The Machinist Handbook for Precision Machining and Equipment Maintenance.

By Bhagwati Prasad Gupta
Preface:

Today most of the countries are heading towards industrial liberalization in order to compete in the global market. The opening up of the economy has made the market of the developing countries’ more attractive for international companies. It would mean increased competition. So the manufacturing companies would like to focus on certain diversification areas which would enable introduction of a steady stream of new products to meet the long term objectives of growth and profitability by maximum utilization of existing machine tools and equipment with minimum additional investment. For the success in the export front one will have to show its machining capability to meet international standards of quality which demands the maintenance of accuracy and other related parameters of the existing machining equipment.

To meet the challenge, each machinist is required to be trained in precision machining besides developing the knowledge of engineers and technologists working on the shop floor. For this, different types of books on machining and machinist handbooks need to be referred as information required are normally not available in machining manuals supplied by the manufacturers along with the machine-tools and equipment. It was therefore felt necessary to compile all such study material and the data in the form of a handy book to enable them refer during the process of learning for achieving better results while dealing with machine tools and equipment.

To start with different types of motions, required in machine tools for different types of machining processes have been described. Cutting parameters related with different types of material, have been tabulated to facilitate the calculation of cutting forces coming onto the cutting tools and the jobs being machined, for estimating the required torque and the power.

Based on the above data only, type of motor and drive could be selected and overloading could be prevented. So various types of D.C. and A.C. variable speed drives have been described starting from the conventional Ward Leonard system, thyrister, SCR servo-drive to inverter drive, vector control and brush less drives for controlling the motions of D.C. and A.C. motors including induction, synchronous and brushless type. For regulation of speed, different types of speed and feed gearboxes along with the mode of speed change in steps have been given.
Now days stepless variable speed drives are preferred. So PIV, ball disc drives and hydraulic stepless drive like hyvari drives have also been described. Details of different types of planetary gear boxes and cyclo drive are given along with the geared-motor. Backlash-free gearing arrangement and timing belts are the essential features of CNC machines. Mechanisms for rapid traverse, reversing, periodic intermittent motion and rectilinear motion have been given along with precision rack, worm rack-pinion, lead screw and ball screw arrangements.

As regards to assemblies and systems, the design aspects of spindle unit with different types of arrangement of sliding bearings, hydrostatic and air bearing and rolling bearings have been described. Frame-housings, slide-ways, low friction guide ways including synthetic linings, hydrostatic and pneumatic guide ways, rolling ways e.g. tychoways and LM guides have been given in detail. In addition to the conventional friction clutches and rigid couplings, latest design clutches like toothed clutches, over running clutches, universal coupling, gear, compression, bellow couplings and safety clutches and protective devices have been explained in detail.

The chapter on control system includes not only the lever and pre-selection of speed but also the automatic controls like logic switching, PLC, numerical control and CNC system. Latest lubrication system like centralized automatic lubrication including metering cartridge and monitoring units have been described in addition to the conventional manual, splash and pressurized lubrication devices. In the Hydraulic system, not only the industrial circuit has been touched but its components like various types of pumps, valves and actuators etc. have been given in much detail.

Besides the details of the mechanisms illustrated by more than 175 figures, due care has been taken to indicate the maintenance aspects of all such systems, assemblies and major components. Mathematical formulae and equations have been provided along with technical data in more than thirty tables which can be used by the designers and application and maintenance engineers for verification purposes. This has made this book useful not only for students of industrial machines but also for the machinists, technologists and practicing engineers in the field of design, manufacturing and maintenance.
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Chapter 1: MOTIONS IN MACHINE TOOLS

1.1 PRIMARY MOTIONS AND AUXILIARY MOTIONS

To obtain a part of the required shape and size on a machine tool, certain co-ordinated motions must be imparted to its working members. The cutting edge of the tool should move in a particular manner with respect to the work-piece being machined. Depending upon their participation in the process of formation of the required shape these motions could be classified into either primary or auxiliary.

**Primary Motion**
Primary motions include the principal or cutting movement and the feed movement. They serve the purpose of removing metal (chips) from the work piece. The speed of the principal movement determines the optimum cutting speed, while the speed of feed movement gives shape and required degree of surface finish. Working motions are of two types: rotary and translatory. Primary motions come from main drive. Motion for feed movement may be taken either from main drive or separate feed motor.

