{r0}^2 \mu C_F p_p h^6 + 3\mu^2 C_F^2 (\mu^2 C_F^2 + 4C_{r0}^2 \rho h^6 p_p)^{1/2} - 3\mu^3 C_F^3 - 12 C_{r0}^2 \rho h^6 p_p}{h^7 \rho C_{r0}^2 \sqrt{\mu^2 C_F^2 + 4C_{r0}^2 \rho h^6 p_p}}\right] AC_L$$

$$= \left[\frac{-6\rho C_{r0}^2 \mu C_F p_p h^6 - 3\mu^3 C_F^3 + 3\mu^2 C_F^2 (\mu^2 C_F^2 + 4C_{r0}^2 \rho h^6 p_p)^{1/2}}{h^7 \rho C_{r0}^2 \sqrt{\mu^2 C_F^2 + 4C_{r0}^2 \rho h^6 p_p}}\right] AC_L \qquad (4.55')$$

It is clear from the foregoing analysis that gap h and flow Q both decrease with pocket pressure p_0 in capillary as well as orifice compensated pad bearings (*see* Eqs (4.52) and (4.53) for capillary and Eq. (4.52′) and (4.53′) for orifice compensated pad bearings). This restricts the application of such hydrostatic pad bearings because under heavy loads the gap reduces to such an extent that the liquid friction conditions between the pads are violated. The range of application of hydrostatic pad bearings may be expanded by using the capillary or orifice restrictor in conjunction with special spool valves. Two such arrangements are discussed below.

Figure 4.31 shows a constant-flow restrictor using a spool valve. The spool displacement is controlled by a weak spring. The lubricant coming out of the restrictor at pressure p_0 is fed not only to the pocket of the pad bearing but also to the back-pressure chamber of the spool valve. Any increase in p_0 pushes the spool rightwards and increases p_2 such that the pressure drop $(p_2 - p_0)$ across the restrictor and hence, the flow rate remain constant. For the pad bearing using a constant-flow restrictor,

$$Q = \frac{p_2 - p_0}{R_r} = \text{const} \tag{4.56}$$

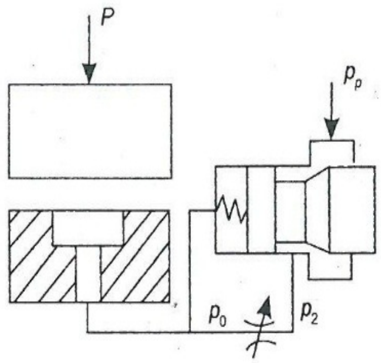


Fig. 4.31 *Schematic diagram of pad bearing using constant flow restrictor*

where R_r is the hydraulic resistance of the restrictor in kgf · s/m^5. Keeping in mind Eqs (4.45), (4.46) and (4.56), we get

$$Q = \frac{p_0 h^3}{\mu C_F} = \frac{p_2 - p_0}{R_r} = \text{const} \tag{4.57}$$

From Eq. (4.57), the expression for bearing clearance is found as

$$h = \left(\frac{Q \cdot \mu \cdot C_F}{p_0}\right)^{1/3} \text{m} \tag{4.58}$$

As Eq. (4.58) is exactly similar to Eq. (4.48) for hydrostatic slideway without restrictor, the stiffness of hydrostatic slideway with constant flow restrictor can be found from the expression given below (*see* Eq. (4.50)).

$$K_{Cf} = \frac{3P}{h} \tag{4.59}$$

i.e., the same as for the directly connected pad bearings.

A pad bearing using a constant-gap restrictor is shown in Fig. 4.32. If A_0 and A_2 be the effective area of the opposite faces of the spool, such that $A_0 = \gamma A_2$, the pressures acting on these faces would be related by the expression $p_2 = \gamma p_0$. This implies that an increase of pocket pressure by Δp_0 will increase the input pressure to the restrictor p_2 by an amount Δp_2 which is γ times greater than Δp_0. Since $\gamma > 1$,

$$(p_2 + \Delta p_2) - (p_0 + \Delta p_0) > p_2 - p_1$$

Therefore, the flow through the restrictor will increase such that the bearing clearance remains constant. In this case,

$$Q = \frac{p_0 h^3}{\mu \cdot C_F} = \frac{p_0(\gamma - 1)}{R_C} \tag{4.60}$$

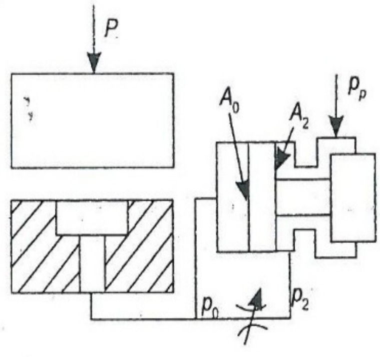


Fig. 4.32 *Schematic diagram of pad bearing using constant-gap restrictor*

wherefrom,

$$h = \left(\frac{(\gamma - 1) \cdot \mu \cdot C_F}{R_r} \right)^{1/3} \text{ m} = \text{const} \tag{4.61}$$

It is evident from Eq. (4.61) that the bearing clearance is independent of the pocket pressure, i.e., the load. Therefore, theoretically the hydrostatic slideway using a constant-gap restrictor has infinite stiffness.

It may be mentioned that in the constant flow restrictors, the flow is maintained constant only till pocket pressure reaches a value close to pump pressure p_p. As p_0 approaches p_p, the spool valve opens the flow path through the capillary or orifice restrictor and flow Q falls rapidly. Similarly, in case of the constant gap restrictor, the bearing gap is maintained constant only till pocket pressure p_0 reaches a value such that γp_0 approaches pump pressure p_p. Beyond this value, the pad bearing behaves as a simple capillary or orifice restrictor.

For the sake of performance comparison, the Q–p_0 and h–p_0 relationships for various restrictors are shown in Fig. 4.33a and b, respectively. When operating under identical conditions of flow, film thickness, load and pump pressure, the constant-gap restrictor displays infinite stiffness, while constant-flow, orifice and capillary restrictors provide decreasing stiffness in the given order. The stiffness of capillary and orifice restrictor controlled slideways can be raised to that of slideways using constant-flow restrictors simply by changing the supply pressure from the pump or the hydraulic resistance of the restrictor, both of which change the flow. However, it should be borne in mind that this would entail increased energy ($p_p \times Q$) supply to the pad bearings.

The optimum viscosity of oil employed in hydrostatic slideways is determined from the condition of minimum energy losses consisting of losses due to shearing of oil and its flow through the bearing clearance. These losses are assessed in terms of heat generation. The optimum absolute viscosity of oil can be determined from the following relationship:

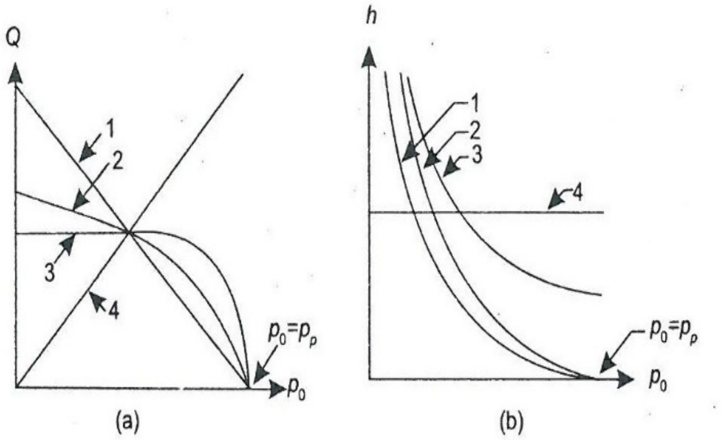
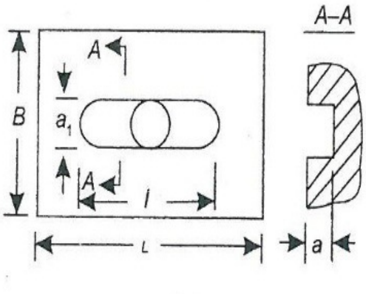


Fig. 4.33 *Effect of pocket pressure on (a) flow (b) bearing gap: 1. Capillary restrictor 2. Orifice restrictor 3. Constant-flow restrictor 4. Constant-gap restrictor*

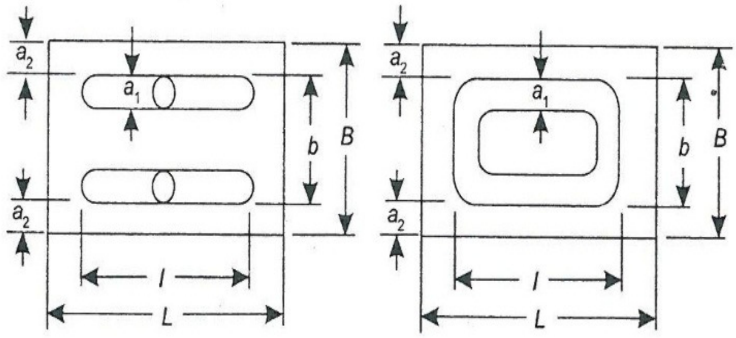
$$\mu_{\text{opt}} = \frac{h^2}{v} \sqrt{\frac{p_0 \cdot p_p \cdot C_L}{C_F \cdot A'}} \text{ kgf} \cdot \text{s/m}^2 \tag{4.62}$$

where A' = pad area – 3/4 (pocket area), m^2.

The size and shape of pockets can be assigned as per the following recommendations:



(a)



(b)

Fig. 4.34 *Pocket geometry for (a) narrow slideways (b) wide slideways*

1. For narrow slideways (Fig. 4.34a), $B < 50$ mm:

$$a = 0.5a_1$$
$$a_1 = 0.1B$$

2. For wide slideways (Fig. 4.34b), $B > 50$ mm:

$$a_2 = 2a_1$$
$$b = B - 2a_2$$

Closed Hydrostatic Slideways Hydrostatic lubrication is possible for all profiles and profile combinations in both open and closed-type slideways (Fig. 4.35). Open hydrostatic slideways can be employed only when

1. the tilting moment is not large,
2. the weight of the table and workpiece represents the main load,
3. the ratio of maximum load to minimum does not exceed 2, and
4. the minimum load is sufficient to provide the necessary stiffness of the oil film.

If any of the above conditions is violated, closed-type slideways, must be used. In these slideways, it is possible to increase the stiffness by raising pressure without the apprehension of overturning the guided member.

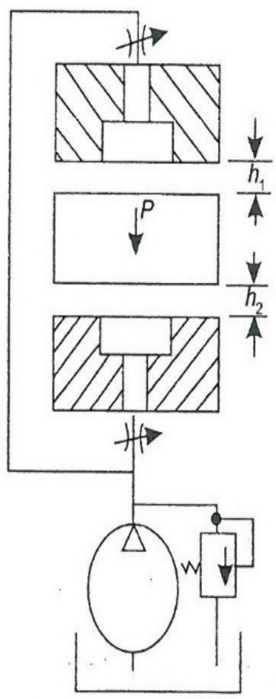


Fig. 4.35 *Schematic diagram of closed hydrodynamic slideways*

The load capacity of a closed-type slideway with a capillary restrictor is determined from the expression,

$$P = p_p A C_L C_p(\varepsilon, k) \tag{4.63}$$

and its stiffness by

$$K = -\frac{3 p_p A C_L}{h_0} C_k(\varepsilon, k) \tag{4.64}$$

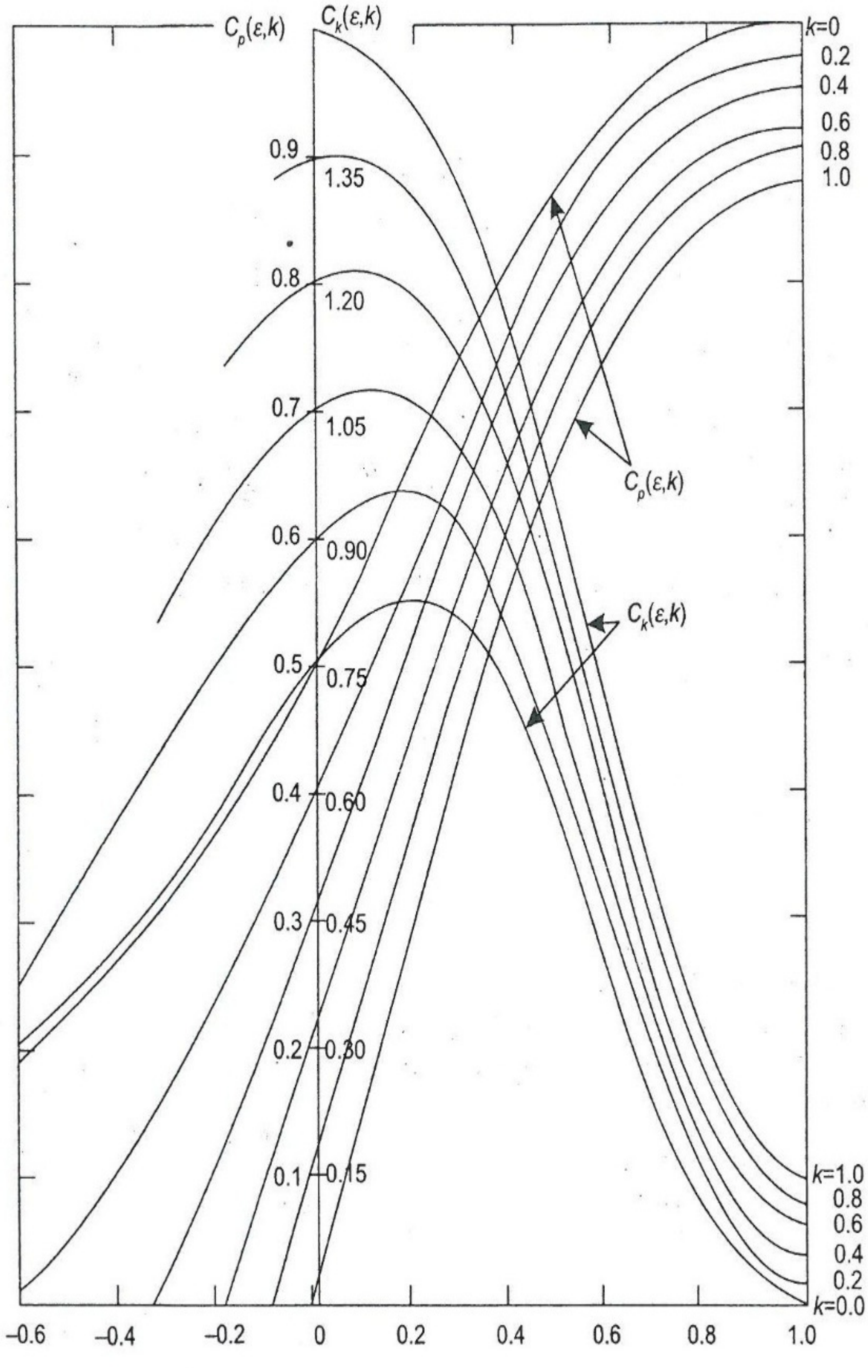


Fig. 4.36 *Design curves for computing $C_p(\varepsilon, K)$ and $C_k(\varepsilon, K)$ as a function of ε*

MODULE 3

Coefficients $C_p(\varepsilon, k)$ and $C_k(\varepsilon, k)$ in Eqs (4.63) and (4.64) depend upon

1. $\varepsilon = \dfrac{h_0 - h_1}{h_1}$

   where h_0 is the initial bearing clearance, and

2. $k = \dfrac{A_2 C_{L2}}{A_1 C_{L1}} \cdot \dfrac{C_{F2}}{C_{F1}} \cdot \dfrac{C_{rc2}}{C_{rc1}}$

The values of coefficients $C_p(\varepsilon, k)$ and $C_k(\varepsilon, k)$ can be determined from the curves of Fig. 4.36.[3]

Design of Aerostatic Slideways

The operating principle of aerostatic slideways is similar to that of hydrostatic slideways, the basic difference being that the lubricating medium in this case is compressed air instead of oil. The important positive features of aerostatic slideways are:

1. It is not necessary to collect oil.
2. The moving member can be reliably clamped in the desired position after cutting off the supply of compressed air.
3. Due to air pressure the mating surfaces are effectively protected from dust, abrasive particles and chips.
4. The pulling force required to displace even heavy units is very small (approximately 1 kgf) due to negligibly small friction at all sliding velocities.

The aerostatic slideways have lower stiffness (approximately of the order of 10 kgf/micron) as compared to hydrostatic slideways due to compressibility of air. They also have poorer damping properties because the viscosity of air is much lower than that of oil. Owing to these limitations, aerostatic slideways are not suitable for feed and primary cutting motions, though they are finding ready application in setting motion trains of precision machine tools, such as jig-boring and jig-drilling machines.

The compressed air can be supplied to the aerostatic slideway either from a central supply line or from an individual compressor. In slideways of average width, air is supplied at the interface of sliding surfaces by straight triangular microgrooves (Fig. 4.37). A hole at the middle of the groove is connected to the source of air supply through an orifice restrictor. In case of slideways of considerable width, closed rectangular grooves of the type shown in Fig. 4.35 are employed.

For stability, the volume of air in the microgroove should be four to five times less than in the clearance between the sliding surfaces. Keeping this in view, the depth of the triangular groove is determined from the expression

$$t \le \sqrt{0.7\, Bh} \text{ m} \qquad (4.65)$$

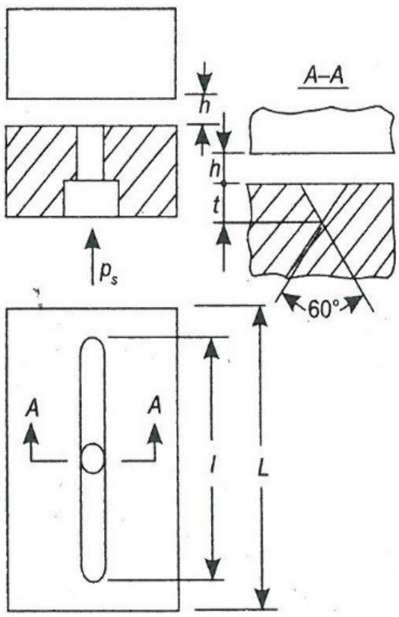


Fig. 4.37 *Schematic diagram of an aerostatic pad bearing*

where B = width of slideways, m

h = gap between sliding surfaces, m

The load capacity of the air pocket is determined on the assumption that air flow is continuous and it occurs only in the transverse direction. It may be found from the following relationship:

$$P = B \cdot l \cdot f_p(k) \tag{4.66}$$

where $f_p(k)$ is the load coefficient which depends upon the geometrical characteristic of the slideway

$$k = 17.3\frac{l^3h^3}{Bt^4}$$

The value of coefficient $f_p(k)$ can be taken from Fig. 4.38[4] for different values of k and pocket pressure p_0.

The stiffness of an open-type aerostatic slideway is given by the expression,

$$K = \frac{Bl}{h}f(k, p_s) \tag{4.67}$$

The stiffness coefficient $f(k, p_s)$ depends upon k and supply pressure p_s. Approximately,

$$f(k, p_s) = 0.5p_s \tag{4.68}$$

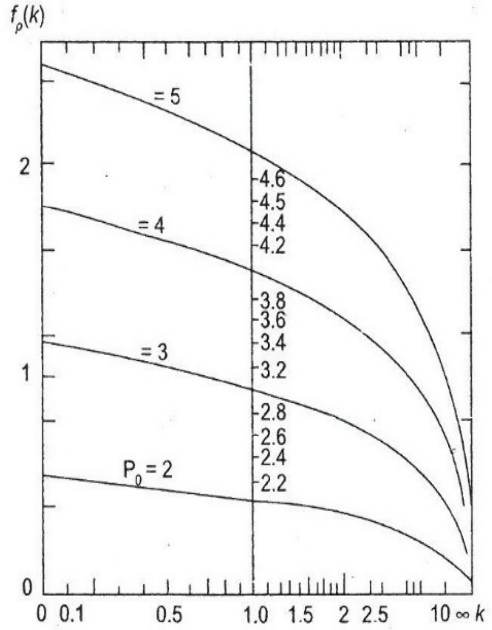


Fig. 4.38 *Design curves for computing $f_p(k)$ as a function of k for different values of p_0*

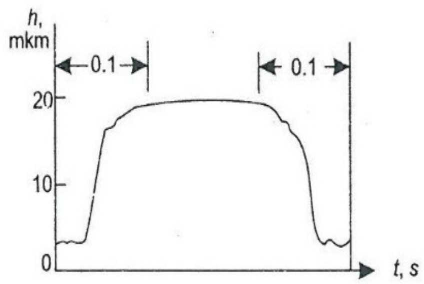


Fig. 4.39 *Transient process during raising and lowering of aerostatic slideways*

It is evident from Eqs (4.67) and (4.68) that the stiffness of aerostatic slideways can be improved by raising the supply pressure. The compressed air from a centralised supply line is supplied at a pressure of 2–5 kgf/cm^2. For a higher supply pressure, individual compressors must be used. When the air supply is switched on, the slideway clearance attains a stable uniform value only after the passage of a certain time period. Similarly, when the air supply is shut off, the moving member takes some time before it settles on the slideway surface. Though the transient period is only of the order of 0.1 s (Fig. 4.39), it should be taken into account at the time of starting a setting motion or clamping the moving member when it has reached a desired position.

MODULE 3

Design of Anti-Friction Guideways

In anti-friction ways, intermediate rolling members (balls or rollers) are inserted between the sliding surfaces, thus changing the nature of friction from sliding to rolling. They have the following positive features:

1. Low friction as compared to slideways.
2. Uniformity of motion even at slow speeds due to virtual absence of the stick-slip phenomenon.
3. High stiffness if the rolling members are preloaded.
4. Possibility of using high velocities of motion.

These properties explain the wide application of anti-friction guideways in both feed as well as primary cutting motion trains of high-accuracy machine tools.

The contact of rolling elements with guideway surfaces occurs over a point or a line. It is, therefore, necessary for the guideway material to have high contact strength to be able to transmit large loads through the small contact area. Also, the line or point contact follows the profile of the guideway surface and reproduces it on the machined surface. This necessitates that guideway surfaces be machined with a high degree of accuracy. Cast iron guideways are rarely used as they have poor contact strength; as compared to steel guideways they permit 10 times less load when rollers are used, and 30 times less load when balls are used as rolling elements. Materials that are commonly used for making anti-friction-guideways are:

1. Spindle steels hardened to RC = 60–62.
2. Low carbon structural steel 20Cr1Mn*60* Si27Ni25(AiSi5120) case-hardened to a depth of 0.8–1.0 mm.

Anti-friction guideways employ the same shapes as slideways. These shapes can be obtained by an appropriate surface profile or by changing the profile and location of the rolling elements. Anti-friction ways can be open or closed type. A few examples of open ways are shown in Fig. 4.40. It may be noted that the V profiles of Fig. 4.40c and d have been obtained by different methods. Open-type anti-friction ways are employed only when the dead weight of the moving member constitutes the major load that does not change appreciably during the cutting operation.

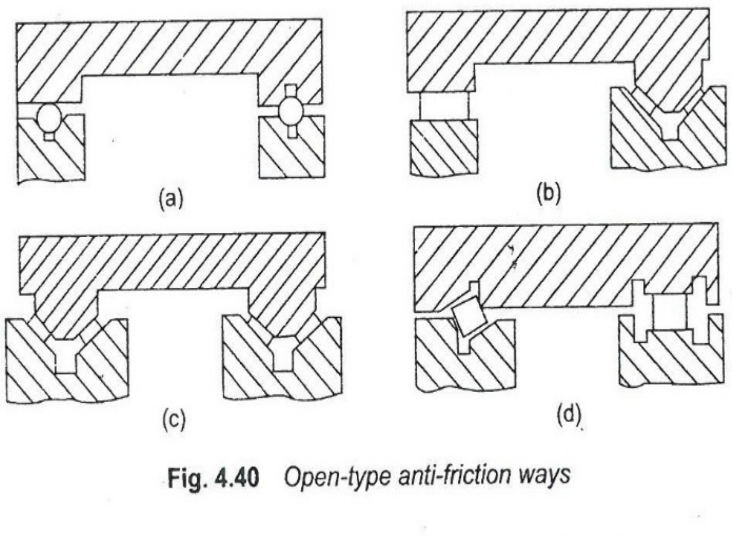


Fig. 4.40 *Open-type anti-friction ways*

Closed-type anti-friction guideways (Fig. 4.41) are used when working loads are relatively large and guideways are required to have high stiffness. Higher stiffness is achieved through preloading of rolling members. As a matter of fact, horizontal rolling members automatically experience some preloading due to the weight of the moving member. However, such preloading cannot be varied to yield optimum stiffness. A better method of preloading both horizontal as well as vertical rolling members is by means of flat strips and taper gibs (*see* Sec. 4.2.3). It should be kept in mind that strips and gibs should be located on that of side of the guideway which is loaded less in order to achieve higher stiffness. The recommended preloading is

1. 2–3 microns for precision machines using spherical balls,
2. 7–10 microns for general machines tools using spherical balls, and
3. 15–25 microns for general machines tools using rollers.

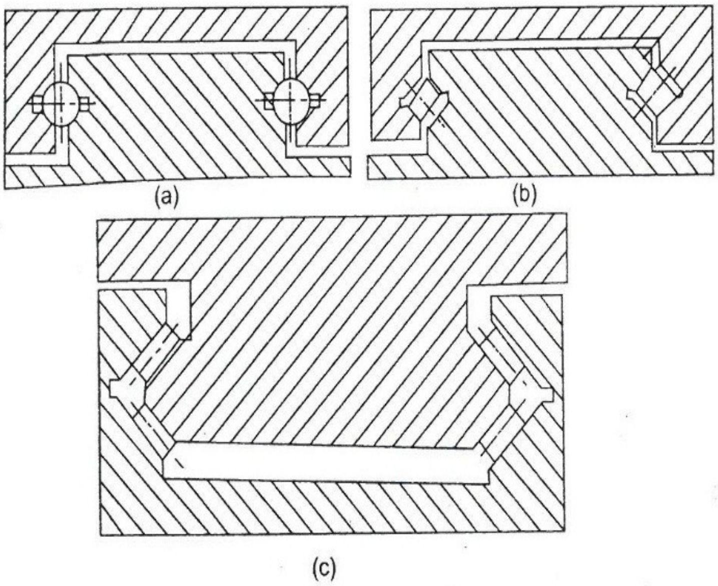


Fig. 4.41 *Closed-type anti-friction ways*

One of the major practical problems associated with the application of anti-friction guideways is that the rolling elements lag behind the moving member. For instance, the linear velocity of the centre of the rolling element is only half that of the moving member (Fig. 4.42). This necessitates that

1. either the travel of the moving member should be restricted, or
2. there should be provision for recirculating the balls so that there is always a sufficient number of rolling members between the stationary surface and the moving member.