**Auxiliary Motion**
These are used to prepare the process of cutting and ensure successive machining operations.
- a) Motion for changing speeds and feed.
- b) Setting up motions according to the work piece shape and size.
- c) Control motions in the process of operation.
- d) Motions for feeding and clamping mechanism for work piece.
- e) Motions to clamp and unclamp certain machine members.
For obtaining these motions different types of drives and motion transmitting and converting mechanisms are required depending upon types of operations to be performed.

1.2. SOURCE OF MOTION

Main drive is the main source for primary motion and constitutes the arrangement from the motor to the spindle, which varies depending upon the specification and type of the machine. Drives for spindle as well as feed are required to:
- a) Transmit the power and torque required for the cutting process.
- b) Provide for engagement disengagement and reversal of principal motion.
- c) Ensure highly accurate and smooth motion at all available speeds.
Main drives can be divided into two groups:
- i) Drives for producing rotating movements.
- ii) Drives for producing rectilinear movements.
The first group includes devices that transform the rotation of an input shaft driven by an electric motor into a rotation of an output shaft (e.g. main spindle, cam- shaft etc.) at the required speed and in the desired direction. The devices of the second group transform a rotational input movement into a straight line reciprocating movement of table, ram etc. The drives should be able to transmit the required forces and torque at the desired speeds.
**Cutting Forces**

Cutting Forces/torque acting during various machining operations are shown in the fig.1 and formulae for calculating the same are given in the table below.