A limited travel anti-friction guideway is shown in Fig. 4.43. The rolling elements are held in a cage of length l_c such that

$$l_c = l_g - \frac{l}{2} \tag{4.69}$$

where l_c = length of the cage

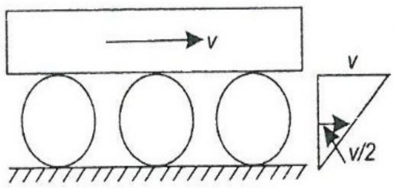


Fig. 4.42 *Velocity distribution along the height of a rolling element*

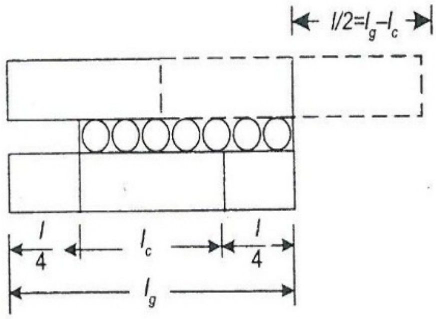


Fig. 4.43 *Schematic diagram of a limited travel anti-friction way*

MODULE 3

l_g = length of the stationary guideway

l = length of travel of the moving member

Equation (4.69) is valid for the condition that the roller cage is at the middle of the guideway when the moving member is in the centre position. The total travel of the moving member is made up of $l/2$ to the left of the centre and $l/2$ to the right.

The schematic diagram of an anti-friction guideway with recirculation of rolling elements to provide unlimited travel is shown at Fig. 4.44. Rolling elements are generally not enclosed in a cage. However, in some designs rolling elements alternate with idle elements of a slightly smaller diameter which thus serve as a separator.

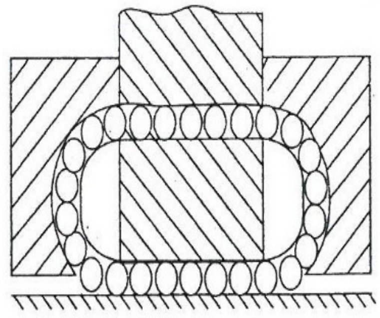


Fig. 4.44 *Schematic diagram of an unlimited travel way with recirculating rolling elements*

The important design parameters of anti-friction guideways are now discussed below.

The frictional force acting on the faces of an anti-friction guideway is given by the expression

$$F = nF_c + \frac{f_r}{r_{eq}} N \,\text{kgf} \tag{4.70}$$

where F_c = constant component of the frictional force which is approximately equal to 0.4–0.5 kgf

n = number of faces of the guideway

f_r = coefficient of rolling friction; for cast iron ways, $f = 0.0025$ cm and for general steel ways, $f = 0.001$ cm

$r_{eq} = r/k$ = equivalent radius of the rolling element, where r is the radius of rolling element in cm and k the reduction coefficient; for open ways, $k = 1.4$–1.5, while for closed ways, $k = 2.8$

$N = P_z + W_t + W_w$ = normal load on the way, kgf, where P_z is the vertical component of the cutting force, W_t the weight of the table and W_w the weight of the workpiece

As is evident from Eq. (4.70), the frictional force is virtually independent of the travel velocity. Having determined the frictional force, the required pulling force can be found from the expression,

$$Q = F + P_x \,\text{kgf} \tag{4.71}$$

where P_x = axial component of the cutting force, kgf.

The load capacity of the anti-friction ways is determined from the consideration of contact strength. For anti-friction roller ways it is given by the expression,

$$P = K \cdot b \cdot d, \text{kgf} \tag{4.72}$$

and for anti-friction ball ways by the expression,

$$P = Kd^2, \text{kgf} \tag{4.73}$$

In Eqs (4.72) and (4.73)

b = length of the roller, cm

d = diameter of the rolling member, cm

K = load coefficient, kgf/cm^2

Load coefficient K represents a conditional stress with limiting values as given in Table 4.3.

Table 4.3 *Limiting values of load coefficient K*

Type of rolling member	*K, kgf/cm^2*	
	Ground steel ways	*Cast iron ways*
Ball	6	0.2
Roller	150–200	13–20

Equations (4.72) and (4.73) may be employed for

1. calculating load P, if the dimensions of rolling members have been selected, or
2. for selecting proper dimensions of the rolling elements

For both the design problems, load P is calculated and checked to ensure that it is less than the load P_{max} on the maximum loaded rolling member

$$P = b \cdot t \cdot p_{max} \text{ kgf} \tag{4.74}$$

where b = length of the roller or diameter of the ball, cm

t = pitch of rolling members, i.e. distance between the centres of two adjacent rolling members, cm

P_{max} = maximum pressure acting on the guideway surface determined as per the procedure explained in Sec. 4.3.1, assuming the rolling members to be replaced by a continuous elastic surface, kgf/cm^2

The number of rolling elements in a guideway using rollers is determined from the expression,

$$16 \le Z \le \frac{q}{4} \tag{4.75}$$

and in a guideway using spherical balls from

$$16 \le Z \le \frac{P}{3\sqrt{d}} \tag{4.76}$$

In Eqs (4.75) and (4.76)

Z = number of rolling elements

q = load per unit length of the roller, kgf/cm

P = load per ball, kgf

d = diameter of the spherical ball, cm

It has been observed from practise that if $Z < 16$, the profile errors of the guideway are copied by the rolling elements and transferred to the machined surface. On the other hand, if Z violates the other inequality constraint, a large number of rolling elements remains partially under loaded and some are not loaded at all.

The design of anti-friction ways for stiffness involves calculation of the elastic displacement under the working loads. They can be determined for anti-friction roller ways from the equation

$$\delta = C_r \cdot q \tag{4.77}$$

and for anti-friction ball ways from the equation

$$\delta = C_b \cdot P \tag{4.78}$$

Compliance coefficients C_r and C_b depend upon the load, diameter of the rolling members, material of the guideway and its surface finish. Their values may be obtained from the curves given in Fig. 4.45.

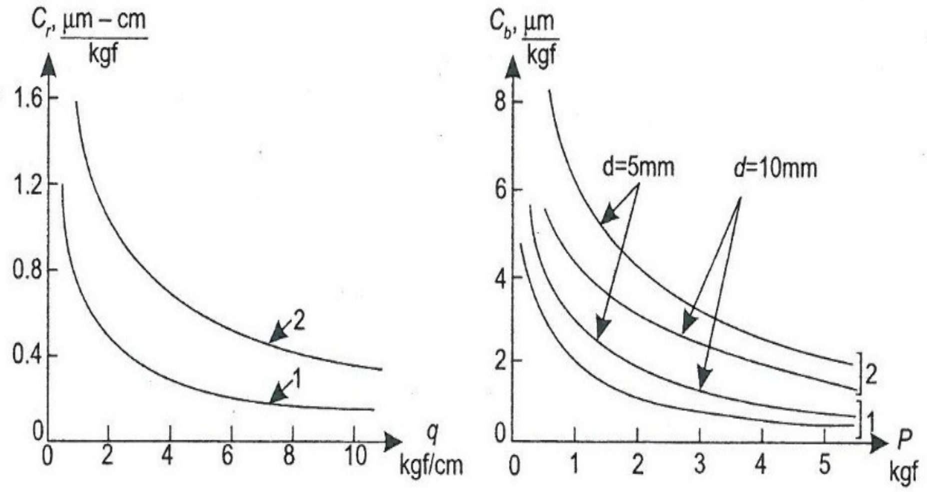


Fig. 4.45 *Design curves for computing the compliance coefficient for (a) roller anti-friction ways (b) ball anti-friction ways. 1. Ground steel ways 2. Scraped cast iron ways*

The linear relationships of Eqs (4.77) and (4.78) between load and displacement are valid only under the assumption that guideways and rolling members are ideally manufactured and assembled. In fact, manufacturing and assembly errors, some of which are depicted in Fig. 4.46[5], can often be almost as large as the elastic displacement. An increase in the magnitude of the manufacturing error results in a reduction of the guideway stiffness. The effect of the manufacturing error on the guideway stiffness can be accounted through curves depicted in Fig. 4.47.

MODULE 3

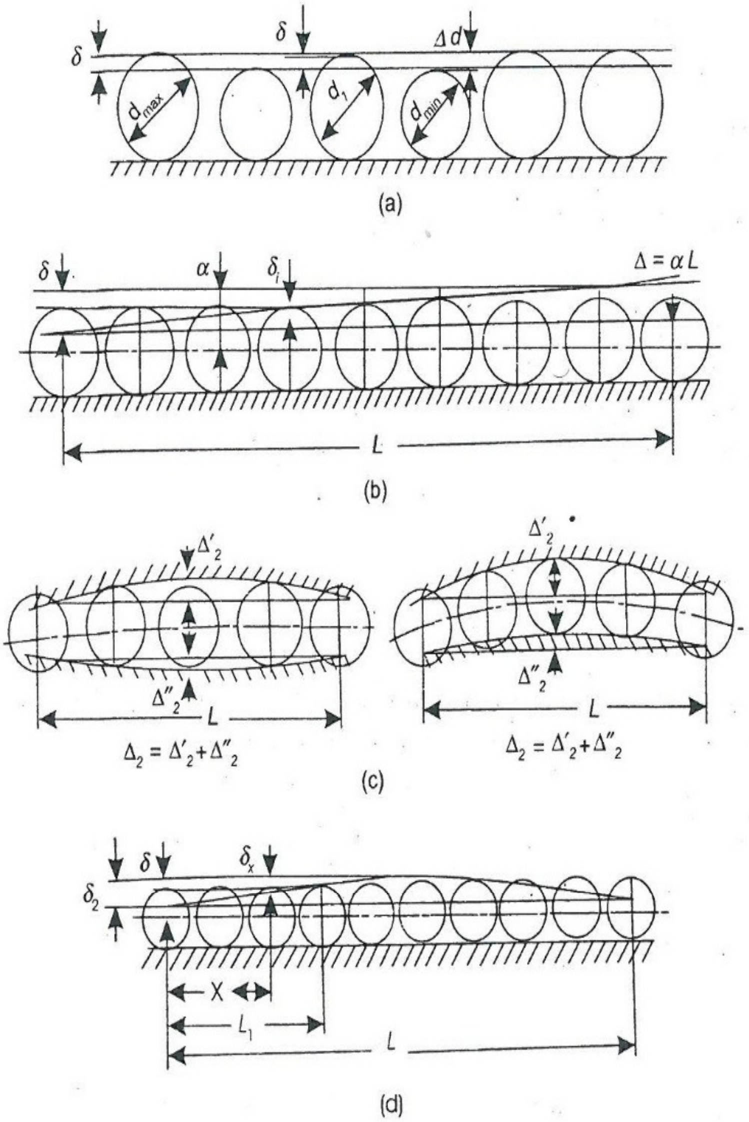


Fig. 4.46 *Errors in anti-friction ways due to (a) difference in ball size (b) lack of parallelness (c) and (d) lack of straightness*

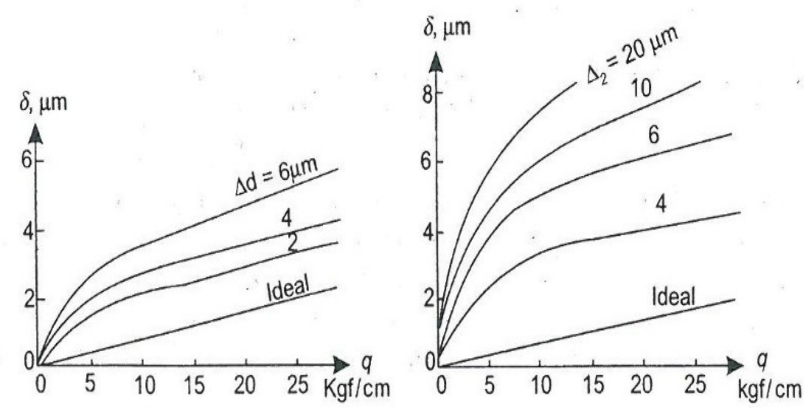


Fig. 4.47 *Effect of manufacturing errors on stiffness of roller anti-friction ways: (a) Errors due to difference in size of elements (b) Errors due to lack of straightness*

MODULE 3

Combination Guideways

While discussing the design of slideways operating under conditions of semi-liquid friction (Sec. 4.3), it was mentioned that they are distinguished by a relatively high coefficient of friction ($\mu = 0.15–0.6$). This creates difficulties in their application for heavy-duty machine tools. The performance and life of conventional slide-ways can be significantly improved and the scope of their application considerably expanded by combining them with other guideways which provide relief from the normal load. The two commonly used types of combination guides are discussed below.

Slideways with Hydraulic Relief

This slideway can be considered as intermediate between conventional slideways and hydrostatic slideways. The normal load on contacting surfaces is partially relieved by supplying oil to the interface at a suitable pressure. This pressure, however, is not large enough to lift the moving member as in hydrostatic slideways. This type of combination guideway incorporates the positive features of

1. the hydrostatic guideway characterised by a low coefficient of friction, and
2. the conventional slideway characterised by high contact strength and reliable clamping of the moving member in any desired position.

The oil is supplied into the clearance between the sliding surfaces through appropriately interspaced pockets. The shape and geometry of the pockets, and the arrangement utilised for pumping oil is the same as explained earlier for hydrostatic slideways (see Sec. 4.4.2).

The frictional force in a slideway with hydraulic relief is given by the relationship,

$$F = f \cdot N\left(1 - \frac{H}{N}\right) \tag{4.79}$$

where f = coefficient of sliding friction
N = normal load on the guideways
H = hydraulic relief

The hydraulic relief can be calculated from Eq. (4.44). Generally, pocket pressure is selected such that $H = 0.7N$. If the load on the slideway is not uniformly distributed along its length, the oil supply is regulated by restrictors. However, the restrictors can be dispensed with when the load is uniform and more or less constant, as is the case when the moving member is heavy and the cutting force is relatively small. Slideways with hydraulic relief are particularly effective and feasible in the latter case which explains their increasing application in tables of plano-milling, copy milling and grinding machines.

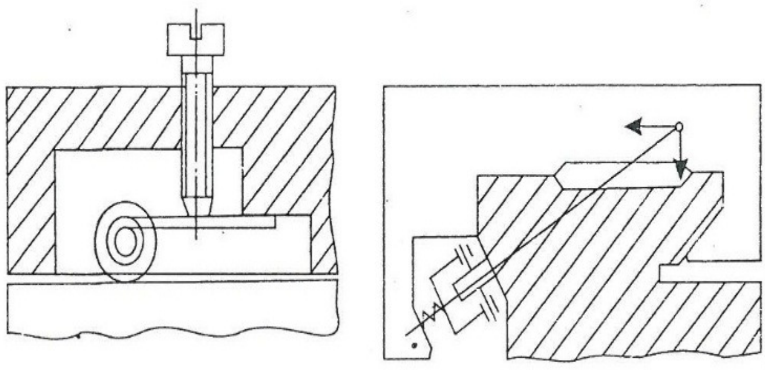
Slideways with Rolling members

In this type of slideways, a part of the normal load is taken up by spring-loaded rollers thus reducing friction between the sliding members. This results in

1. longer life of the slideways, and
2. greater accuracy of motion, particularly at low speeds due to less dominant influence of the stick-slip phenomenon.

MODULE 3

Schematic diagrams of combination guideways subjected to vertical load and compound load are shown in Fig. 4.48a and Fig. 4.48b, respectively. The total load is distributed between the slideway and the anti-friction way. The share of the rolling members in the total load can be regulated by the tightening of the leaf spring (Fig. 4.48a) or helical spring (Fig. 4.48b).



(a) (b)

Fig. 4.48 *Schematic diagram of combination guideways using anti-friction rolling members for (a) vertical loading (b) compound loading*

The friction force in this type of combination guideway is given by the expression,

$$F = nF_c + f \cdot N_1 + \frac{f_r}{r_{eq}} \cdot N_2 \tag{4.80}$$

Here N_1 and N_2 represent the share of normal load taken up by the slideway and the anti-friction way, respectively. The rest of the notation is the same as was used in Eqs (4.70) and (4.79).

Combination slideways with rolling members are employed when it is too difficult to provide hydraulic relief. Their most widespread application is in rotary tables of gear-hobbing machines and vertical boring and turret lathes, in which a part of the load is taken up by a central anti-friction thrust bearing. They may also be encountered in heavy-duty horizontal-boring and jig-boring machines and medium-sized lathes.

Protecting Devices for Slideways

Wear of slideway surfaces is seriously affected by dirt, abrasive particles and chips penetrating into the gap between them. The life of slideways can be considerably improved by employing devices which protect the slideway surfaces from foreign matter. Protecting devices can be broadly classified into the following three categories:

1. Seals
2. Covers
3. Intermediate steel strips.

1. **Seals** Seals are fixed at the ends of the moving member and pressed against the stationary guideways by means of clamping plates. The simplest seal consists of a block of felt (Fig. 4.49a). However, felt seals wear out quickly; their effectiveness can be improved by adding a rubber seal (Fig 4.49b). Still better protection

is achieved by adding a brass chip cleaner to the felt and rubber seals (Fig. 4.49c). The performance of all types of seals can be improved by inserting a leaf spring between the clamping plate and the seal (Fig. 4.49d).

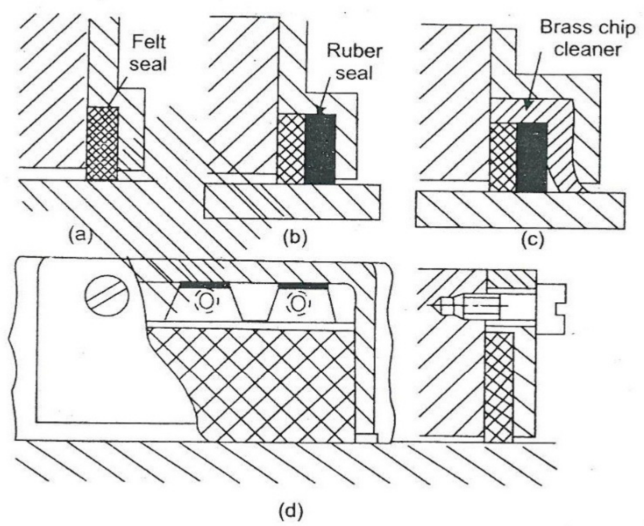


Fig. 4.49 *Various types of protective seals*

2. **Covers** If the stationary guideway is shorter than the moving member by an amount, equal to the maximum travel, then the slideway surfaces are always covered by the moving member. Generally, the situation is just the reverse, i.e., the moving member is shorter in length than the stationary guideway. In such cases, welded, cast or stamped cover plates must be provided at the ends of moving members (Fig. 4.50a). Such cover plates are employed on turret lathe carriages and grinding machine tables in which the stroke is relatively small. Telescopic covers (Fig. 4.50b) are employed when the moving member has a large stroke. Telescopic covers are made of 1.5–3.0 mm thick sheets of steel. In heavy-duty machine tools, the telescopic covers are also quite heavy and must be supported on rollers. In grinding machines where there is no chance of the protective device being damaged by large chips, concertina type leather covers are employed (Fig. 4.50c). The leather cover surrounds the otherwise open slideway surfaces and seals them off hermetically.

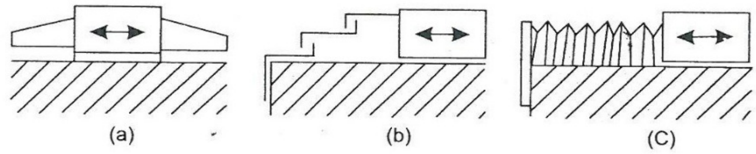


Fig. 4.50 *Various types of protective covers*

3. **Intermediate Steel Strips** Accurate thin steel strips of constant thickness may be inserted between the sliding surfaces to avoid exposing the accurately ground slideway surfaces to dirt, chips, etc. The steel strip is stretched tight by a tensile preload. Three methods of holding the protective strip are schematically depicted in Fig. 4.51. Protective strips are generally employed on machine tools with a long stroke.

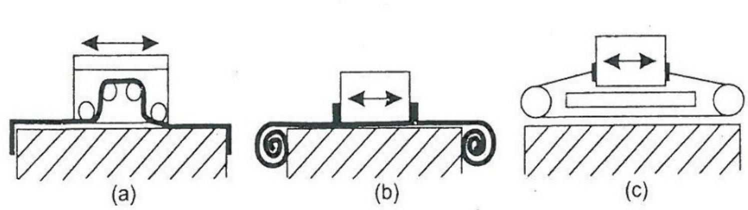


Fig. 4.51 *Various methods of holding the protective strip*

It should be noted that in machine tools using covers or intermediate steel strips, seals are used as additional protection.

MODULE 3

5 FUNCTIONS OF SPINDLE

The spindle is one of the most important elements of the machine tool.

The functions of the spindle of machine tool are as follows:

(a) It clamps the workpiece or cutting tool in such a way that the workpiece or cutting tool is reliably held in position during the machining operation.

(b) It imparts rotary motion or rotary cum translatory motion to the cutting tool or workpiece.

(c) It is used for centring the cutting tool in drilling machine, milling machine, etc. while it centres the workpiece in lathes, turrets, boring machine, etc. Requirements of spindle are :

   (a) The spindle should rotate with high degree of accuracy. The accuracy of rotation is calculated by the axial and radial run out of the spindle nose. The radial and axial run out of the spindle nose should not exceed certain permissible values. These values depend upon the required machining accuracy. The rotational accuracy is influenced mainly by the stiffness and accuracy of the spindle bearings especially by the bearing which is located at the front end.

   (b) The spindle unit must have high dynamic stiffness and damping.

   (c) The spindle bearing should be selected in such a way that the initial accuracy of the unit should be maintained during the service life of the machine tool.

   (d) Spindle unit should have fixture which provides quick and reliable centering and clamping of the cutting tool or workpiece.

   (e) The spindle unit must have high static stiffness. Maximum accuracy is influenced by the bending, axial as well as torsional stiffness.

   (f) The wear resistance of mating surface should be as high as possible.

   (g) Deformation of the spindle due to heat transmitted to it by workpiece, cutting tool, bearings etc. should be as low as possible. Otherwise it will affect the accuracy of the machining accuracy.

6 DESIGN OF SPINDLE

Figure 6 shows schematic diagram of spindle. A spindle represents a shaft with

(a) length '*a*' which is acted upon by driving force F_2, and

(b) cantilever of length '*m*' acted upon by external force F_1.

The spindle is basically designed for bending stiffness which requires that maximum deflection of spindle nose should not exceed a prespecified value, i.e.

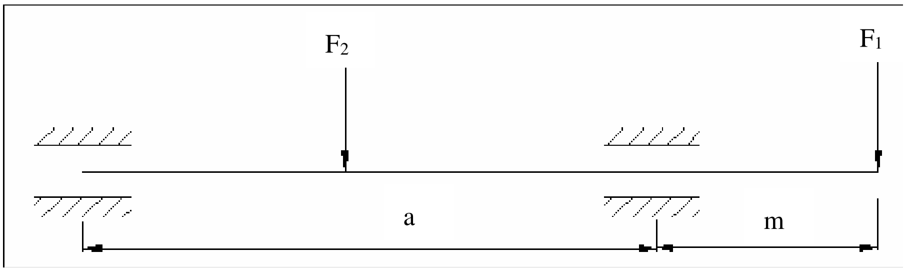


Figure 6 : Principle of Working of Spindle

$$d_{max} \leq d_{per} \quad \text{. . . (28)}$$

The total deflection of spindle nose consists of deflection d_1 of the spindle axis due to bending forces F_1 and F_2 and deflection d_2 of the spindle axis due to compliance of the spindle supports. When the spindle has tapered hole in which a center or cutting tool is mounted, the total deflection of the center or cutting tool consists of deflections d_1, d_2 and d_3 of the center or cutting tool due to compliance of the tapered joint.

6.1 Deflection of Spindle Axis due to Bending

To calculate the deflection of the spindle nose due to bending, one must establish a proper design diagram. The following guidelines may be used in this regard.

(a) If the spindle is supported on a single anti-friction bearing at each end, it may be represented as a simply supported beam, and

(b) If the spindle is supported in a sleeve bearing, the supported journal is analyzed as a beam on an elastic foundation; for the purpose of the design diagram the sleeve bearing is replaced by a simple hinged support and a reactive moment M_r acting at the middle of the sleeve bearing.

The reactive moment is given as :

$$M_r = C \cdot M$$

where M = bending moment at the support, and

C = constant = 0 for small loads and 0.3 to 0.35 for heavy load.