### Table 1.1: Formulae for calculating cutting forces

<table>
<thead>
<tr>
<th>Machining operation (no. as per view in Fig1.1)</th>
<th>Formula for cutting force/torque</th>
<th>c</th>
<th>X</th>
<th>y</th>
<th>k</th>
<th>Remark</th>
</tr>
</thead>
<tbody>
<tr>
<td>a) Planing on planner or shaper</td>
<td>$P_z$</td>
<td>2000</td>
<td>1</td>
<td>0.75</td>
<td></td>
<td>P in newton</td>
</tr>
<tr>
<td>b) Turning operation</td>
<td>$\frac{P_y}{P_x} = ct^x s^y$</td>
<td>1250</td>
<td>0.9</td>
<td>0.75</td>
<td></td>
<td>S in mm/rev</td>
</tr>
<tr>
<td></td>
<td></td>
<td>650</td>
<td>1.2</td>
<td>0.65</td>
<td></td>
<td>t in mm (feed)</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>t in mm (depth of cut)</td>
</tr>
<tr>
<td>c) Facing or cutting operation on lathe etc</td>
<td>$P_z = cB^x s^y$</td>
<td>2000</td>
<td>1</td>
<td>0.75</td>
<td></td>
<td>B is cutting width in mm</td>
</tr>
<tr>
<td>d) External broaching</td>
<td>$P_z = cs^y b_z$</td>
<td>2100</td>
<td></td>
<td>0.83</td>
<td></td>
<td>Z = No of teeth on broaching in contact with job</td>
</tr>
<tr>
<td></td>
<td>$P_y = (0.6-0.8) P_z$</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>$\beta$ - Helix angle of teeth, b-width of broach in mm</td>
</tr>
<tr>
<td></td>
<td>$P_x = 0.3P \cdot \text{ctg} \beta$</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>During deep broaching/drilling without groove for chip removal force may be 2-3 times more</td>
</tr>
<tr>
<td>e) Internal broaching</td>
<td>$P = c d s^y z$</td>
<td>2000 to 3000</td>
<td>0.85</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>f) Drilling m/c</td>
<td>$M = c d^x s^y$</td>
<td>350</td>
<td>1.9</td>
<td>0.8</td>
<td>0.7</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td>850</td>
<td>1.0</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>g) Reaming</td>
<td>$M = c d^x s^y t^k$</td>
<td>120</td>
<td>1.0</td>
<td>0.75</td>
<td>0.75</td>
<td>0.95</td>
</tr>
<tr>
<td></td>
<td></td>
<td>48</td>
<td>0.75</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>h) Cylindrical milling (direction of cutter motion reverse of feed motion)</td>
<td>$P_o = s B z S^y z^t \left(\frac{t}{D}\right)^k$</td>
<td>682</td>
<td></td>
<td>0.72</td>
<td>0.86</td>
<td></td>
</tr>
<tr>
<td></td>
<td>$P_z = (0.5-0.6) P_o$</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>Z = no of teeth</td>
</tr>
<tr>
<td></td>
<td>$P_s = (1-1.2) P_o$</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>Sz = mm/tooth</td>
</tr>
<tr>
<td></td>
<td>$P_n = \pm 0.2 P_o$</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>D = Diameter in mm</td>
</tr>
<tr>
<td></td>
<td>$P_x = 0.3 P_o \cdot \text{tan} \beta$</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>B = width in mm</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>$\beta$ = Helix angle of milling cutter</td>
</tr>
<tr>
<td>i) Cylindrical milling (direction of feed motion &amp; cutter speed motion are same)</td>
<td>$P_o = c B z S^y z^t \left(\frac{t}{D}\right)^k$</td>
<td>600 to 650</td>
<td>0.72</td>
<td>0.86</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>$P_z = (0.5-0.6) P_o$</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>P in newton</td>
</tr>
<tr>
<td></td>
<td>$P_s = (0.8-0.9) P_o$</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>S in mm/rev</td>
</tr>
<tr>
<td></td>
<td>$P_n = (0.3-0.4) P_o$</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>t in mm (feed)</td>
</tr>
<tr>
<td></td>
<td>Formula</td>
<td>Po</td>
<td>k1</td>
<td>k2</td>
<td>k3</td>
<td></td>
</tr>
<tr>
<td>----------------</td>
<td>--------------------------------------------------------------------------</td>
<td>------</td>
<td>-------</td>
<td>-------</td>
<td>-------</td>
<td></td>
</tr>
<tr>
<td>j) Surface milling</td>
<td>$P_o = c t ^x S ^y \left( \frac{B}{D} \right)^k$</td>
<td>824</td>
<td>0.95</td>
<td>0.8</td>
<td>1.1</td>
<td></td>
</tr>
<tr>
<td>Teeth milling</td>
<td>$P_o = c t ^x S ^y m ^{k_1} v ^{k_2}$</td>
<td>420</td>
<td>0.15</td>
<td>0.7</td>
<td>K_1=1.1 K_2=-0.1</td>
<td></td>
</tr>
<tr>
<td>k) Cylindrical grinder</td>
<td>$P = c t ^x S ^y v ^{k_1}$</td>
<td>22</td>
<td>0.6</td>
<td>0.7</td>
<td>0.7</td>
<td></td>
</tr>
<tr>
<td></td>
<td>$P_y = (1-3)P_z$</td>
<td></td>
<td></td>
<td></td>
<td>S(mm/rev)</td>
<td></td>
</tr>
<tr>
<td></td>
<td>$t$(mm/double stroke)</td>
<td></td>
<td></td>
<td></td>
<td>V(m/min.)</td>
<td></td>
</tr>
</tbody>
</table>
Fig 1.1 - Cutting forces and other parameters of machine tools during various machining operations
1.3. DETERMINATION OF POWER REQUIREMENT OF MAIN DRIVE

It is often very difficult to determine the power requirement of the main drive of a new machine tool. This is due to lack of sufficient data such as:
1) The laws governing the cutting and feed forces in various chip removals processes.
2) Operating conditions.
3) Friction losses in the drive.

Useful power is calculated for the maximum cutting speeds and feeds.

Power rating of electric motor depends upon the power required to overcome each component of the cutting force.

\[
P_z = K(a + 0.4c)b \quad \text{kgf}
\]

\[
P_N = \sqrt{P_x^2 + P_y^2} = K_b(0.4a + c)
\]

where
- \(P_z\) - tangential cutting force in kg at the cutting point
- \(P_x\) - Force component in longitudinal feed direction
- \(P_y\) - " " " " cross feed direction
- \(a\) - Thickness of undeformed chip
- \(b\) - Width " " " "
- \(c\) - Mean width of flank wear land equal to half of the permissible flank wear

Cutting power

\[
N_c = \frac{P_zV}{60 \times 75 \times 1.36} = \frac{P_zV}{60 \times 102} \quad \text{kW}
\]

where \(V\) - cutting speed in meter/min

Tangential cutting force can be calculated by the following formula also.