MODULE 3

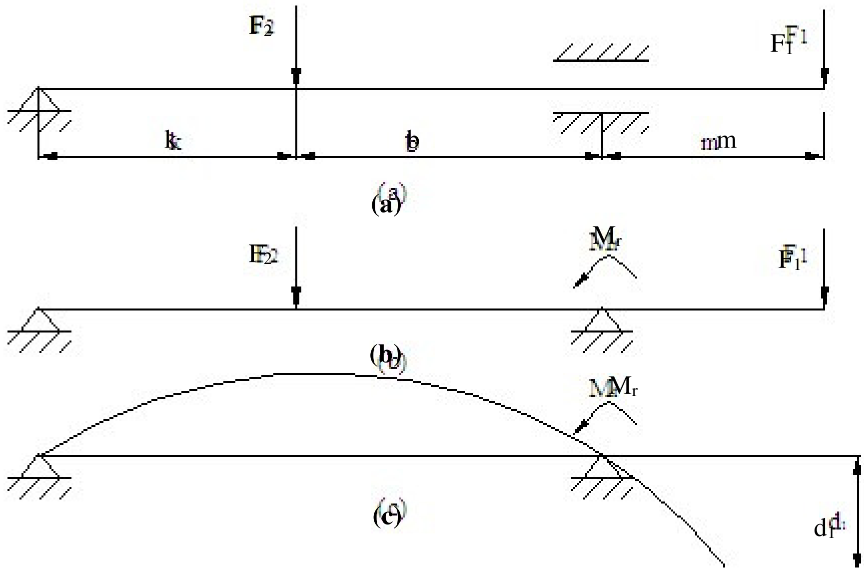


Figure 7 : Effect of Various Force on Spindle

Figure 7(a) shows schematic diagram of spindle. Figure 7(b) depicts the design diagram of the spindle and figure 7(c) illustrates deflected axis of the spindle.

Consider the spindle shown in Figure 7(a). By replacing the rear ball bearing by a hinge and the front sleeve bearing by a hinge and reactive moment M_r, the spindle can be reduced to the design diagram as shown in Figure 7(b). The deflection at the free end of the beam (spindle nose) can be determined by Macaulay's method and is found out to be

$$d_1 = \frac{1}{3EI_a}\left[F_1 m^2(a+m) - 0.5F_2 kbm\left(1-\frac{k}{a}\right) - M_r am\right] \qquad \ldots (29)$$

where E is Young's modulus of the spindle material.

I_a is average moment of inertia of the spindle section.

The deflection of the beam is shown in Figure 7(c).

6.2 Deflection of Spindle Axis due to Compliance of Spindle Supports

Let δ_E and δ_G represent the displacement of the rear and front support respectively. Owing to compliance support, the spindle deflects are shown in Figure 13.8. From similarity of triangles OHH' and OGG'

$$\frac{d2}{m+x} = \frac{\delta_G}{x}$$

$$\therefore \qquad d_2 = \left(1 + \frac{m}{x}\right)\delta_G \qquad \ldots (30)$$

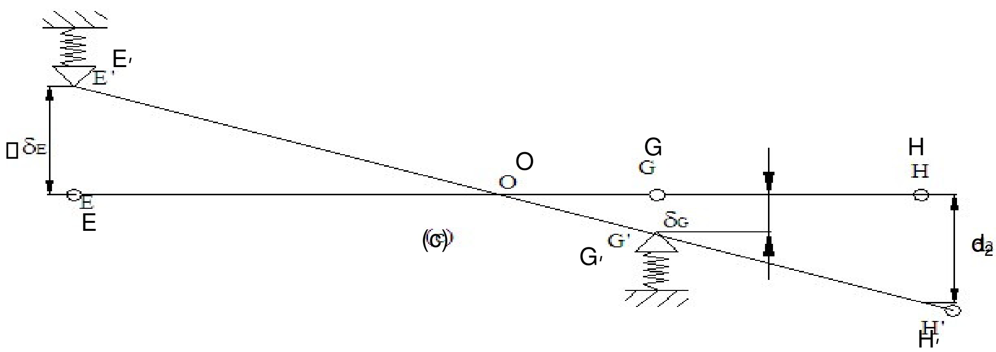


Figure 8 : Deflection of the Spindle due to Compliance of Support

From similarity of triangles *OEE*' and *OGG*', we get

$$\frac{\delta_G}{x} = \frac{\delta_E}{a-x}$$

On substituting these values of x in Eq. (30), d_2 changes to,

$$\therefore \quad d_2 = \frac{1+m}{a}\delta_G + \delta_E \frac{m}{a} \qquad \ldots (31)$$

Hence it is clear from above equation that displacement δ_G of the front bearing has greater influence upon deflection d_2 of spindle nose than displacement δ_E of the rear bearing.

$$\text{Displacement } \delta_E = \frac{R_E}{S_E}$$

R

and

$$\delta_G = \frac{R_G}{S_G}$$

Where R_E and R_G are the support reactions at E and G respectively.

S_E and S_G are stiffness at E and G respectively.

At equilibrium,

$$\sum M_E = 0$$

$$\therefore \quad R_G a - F_2 k + M_r - F_1 (m + a) = 0$$

$$\therefore \quad R_G = F_2 k - M_r + F_1 (m + a)$$

Similarly $\quad \sum M_G = 0$

$$\therefore \quad R_E a - F_2 b - M_r + F_1 m = 0$$

$$\therefore \quad R_E = \frac{(F_2 b + M_r - F_1 m)}{a}$$

$\therefore$Deflection,

$$d_2 = \frac{F_2 k - M_r + F_1 (m+a)}{a.S_b}\left(1 + \frac{m}{a}\right) + \frac{F_2 b + M_r - F_1 m}{a.S_a}\left(\frac{m}{a}\right) \qquad \ldots (32)$$

Hence total deflection d (shown in Figure 9) is obtained as

$d = d_1 + d_2$

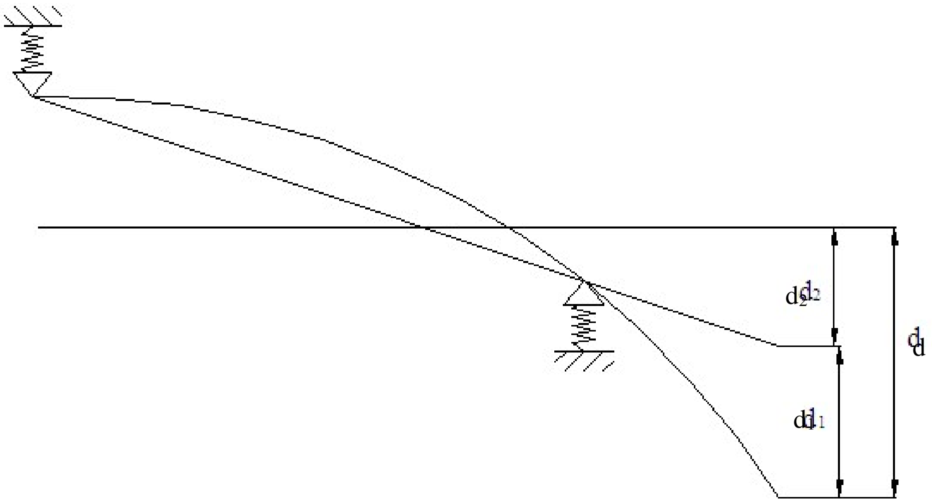


Figure 9 : Total Deflection of Spindle Axis

6.3 Optimum Spacing between Spindle Supports

The ratio 'τ' is an important parameter in spindle design. *a*

Where, □= $\frac{a}{m}$. The optimum value of this ratio is the one that makes sure that minimum

Total deflection'd' can be determined from differentiating it partially with respect to τ.
For minimum deflection *d*, $\frac{dd}{d\tau}$= 0.
The point of the minimum of the $d_1 + d_2$ curve, gives the optimum value of ratio a/m which generally lie between 3 and 5. The value of τ depends upon ratio of stiffness opt

of the front and rear bearings, □= $\frac{s_E}{S_G}$and factor $J = \frac{S_E I_E}{S_G I_G}$

Where $S_E = \frac{3EI_m}{m^3}$ = bending stiffness of the cantilever.

I_m = average moment of inertia of the spindle over cantilever.

I_a = average moment of inertia of the spindle over the supported length.

An opposite constraint on maximum span stems from the requirement that for normal functioning of the spindle driving gear, the stiffness of the span should not be less than 245-260 N/µm. This constraint is expressed through the following relationship:

$$a \leq \frac{d_a^{4/3}}{i^{1/3}}$$

When the spindle is supported on hydrostatic journal bearings, the maximum deflection at the middle of the span should satisfy the condition:

$$d_{a\max} \leq 10^{-4} a$$

And the maximum span length a_{max} should be limited by the above constraint. This constraint is based upon the requirement that the maximum misalignment due to deflection of the journals should not exceed one third of the bearing gap.

6.4 Deflection due to Compliance of the Tapered Joint

The spindle ends of most machine tools have a tapered hole for accommodating a center (in lathe) or cutting tool shanks (in milling and drilling machine). The deflection of center or shank at a distance *d* from the spindle axis where the force *F* is acting is given by equation :

$$d_3 = □ + □d$$

where □= displacement of the shank or center at the edge of taper due to contact compliance, and

□ = angle of slope of the shank or at the edge of taper.

If manufacturing the errors of the taper are ignored, □ and □ can be calculated from following equations :

$$\Delta = \frac{4\varphi D C_1}{\pi D}(\varphi d C_2 + C_3), \mu m$$
$$\emptyset = \frac{4F\varphi^2 C_1}{\pi D}(2\varphi d C_4 + C_2)$$

where C_1 = coefficients of contact compliance

C_2, C_3, C_4 = coefficients that account for the diameter variation along the length of taper.

$$\Psi = \left(\frac{1}{2.3 C_1 D^4}\right)^{1/4}, cm^{-1}$$

D and *d* are expressed in cm.

Generally displacement □ due to contact compliance can be ignored in comparison with the displacement due to bending of the shank or center.

Hence, $$d_3 = \frac{4F\psi^2 C1}{\pi D}(2\psi d C4 + C2)\, d, \mu m \qquad \text{. . . (33)}$$

6.5 Permissible Deflection and Design for Stiffness

The deflection d_1, d_2 and d_3 are calculated from equations (29), (32) and (33) respectively. The total deflection of the spindle nose can be determined as the sum of three deflections. The design of stiffness can be carried out in accordance with equation (13.28). The permissible deflection of the spindle nose d_{per} depends upon the machining accuracy required for the machine tool. Generally, it should be less than one-third of the maximum permissible tolerance on radial run out of the spindle nose. The machining accuracy depends not only upon the radial stiffness but also upon its axial and torsional stiffness. The axial displacement of the spindle unit consists of the axial deformation of the spindle and the deformation of the spindle thrust bearing. The torsional stiffness of machine tool spindles significantly influences the machining accuracy in metal removal operations, such as gear and thread cutting in which the feed and primary cutting motions are kinematically linked. The torsional deformation of the spindle unit consists of the deformation of the spindle and deformation of drive components. As stated earlier, spindles are designed for stiffness, primarily radial. However in heavily loaded spindles the stiffness design must be proved by a strength check against fatigue failure. The strength check requires that

$$f \geq f_{\min}$$

where f = factor of safety against fatigue failure.

$f_{\min}$ = minimum value of safety factor

= 1.3 to 1.5

If spindles are subjected to combined bending and torsion, the factor of safety f is determined from the equation,

$$f = \frac{(1-\eta^4) d_e^3 \sigma_e}{10\sqrt{(kM_b)^2 + (bT)^2}} \quad \ldots (34)$$

where $\eta = \frac{d_i}{d_e}$ =ratio of the internal diameter to the external d_e

d_i = internal diameter of the spindle, mm

d_e = external diameter of the spindle, mm

σ_e =Endurance limit of the spindle material, N/mm^2

M_b = mean value of the bending moment acting on the spindle, N. mm

T = mean value of the torque acting on the spindle, N. mm

k = coefficient that accounts for variation of bending moment and stress concentration.

b = coefficient that accounts for variation of torque and stress concentration.

Coefficient k can be determined from the following expression :

$$k = U_\sigma (1 + Q)$$

where U_σ= dynamic stress concentration coefficient for normal stresses

= 1.7 to 2.0

$Q = \frac{M_{ba}}{M_b}$ = ratio of amplitude of the bending moment to its average value.

Coefficient b can be calculated from the following formula :

$$b = \frac{\sigma_e}{\sigma_y} + U_\tau Q_t \quad \ldots (35)$$

where □ = yield stress of the spindle material. y

$U_{\square}$= dynamic stress concentration coefficient for shearing stress.

= 1.7 to 2.0

$$Q_t = \frac{T_a}{T} = \text{ratio of amplitude of torque to its average value.}$$

The values of Q and Q_t generally depend upon the machining conditions. For example, in a super-finishing operation there is virtually no variation of the bending moment and torque, i.e. $M_{ba} = T_a = 0$ and therefore $Q = Q_t = 0$.

SAQ 2

(a) What are functions and requirements of spindle?
(b) Design the spindle for machine tool.
(c) Explain the deflection of spindle due to bending and spindle support.

7 SUMMARY

The guideway is used to ensure moving of the cutting tool or machine tool operative element along predetermined path. Slideways and anti friction slideways are two types of guideways. Slideway is basically designed for wear resistance and stiffness. The pressure acting on mating surfaces create wear. Hence it is essential to resist wear. The design for stiffness specifies that the deflection of cutting edge due to contact deformation of slideways should not exceed permissible value, which is required to maintain the accuracy. Spindle is generally used for centering and clamping the workpiece, and imparting rotary motion.

Spindle is basically designed for bending stiffness.

8 KEY WORDS

Guideways **:** Its function is to ensure that the cutting tool or machine tool operative element moves along predetermined path.