Cutting force \(P_z = K.A\)

Where \(A\) - area of cross-section of chip in \(\text{mm}^2 = \text{s.t.}\)

where \(s\) is width of cut in feed per revolution and \(t\) is depth of cut in mm

\(K\) - cutting coefficient tabulated below for various types of material

<table>
<thead>
<tr>
<th>Cutting Material</th>
<th>Tensile Strength Kg/mm²</th>
<th>Cutting Coeff. K</th>
</tr>
</thead>
<tbody>
<tr>
<td>Steel</td>
<td>40-50</td>
<td>150</td>
</tr>
<tr>
<td></td>
<td>50-60</td>
<td>160</td>
</tr>
<tr>
<td></td>
<td>60-70</td>
<td>178</td>
</tr>
<tr>
<td></td>
<td>70-80</td>
<td>200</td>
</tr>
<tr>
<td></td>
<td>80-90</td>
<td>220</td>
</tr>
<tr>
<td></td>
<td>90-100</td>
<td>235</td>
</tr>
<tr>
<td></td>
<td>100-110</td>
<td>155</td>
</tr>
<tr>
<td>Cast Iron</td>
<td>140-160</td>
<td>100</td>
</tr>
<tr>
<td></td>
<td>160-180</td>
<td>108</td>
</tr>
<tr>
<td></td>
<td>180-200</td>
<td>114</td>
</tr>
<tr>
<td></td>
<td>200-220</td>
<td>120</td>
</tr>
<tr>
<td>Metal</td>
<td>-</td>
<td>35-55</td>
</tr>
<tr>
<td>-----------</td>
<td>------</td>
<td>--------</td>
</tr>
<tr>
<td></td>
<td></td>
<td>95-115</td>
</tr>
<tr>
<td>Aluminium</td>
<td>-</td>
<td>40</td>
</tr>
<tr>
<td></td>
<td></td>
<td>110</td>
</tr>
</tbody>
</table>

**Turning**

Feed per minute $S_m = S \times n$

Where $S_m$ is feed per minute & $S$ is feed per revolution

$n$ is speed in revolution per minute

Cutting speed $V$ in m/min $= \frac{\pi D n}{1000}$

Where $D$ is diameter of work piece in mm

Metal removal rate in $\frac{cm^3}{min}$, $Q = s.t.v$

Where $s$ is feed per revolution and $t$ is depth of Cut

Average chip thickness $a = s \sin x$

Where $x$ is approach angle

Power required at spindle $N$ in kW $= U.Q.K_n.K_r$

Where $U$ - unit power in $\frac{kW}{cm^3/min}$

$K_n$ - correction factor for flank wear

$= 1.2$ in normal case

$= 3$ in worst case when flanks wear is 0.8 mm and material is hardest

$K_r$ - correction factor for rake angle

$= 0.87$ for $+20^\circ$ rake angle

$= 1$ for $+10^\circ$ rake angle

$= 1.35$ for $-15^\circ$ rake angle

Torque at spindle $T_s = \frac{975N}{n}$ in Kg.M

Tangential cutting force $P_z = \frac{6120N}{v}$
Milling

Feed per minute \( S_m = S_z Z n \)

Where \( S_z \) - feed per tooth of cutting in mm

\( Z \) - no. of teeth in cutter

Cutting speed \( V = \frac{\pi D n}{1000} \) in m/min

Where \( D \) - dia of cutter in m.m.

Metal removal rate \( Q \) in \( \frac{cm^3}{min} = \frac{b t S_m}{1000} \)

Where \( b \) is width of cut

\( t \) is depth of cut

Power required at spindle \( 'N' = U Q \) kW

Where \( U \) - unit power in \( \frac{kW}{cm^3/min} \)

Drilling

Power required at spindle \( N = 1.25 D^2 K n \ (0.056+1.5S)10^{-5} \)

Where \( D \) is drill diameter in mm

\( N \) is rpm of spindle

\( S \) is feed per revolution

\( K \) is material factor = 1 for cast iron

= 1.5 for carbon steel

= 2.3 for alloy steel of 241 BHN

For different type of material, Unit power \( U \) in kW per cubic centimetre per minute is tabulated below for turning and machining operations:

<table>
<thead>
<tr>
<th>Material</th>
<th>Chip thickness 0.025 mm</th>
<th>Chip thickness 0.8mm</th>
</tr>
</thead>
<tbody>
<tr>
<td>Mild steel</td>
<td>66 x 10^{-3}</td>
<td>30 x 10^{-3}</td>
</tr>
<tr>
<td>Med. carbon steel</td>
<td>69 x 10^{-3}</td>
<td>30 x 10^{-3}</td>
</tr>
<tr>
<td>Alloy steel of 110 kg/mm² of</td>
<td>85 x 10^{-3}</td>
<td>36 x 10^{-3}</td>
</tr>
<tr>
<td>Tensile strength</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Stainless steel</td>
<td>110 x 10^{-5}</td>
<td>60 x 10^{-5}</td>
</tr>
<tr>
<td>Cast iron upto 200 BHN hardness</td>
<td>38 x 10^{-3}</td>
<td>18 x 10^{-3}</td>
</tr>
</tbody>
</table>
**Grinding**

Power at spindle N= U.Q.

*For surface grinding:*

\[
\text{Metal removal rate } Q = \frac{b \cdot t \cdot f_t}{1000} \text{ cm}^3 / \text{min}
\]

Where \( b \) is width of grinding in one pass i.e. mm per sec.
\( t \) is depth of grinding
\( t_f \) is table traverse feed rate in mm per min

Tangential cutting force \( P_z = \frac{6120 \cdot N}{V} \) Newton

Where \( V \) is cutting speed in meter per min.

*For cylindrical grinding:*

\[
\text{Metal removal rate } Q = \frac{\pi D_w t f}{1000} \text{ cm}^3 / \text{min}
\]

Where \( D_w \) is diameter of work piece
\( t \) is depth of cut
\( f \) is table traverse feed rate

**Table 1.4: Average unit power 'U' for grinder**

<table>
<thead>
<tr>
<th>Work material</th>
<th>Depth of grinding in mm per pass or in feed in mm/revolution</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>0,01</td>
</tr>
<tr>
<td>Mild steel med.</td>
<td>1.4</td>
</tr>
<tr>
<td>Carbon steel</td>
<td></td>
</tr>
<tr>
<td>Alloy steel</td>
<td>1.3</td>
</tr>
<tr>
<td>Cast iron</td>
<td>1.15</td>
</tr>
<tr>
<td>Aluminium alloy</td>
<td>0.58</td>
</tr>
</tbody>
</table>
Table 1.5: Cutting conditions:

**a) Turning**

<table>
<thead>
<tr>
<th>Work material</th>
<th>Tool material</th>
<th>Cutting speed m/min</th>
<th>Depth in mm / feed in mm per revolution</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td></td>
<td>(5-10)/(0,4-0,6)</td>
</tr>
<tr>
<td>Free machining steel</td>
<td>HSS</td>
<td>20 - 40</td>
<td>40 - 70</td>
</tr>
<tr>
<td></td>
<td>Carbide</td>
<td>90 - 150</td>
<td>120 - 180</td>
</tr>
<tr>
<td>Mild steel</td>
<td>HSS</td>
<td>25 - 35</td>
<td>30 - 50</td>
</tr>
<tr>
<td></td>
<td>Carbide</td>
<td>60 - 120</td>
<td>80 - 150</td>
</tr>
<tr>
<td>Med. carbon steel</td>
<td>HSS</td>
<td>15 - 25</td>
<td>25 - 45</td>
</tr>
<tr>
<td></td>
<td>Carbide</td>
<td>50 - 110</td>
<td>60 - 120</td>
</tr>
<tr>
<td>Alloy Steel</td>
<td>HSS</td>
<td>10 - 15</td>
<td>15 - 25</td>
</tr>
<tr>
<td></td>
<td>Carbide</td>
<td>30 - 65</td>
<td>40 - 80</td>
</tr>
<tr>
<td>Aluminium Alloy</td>
<td>HSS</td>
<td>40 - 70</td>
<td>70 - 100</td>
</tr>
<tr>
<td></td>
<td>Carbide</td>
<td>60 - 150</td>
<td>80 - 180</td>
</tr>
<tr>
<td>Cast iron</td>
<td>HSS</td>
<td>20 - 25</td>
<td>25 - 30</td>
</tr>
<tr>
<td></td>
<td>Carbide</td>
<td>60 - 90</td>
<td>70 - 100</td>
</tr>
</tbody>
</table>

**b) Milling:**

<table>
<thead>
<tr>
<th>Work material</th>
<th>Tool material</th>
<th>Speed M/min.</th>
<th>Feed in mm per tooth</th>
<th>Form Cutter</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>Face Mill.</td>
<td>Slab Mill.</td>
<td>Side Mill.</td>
</tr>
<tr>
<td>Cast iron</td>
<td>HSS</td>
<td>20-30</td>
<td>0,35</td>
<td>0,3</td>
</tr>
<tr>
<td></td>
<td>Carbide</td>
<td>70-100</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Mild steel</td>
<td>HSS</td>
<td>25-40</td>
<td>0,25</td>
<td>0,2</td>
</tr>
<tr>
<td></td>
<td>Carbide</td>
<td>90-130</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Carbon steel</td>
<td>HSS</td>
<td>25-30</td>
<td>0,2</td>
<td>0,15</td>
</tr>
<tr>
<td></td>
<td>Carbide</td>
<td>60-90</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Alloy Steel</td>
<td>HSS</td>
<td>10-20</td>
<td>0,15</td>
<td>0,1</td>
</tr>
<tr>
<td></td>
<td>Carbide</td>
<td>40-55</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Aluminium Alloy</td>
<td>HSS</td>
<td>60-100</td>
<td>0,5</td>
<td>0,4</td>
</tr>
<tr>
<td></td>
<td>Carbide</td>
<td>60-1880</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>
Where primary cutting motion is of reciprocating type, the cutting speed is determined by:

\[ V = \frac{L}{1000T_c} \] meter per minutes

Where, \( L \) is the length of stroke, mm
and \( T_c \) is time of cutting stroke, minutes.

If \( T \) is time of idle stroke which is normally less than \( T_c \) then

\[
\text{Number of stroke per minutes } n = \frac{1}{T_c + T_i} = \frac{1}{T_c \left(1 + \frac{T_i}{T_c}\right)}
\]

Or, \( T_c = \frac{1}{n \left(1 + \frac{T_i}{T_c}\right)} \)

Or, \( V = \frac{L.n \left(1 + \frac{T_i}{T_c}\right)}{1000} \)

Feed per minutes \( S_m = S \times n \) Where \( S= \) feed per stroke

1.4. CALCULATIONS FOR FEED DRIVE

The power requirement of a feed drive can be calculated from the feed force \( F \) and feed rate: taking into account the efficiency, which is of the order of 0.15-0.2 for the conventional system. Feed force is the total sum of cutting force and friction force due to reaction \( R_i \) of weight of saddle etc on the guideways.

1. For the saddles of lathes with vee or combined ways

\[ F = kP_x + \mu(P_z + G) \]

Where k-Factor considering the supporting torque=1.15
Px-cutting force component along feed guide in Newton (axial force)
Pz-tangential cutting force (vertical component in case of lathe m/c)
G-wt. Of the saddle or moving part
\( \mu \)-friction coefficient =0.15-0.18

2. For lathe and milling saddles with right angle flat guide ways.

\[ F = kP_x + \mu(P_z + P_y + G) \]

Where, Py-perpendicular horizontal component of the cutting force on the guide
\( k=1.1 \) and \( \mu=0.15 \)

3. For the tables of milling m/c with dovetail ways.

\[ F = kP_x + \mu(P_z + 2P_y + G) \]

Where, \( k=1.1 \) and \( \mu=0.2 \)
4. For the spindle of drilling m/c

\[ F = P_x + \mu' \frac{2Mt}{d} \]

Where, \( \mu' = 0.15 \) for the quill of drilling m/c
\( d = \) diameter of spindle
\( Mt = \) torque on the spindle
\( K = \) Factor considering the supporting torque
or influence of overturning moment

**Calculating Torque Requirement**

Rotating a lead screw through one revolution produces an axial force \( F \) to move a load \( G \) through a distance equal to thread pitch `h'.