MODULE 4

MACHINE TOOL VIBRATION AND DAMPING

- Introduction
- Types of vibration
- Effects of vibration
- Sources of vibration excitation
- Vibration control in machine tool
- Damping
- Conclusion

Introduction

- Machining and measuring operations are invariably accompanied by vibration.
- To achieve higher accuracy and productivity vibration in machine tool must be controlled.
- For analysis of dynamic behavior of machine tool **rigidity** and **stability** are two important characteristics.

Types of vibration

- Machine tool vibrations may be divided into 3 basic types:-

1. Free or transient vibration
2. Forced vibration
3. Self-excited vibration (Machine tool chatter)

Effects of vibration

- **The effect of vibration on machine tool:-**

Vibration may lead to complete or partial destruction of machine tool.

<u>Disturbing forces</u>

Origin	**Point of action**
Cutting process	Work piece cutting tool
Bearing, belts etc	Bearings of shafts
Motors	Flanges, shafts
Drives	Guides, gear drives

MODULE 4

- **The effect of vibration on the cutting condition**

Three effects on cutting condition

1. Chip thickness variation effect
2. Penetration rate variation effect
3. Cutting speed variation effect

- **The effect of vibration on the work piece**

➢ The major effect is poor surface finish.

➢ It is very important for grinding machine.

➢ Dimensional accuracy of job is also effected.

➢ This is mostly due to chatter vibration.

➢ Chatter marks are proof for the effect of vibration on the work piece

- **The effect of vibration on tool life**

Tool life = $K / (V^{x} . s^{m} . t^{l})$

V-velocity

S-uncut chip thickness

t-width of cut

Ceramic and diamond tools are sensitive to impact loading.

Sources of vibration excitation

- **Inhomogeneous work piece material**

➢ Hard spots or a crust in work piece leads to free vibrations.

➢ Discontinuous chip removal results in fluctuation of the cutting thrust.

➢ The breaking away of a built-up edge from the tool face also impart impulses to the cutting tool

- **Variation in chip cross section**

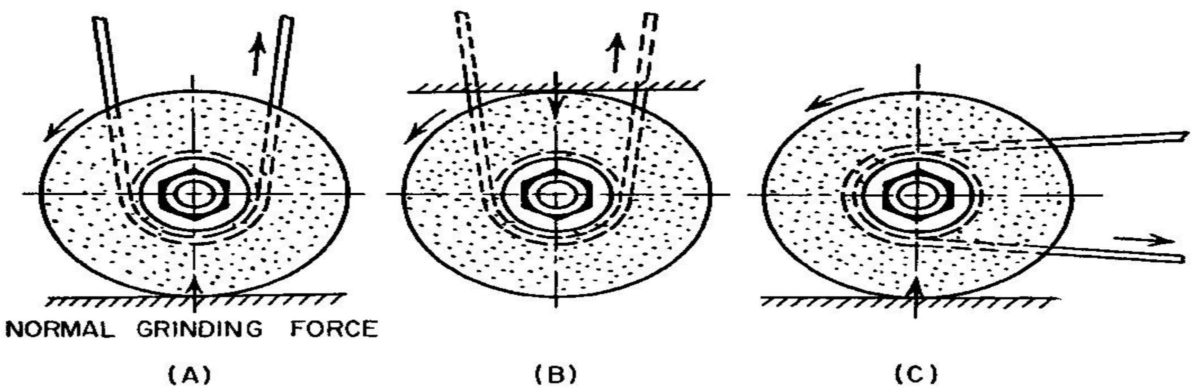
➢ Variation in the cross-sectional area is due to the shape of the machined surface or due to the configuration of tool.

➢ Pulses imparted to the tool and job.

- Pulses have shallow fronts for turning of eccentric parts and steep fronts for slotted parts and for milling/broaching.
- Bouncing of the cutting tool on machined surface can be minimized by closing the recess with a plug or with filler

Disturbances in the work piece and tool drives

- Forced vibration induced by rotation of unbalanced member affect the surface finish & tool life. This can be eliminated by careful balancing or by self-centering.
- Rotating components should be placed in position where effect of unbalance is less.
- Electric motors produce both rectilinear and torsional vibration.
- Rectilinear vibrations are due to a non-uniform air gap between the stator & rotor, asymmetry of windings, unbalance, bearing irregularities, misalignment with driven shaft
- Torsional vibration is due to various electrical irregularities.
- Gear induced vibration are due to production irregularities, assembly errors, or distortion of mesh caused by deformation of shafts, bearings and housing under transmitted load.
- All gear faults produce non-uniform rotation which effect the surface finish & tool life.
- Belt drives are used as filters to suppress high frequency vibration, can induce their own forced vibration.
- Any variation in effective belt radius change belt tension and belt velocity
- This causes a variation of the bearing load and of the rotational velocity of the pulley.
- Another source of belt-induced vibrations is variation of the elastic modulus along the belt length.
- Flat belts generate less vibration than V belts because of their better homogeneity and because the disturbing force is less dependent on the belt tension.



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A) Vibration is minimized when belt tension and normal grinding force point in the same direction.

(*B*) Large amplitudes may arise when the normal grinding force is substantially equal to the belt tension.

(*C*) Vibration due to centrifugal force is likely to be caused by an unbalance of the wheel.

- Uniformity of feed motions is disturbed by, stick-slip.
- The occurrence of stick-slip depends on the interaction of the following factors:-

(1) The mass of the sliding body

(2) The drive stiffness

(3) The damping present in the drive

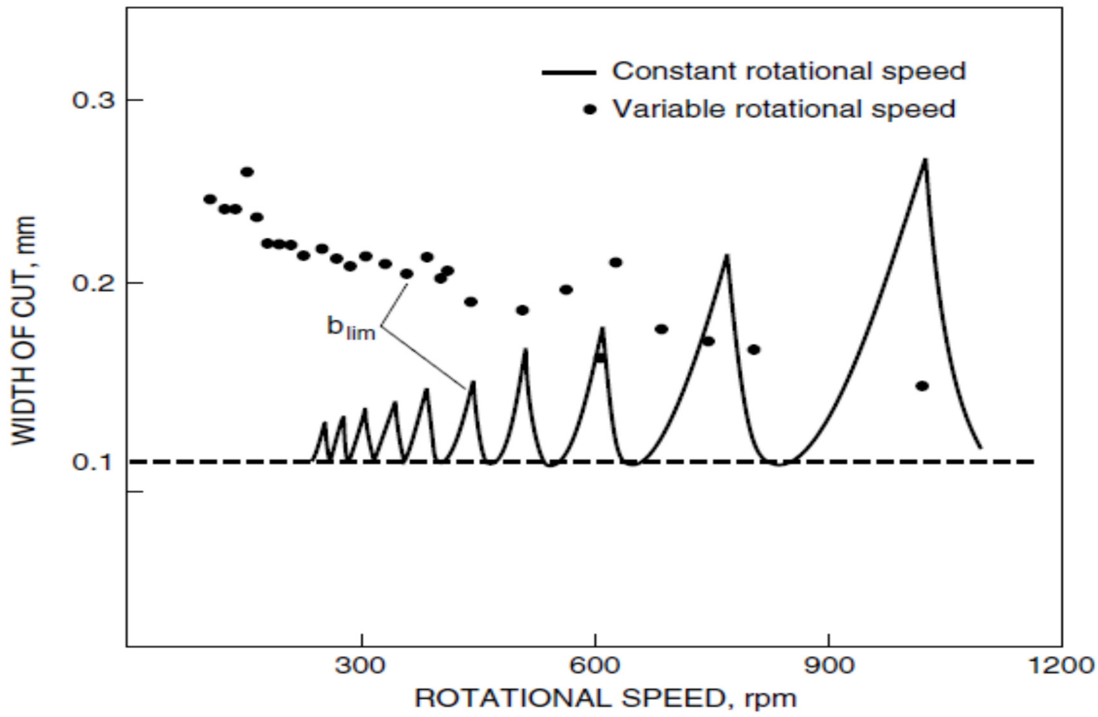
(4) The sliding speed

(5) The surface roughness of the sliding surfaces, and

(6) The lubricant used. It is encountered only at low sliding speeds;

- IMPACTS FROM MASSIVE PART REVERSALS
- Reversal of reciprocating parts produce sharp impacts, which excite both low-frequency solid-body vibrations of the machine and high-frequency structural modes.
- Occur in machine tool like surface grinder and in CNC, CMM.
- The driving forces units have impulsive character and cause free decaying vibrations in both solid-body and structural modes.
- These vibrations excite relative displacements in the work zone between the work-piece and the cutting or measuring tool.
- Reduction in the adverse effects of the impulsive forces can be achieved by enhancing the structural stiffness and natural frequencies
- Increase of structural damping as well as damping of mounting elements (vibration isolators) also results in a reduction in the decay time.
- VIBRATION TRANSMITTED FROM THE ENVIRONMENT
- Shock and vibration generated in presses, machine tools, internal-combustion engines, compressors etc., are transmitted through the foundation to other machines, which they may set into forced vibration.
- Vibration transmitted through the floor may be reduced by vibration isolation.
- MACHINE-TOOL CHATTER
- *Chatter* is a self-excited vibration which is induced and maintained by forces generated by the cutting process.

- It effects surface finish, tool life, production rate and also produce noise.
- Chatter resistance of a machine tool is usually characterized by a maximum stable (i.e., not causing chatter vibration) depth of cut b_{lim}.



- Machine-tool chatter is essentially a problem of dynamic stability. A machine tool under vibration-free cutting conditions may be regarded as a dynamical system in steady-state motion. Systems of this kind may become dynamically unstable and break into oscillation around the steady motion

Vibration control in machine tool

- The vibration behaviour of a machine tool can be improved by
- A reduction of the Intensity of the sources of vibration
- By enhancement of the effective static stiffness and damping.
- By appropriate choice of cutting regimes, tool design, and work-piece design.
- Abatement of the sources is important mainly for forced vibrations.

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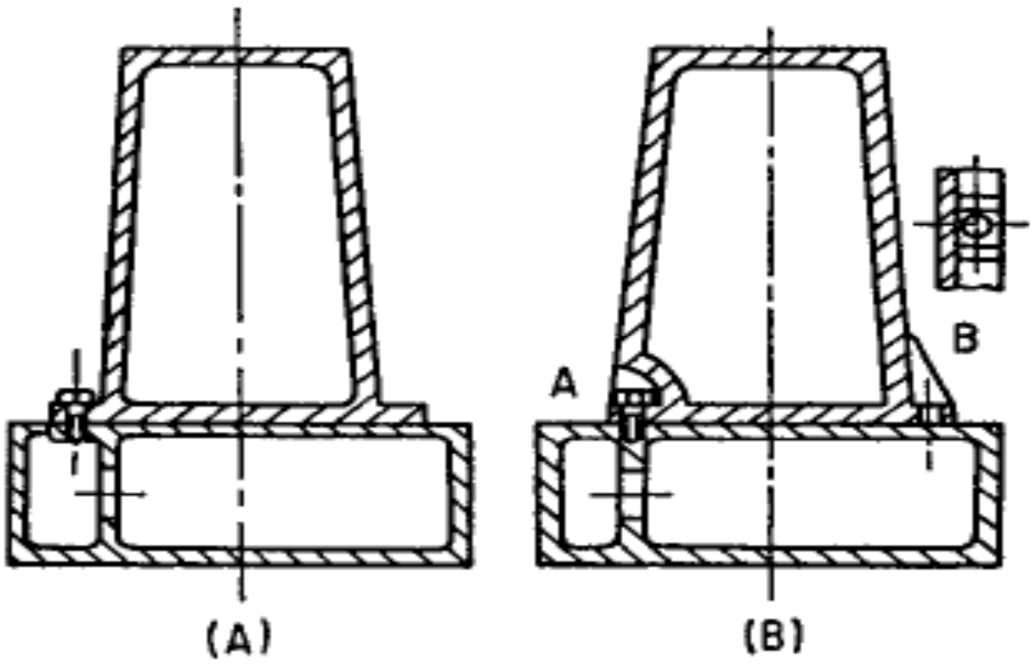
- Stiffness and damping are important for both forced and self-excited (chatter) vibrations.
- Both parameters, especially stiffness, are critical for accuracy of machine tools, stiffness by reducing structural deformations from the cutting forces, and damping by accelerating the decay of transient vibrations.
- Application of vibration dampers and absorbers is an effective technique for the solution of machine-vibration problems.
- Static stiffness k_s is defined as the ratio of the static force P_o, applied between tool and work-piece, to the resulting static deflection A_s between the points of force application.
- A force applied in one coordinate direction is causing displacements in three coordinate directions; thus the stiffness of a machine tool can be characterized by a stiffness matrix.
- Only one or two stiffness are measured to characterize the machine tool.
- Frame parts are designed for high stiffness.
- Damping is determined mainly by joints especially for steel welded frames. Cast iron parts contribute more to the overall damping, while material damping in polymer-concrete and granite is much higher.
- The stiffness of a structure is determined primarily by the stiffness of the most flexible component in the path of the force.
- In many cases the most flexible components of the breakdown are local deformations in joints, i.e., bolted connections between relatively rigid elements such as column and bed, column and table, etc.