Torque \( M = \frac{F.h}{2\pi\eta} \)

Where, \( \eta \) - efficiency of the system

Power requirement \( N = 2\pi n M \)
Or, \( N = 0.1047 \times M \times n \times 10^{-3} \)
Where, \( N \) - Power in kW
\( M \) - torque in newton meter
\( n \) - rpm

If \( V_r \) is the fastest speed (rapid) per minute, maximum speed of the lead screw = \( V_r / h \) revolution per minute

In case rack and pinion of radius \( r_p \) is used in place of lead screw, \( h \) can be substituted by \( 2\pi r_p \) i.e one revolution of pinion will move the saddle by \( 2\pi r_p \) distance.
In position control circuit drive run up time or response time is also important. The approximate run up time from speed zero to fast feed is calculated. The run up time should not exceed 100 milliseconds in position control circuit. If the calculations show run up time more than 100 milliseconds a different gear design must be chosen in order to reduce the external moment of Inertia or, if necessary a large motor has to be used. Run up time \( t_H \) could be calculated by the following formula.

\[
\begin{align*}
t_H &= \frac{\sum J \times \Delta n}{9.56 \times M_b}
\end{align*}
\]

Where, \( t_H \) = run up time in seconds  
\( \sum J \) = total moment of inertia in kgm\(^2\)  
\( \Delta n \) = speed difference in rpm  
\( M_b \) = accelerating torque in newton meter

\[
M_b = M - M_R
\]

Where \( M_R \) = Frictional torque in Newton meter

\[
F_r = \frac{r_F \cdot h}{2\pi \eta}
\]

Where, \( F_r \) is the friction force due to linear moving mass.

If motor is fitted directly with the lead screw

Total moment of inertia \( \sum J = J_{motor} + J_s + J_v \)

Where, \( J_s \) = moment of inertia of lead screw  
\( J_v \) = M.I. of linear moving mass (saddle) referred to lead screw

\[
J_v = G \times \left( \frac{h}{2\pi} \right)^2 \text{ Kg m}^2
\]

\[
J_v = G G x r^2 \text{ Kg m}^2 \text{ for rack pinion drive}
\]

Where \( h \) is lead per revolution of lead screw in meter \( G \) is weight of linear moving mass and \( r \) is pitch radius of pinion.

If gearbox is used between lead screw and motor, moment of inertia for each shaft has to be calculated and to be divided by (gear ratio)\(^2\) each time before adding it to the moment of inertia of the motor. Torque on motor will be the torque on lead screw divided by the gear ratio \( i \).

\[
J_m = \frac{J_s}{i^2} \text{ and } M_{mot} = \frac{M}{i}
\]

Where \( J_s \) is the moment of inertia of lead screw and \( J_m \) is the moment of inertia of lead screw referred to motor shaft.
Driving torque of Feeding screw system

Driving torque of feeding screw system $T_s$ is given by the following equation

$$T_s = T_G + T_P + T_D + T_F$$

where $T_G = \text{Acceleration-torque} = J.\alpha$

$J$ is moment of inertia

$\alpha$-(angular acceleration)$= \frac{2\pi n}{60\Delta t}$ rad/sec$^2$

$n$ is rpm and $\Delta t$ is time of acceleration

Total Moment of inertia ($J$) of all the parts are added by calculating M.I. converted to motor shaft for each component

$$J = J_{motor} + J_{coupling} + J_{load}$$

M.I. of cylindrical load $J = \frac{\pi \rho L D^4}{32}.10^2 \text{kgm}^2$

$=0.77x D^4x Lx 10^{-12} \text{Kgm}^2$ for steel rotor (ie lead screw, gear, pulley etc)

M.I. of object at linear motion$= \frac{W}{4.\left(\frac{P}{\pi}\right)^2}.10^2 \text{kgm}^2$

(converted to motor shaft)

Where, $\rho$=weight per unit volume (kgf/m$^3$)

$L$= length of cylinder (m)

$D$= diameter of cylinder (m)

$W$= weight of linear motion parts (kgf)

$P$= travel of linear motion parts per revolution of motor(m)

Working load torque $T_P = M = \frac{F_h}{2\pi \eta}$

Drag torque $T_D = \frac{mP_{pl}l}{\sqrt{\tan \alpha}.2\pi}$ or $\frac{kP_{pl}l}{2\pi}$

Where, $m$ = coefficient of internal efficiency

(normal 0.05)

$P_{pl}$ = preload (kgf)

$l$= nominal lead(cm)

$\alpha$= lead angle

$k= 0.1-0.3$

Total Friction torque $T_F$= torque req. to overcome friction against supporting bearings, oil seal etc.
The sum of $T_D$ and $T_P$ is usually limited to 10% to 30% of the motor output, depending on the type of the motor.