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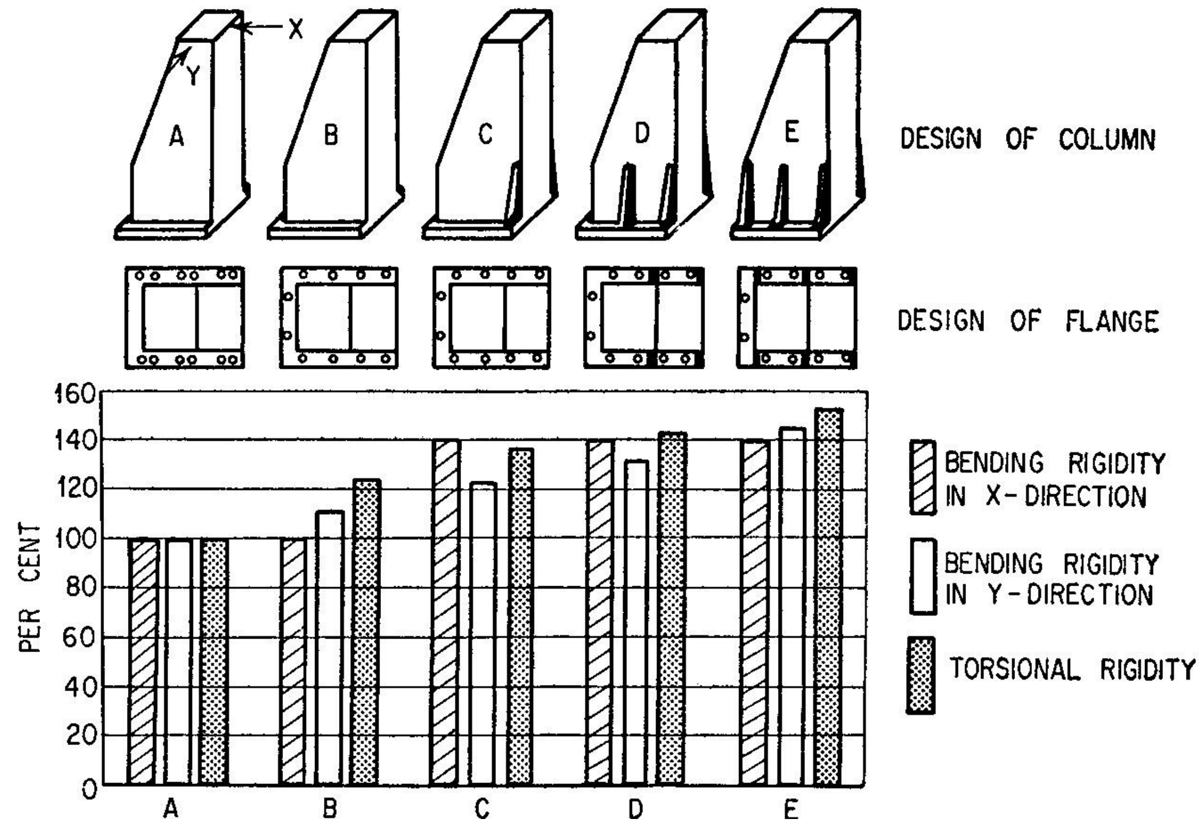
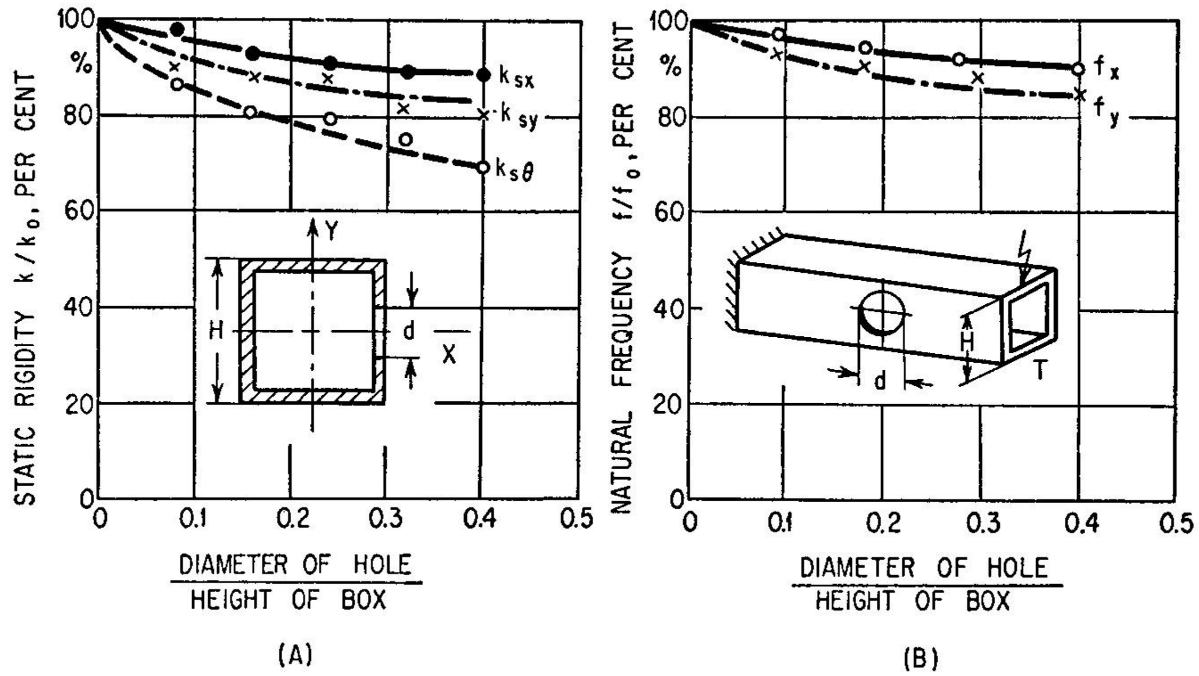
(*A*) Old design, relatively flexible owing to deformation of flange.

(*B*) New design, bolt placed in a pocket (*A*) or flange stiffened with ribs on both sides of bolt (*B*).





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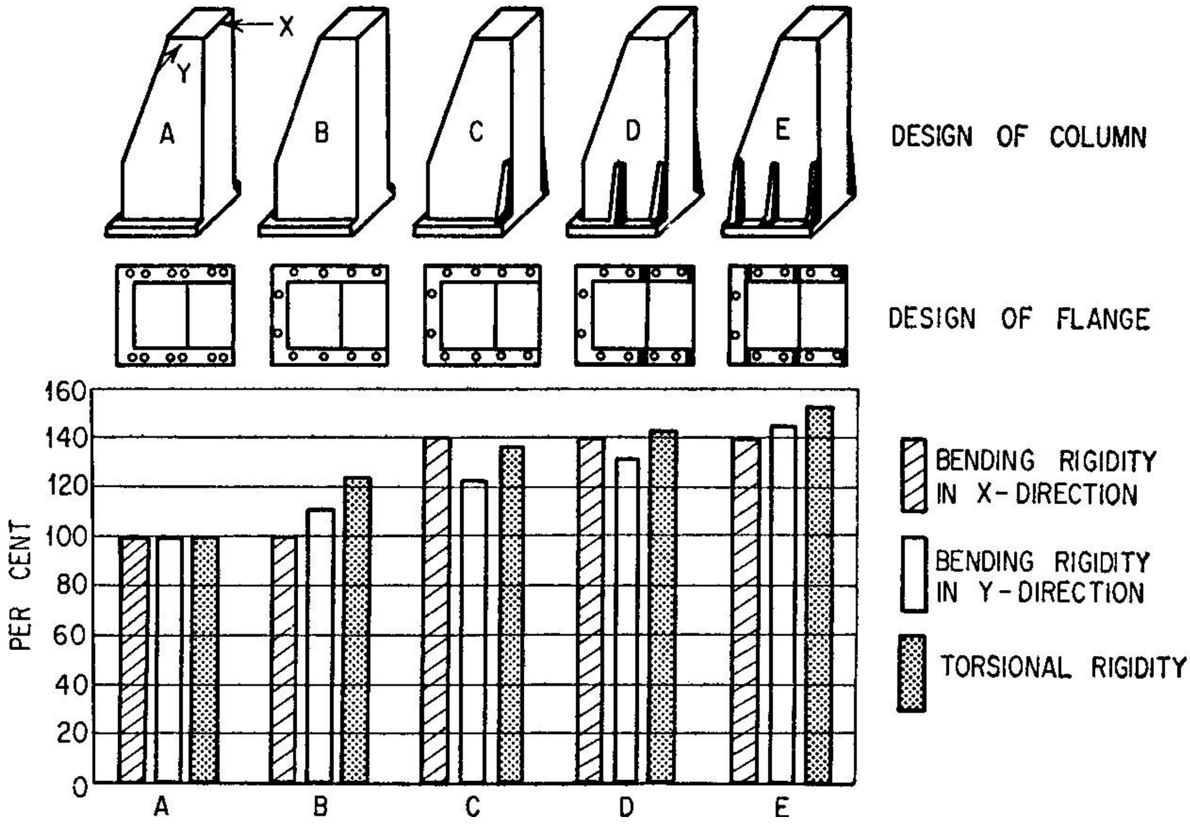
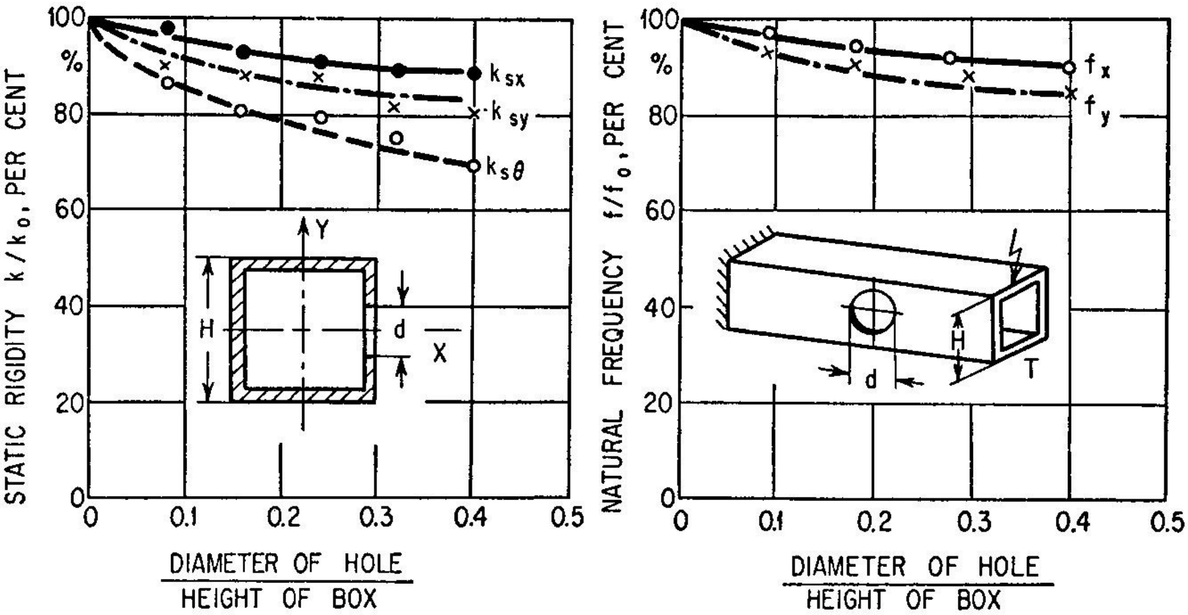

X
Y
A
B
C
D
E
DESIGN OF COLUMN
DESIGN OF FLANGE
160
140
120
100
80
60
40
20
0
PER CENT
BENDING RIGIDITY IN X-DIRECTION
BENDING RIGIDITY IN Y-DIRECTION
TORSIONAL RIGIDITY


STATIC RIGIDITY k/k₀, PER CENT
100 %
80
60
40
20
0
0.1
0.2
0.3
0.4
0.5
k sx
k sy
k sθ
Y
H
d
X
DIAMETER OF HOLE / HEIGHT OF BOX
NATURAL FREQUENCY f/f₀, PER CENT
f x
f y
T
DIAMETER OF HOLE / HEIGHT OF BOX

(A) (B)

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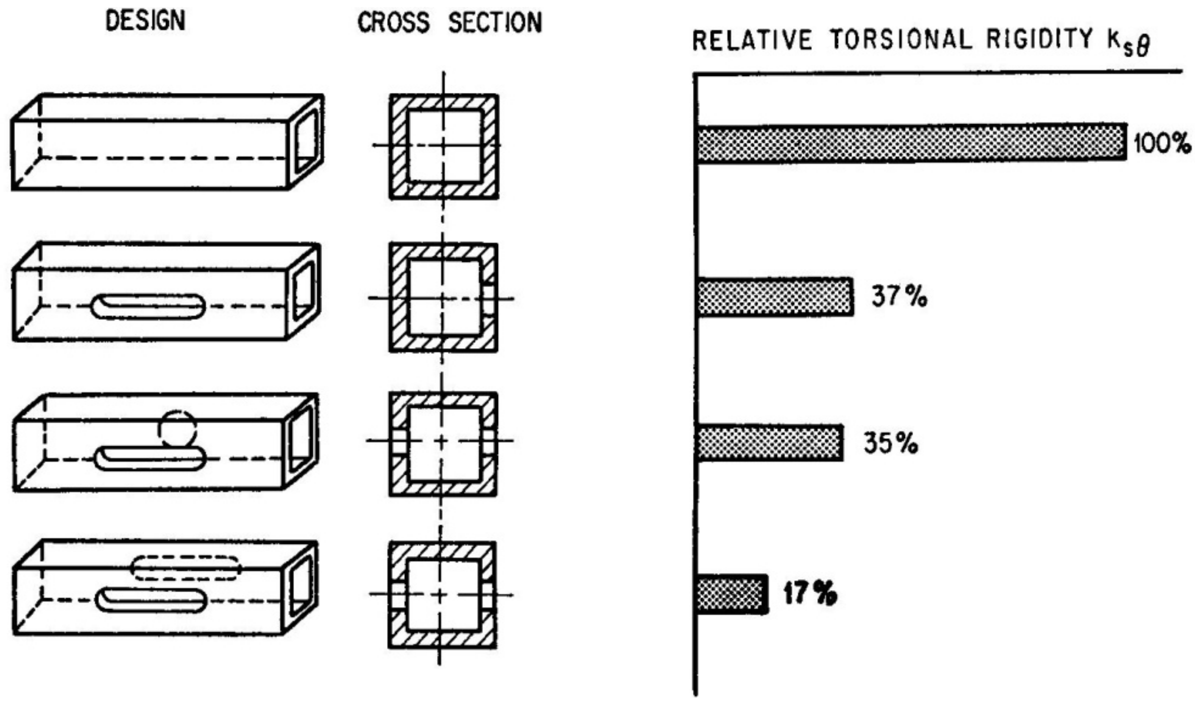
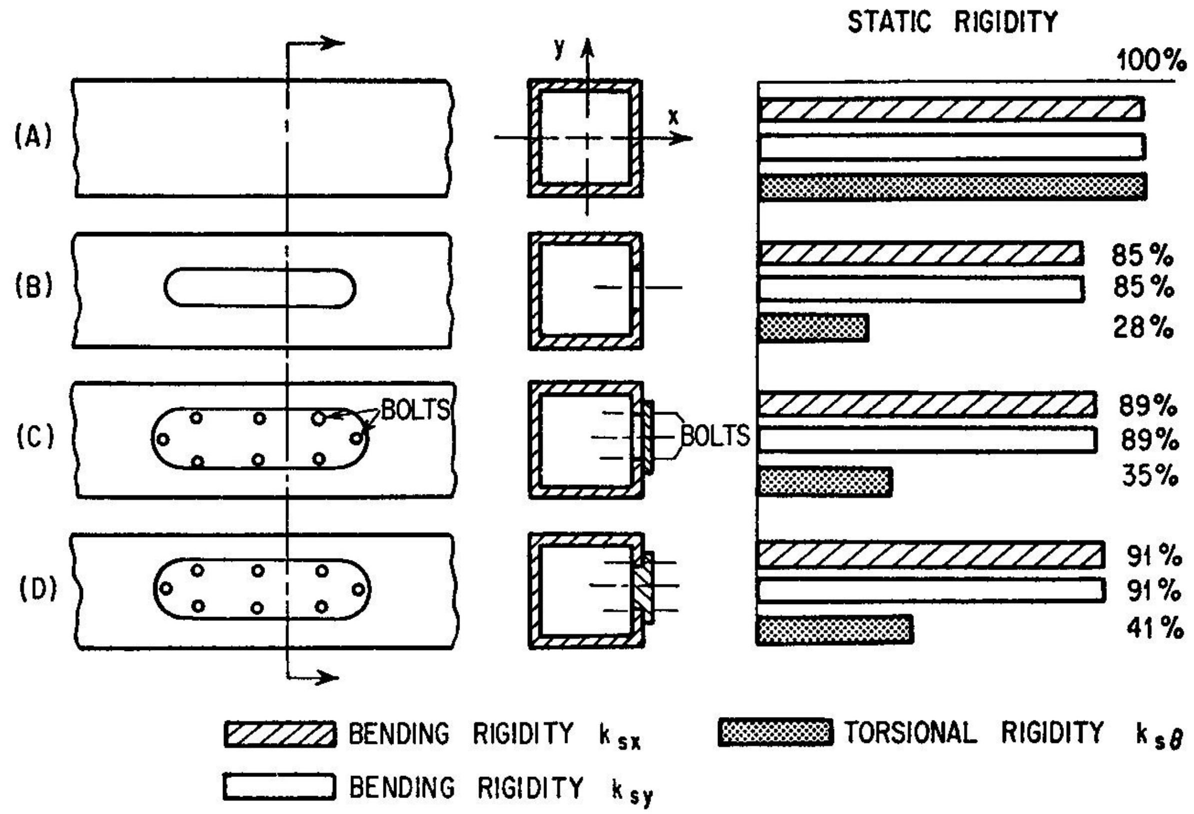
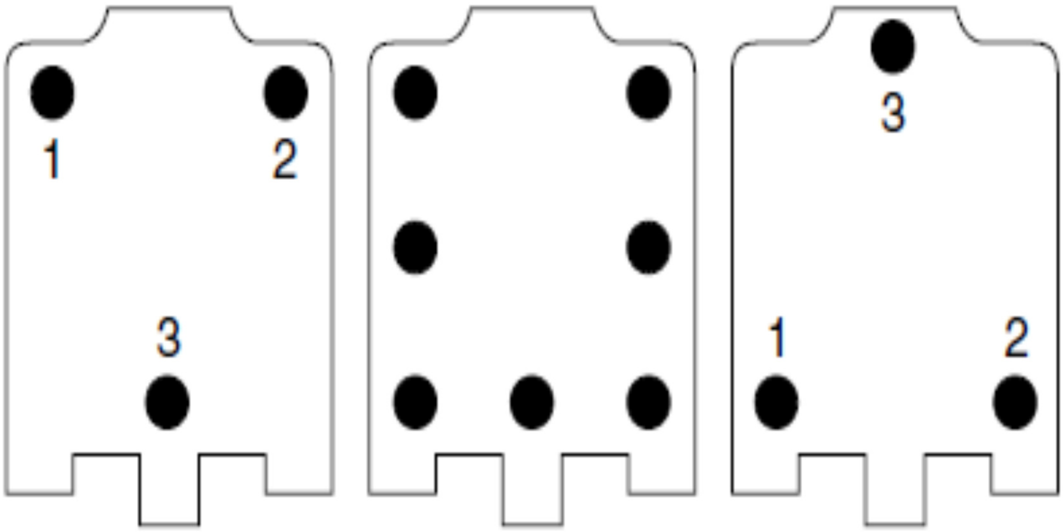




- Welded structural components are usually stiffer than cast iron components but have a lower damping capacity.

- A considerable increase in damping can be achieved by using interrupted welds, but at a price of reduced stiffness.



Tool Design

- Sharp tools are more likely to chatter than slightly blunted tools.
- Since narrow chips are less likely to lead to instability, a reduction of the approach angle of the cutting tool results in improved chatter behaviour.
- With lathe tools, an increase in the rake angle may result in improvement
- In tools having multiple cutting edges by making the distance between the adjacent cutting edges non-equal and/or making the helix angle of the cutting edges different for each cutting edge.

Variation of Cutting Conditions

- small increase or decrease in speed may stabilize the cutting process.
- In high-speed CNC machine tools, this can be achieved by continuous computer monitoring of vibratory conditions.
- An increase in the feed rate is also beneficial in some types of machining (drilling, face milling, and the like).

Damping

- The major part of the damping results from the interaction of joined components at slides or bolted joints.
- The interaction of the structure with the foundation or highly damped vibration isolators also may produce a noticeable damping.

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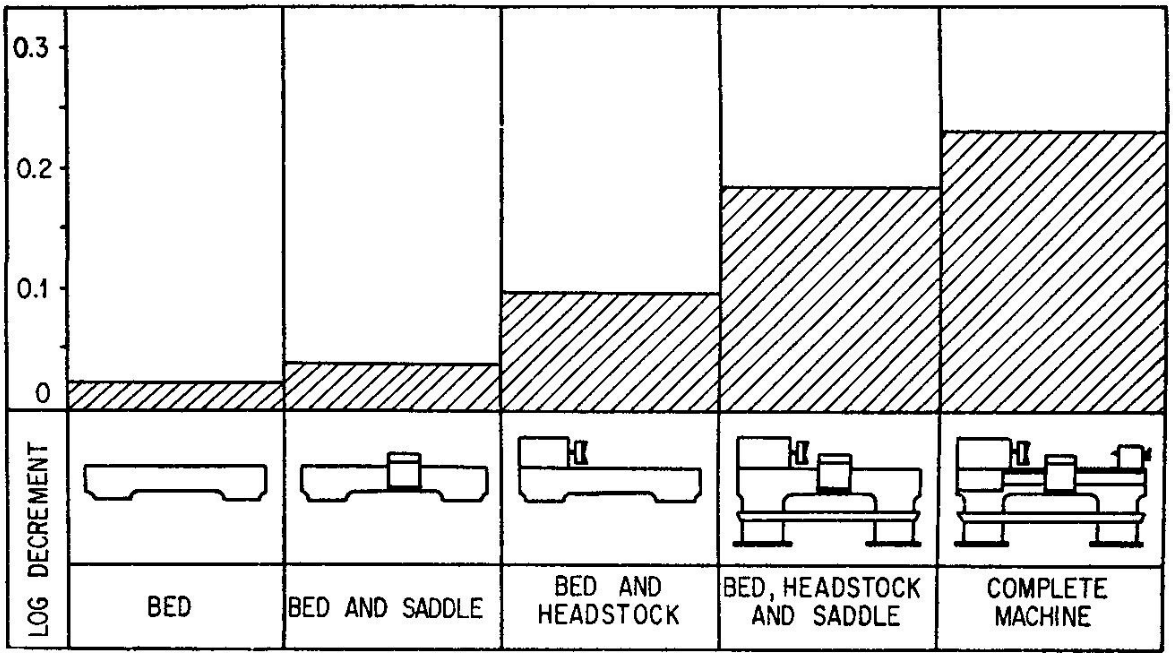
- structural damping is significantly higher for frame components made of polymer-concrete compositions or granite.



- A significant damping increase can be achieved by filling internal cavities of the frame parts with a granular material.
- For cast parts it can also be achieved by leaving cores in blind holes inside the casting.
- Damping can be increased by the use of dampers and dynamic vibration absorbers.
- Damping can be achieved by placing auxiliary longitudinal structural members inside longitudinal cavities within a frame part.
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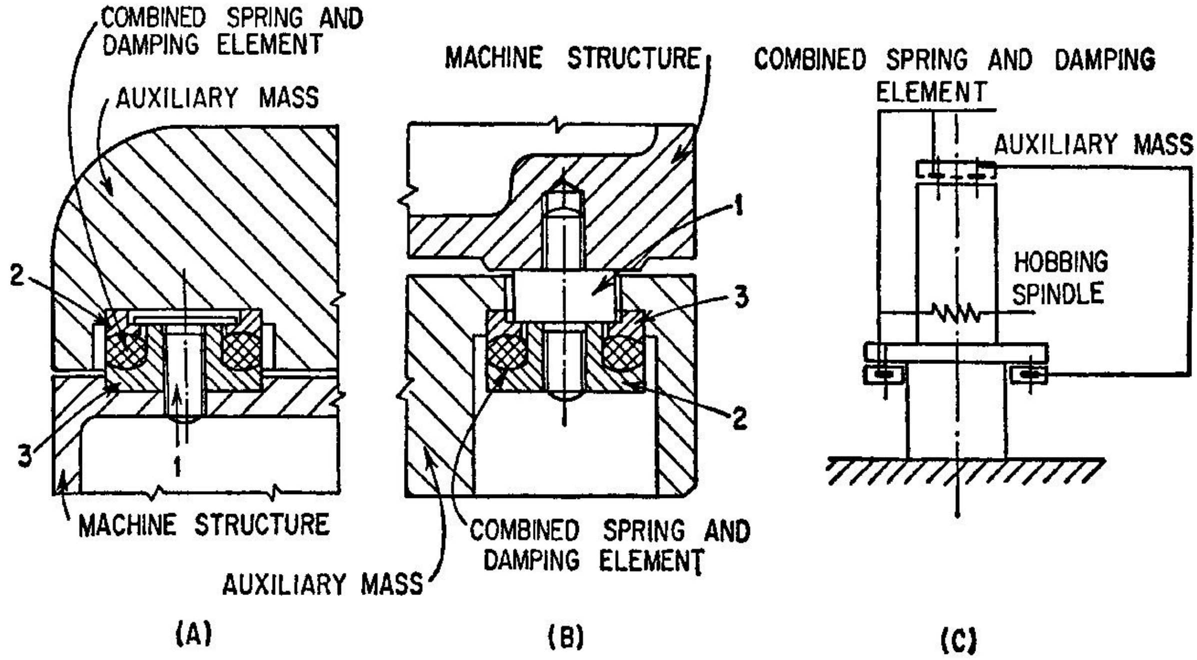

- A variation of the *Lanchester damper* is frequently used in boring bars to good advantage.