In addition to acceleration torque, the motor must be able to provide sufficient over the entire duty cycle. This includes certain amount of constant torque during run phase and deceleration torque during the stopping phase.

$$T_{\text{dec}} = -J\alpha + T_F$$

Root mean square value (rms) of torque required over the entire duty cycle can be calculated by

$$T_{\text{rms}} = \sqrt{\frac{T^2_{\text{acc}}(t_{\text{acc}}) + T^2_{\text{run}}(t_{\text{run}}) + T^2_{\text{dec}}(t_{\text{dec}})}{t_{\text{acc}} + t_{\text{run}} + t_{\text{dec}}}}$$

Where $t$ is the time taken

**Direct Drive**

Because there are no mechanical linkages involved the load parameters are directly transmitted to the motor. The speeds of the motor are the same as that of the load. Therefore, the total inertia is the load inertia plus the motor inertia.

$$J_t = J_l + J_m$$

**Gear Drive**

The mechanical linkages between the load and motor in a gear drive, as with any speed changing system, the load inertia reflected back to the motor is squared function of the speed ratio. Motor speed:

$$S_m = S_l \times N_l$$  \hspace{1cm} (6)$$

$$S_m = \frac{S_l \times N_l}{N_m}$$  \hspace{1cm} (7)$$

Motor torque:

$$T_m = \frac{T_l}{N_e}$$  \hspace{1cm} (8)$$

Reflected load inertia:

$$J_r = \frac{J_l}{N^2}$$  \hspace{1cm} (9)$$

Total inertia motor:

$$J_t = J_r + J_m$$  \hspace{1cm} (10)$$
Tangential Drive:
Consisting of a timing belt and pulley, chain and sprocket or rock and pinion, a tangential drive, also requires reflecting load parameters back to the motor shaft,

Motor speed:
\[ S_m = \frac{V_i}{2\pi R} \]  

Load torque:
\[ T_l = F_l R \]  

Friction torque:
\[ T_f = F_f R \]  

Load inertia:
\[ J_l = \frac{W_{lb} R^2}{g} \]  

Total inertia:
\[ J_t = \frac{W_{lb} R^2}{g} + J_{p1} + J_{p2} + J_m \]

**1.5 POWER RATING OF ELECTRIC MOTOR**

The main spindle drive must provide the cutting force needed to machine the components. The amount of drive power available must be variable, at widely varying speed depending upon the type of material to be machined, type of cutting tool and the type of machining.

Power rating of electric motor can be determined by dividing the cutting power calculated above with the efficiency factor:
\[ Nm = \frac{N}{\eta} \text{ kW} \]

where \( Nm \) - the power required at motor
\( n \) - Coefficient efficiencies of the drive
\[ = \eta_1 \eta_2 \eta_3 \ldots \ldots \]

Where \( \eta_1, \eta_2, \eta_3 \ldots \ldots \) are the coefficients of efficiencies for individual drive.
Overall efficiency lies generally between 0.8 - 0.85 for machine tools with rotary primary motion and 0.6 - 0.7 for machine tools with reciprocating primary motion:

<table>
<thead>
<tr>
<th>Type of transmission</th>
<th>Efficiencies</th>
</tr>
</thead>
<tbody>
<tr>
<td>Belt drive (flat)</td>
<td>0.98</td>
</tr>
<tr>
<td>Belt drive (V belt)</td>
<td>0.96</td>
</tr>
<tr>
<td>Spur gear drive</td>
<td>0.98</td>
</tr>
<tr>
<td>Helical gear drive</td>
<td>0.97</td>
</tr>
<tr>
<td>Bevel gear drive</td>
<td>0.96</td>
</tr>
<tr>
<td>Crank slider mechanism</td>
<td>0.90</td>
</tr>
<tr>
<td>Jaw clutch</td>
<td>0.95</td>
</tr>
<tr>
<td>Friction clutch Operating in oil</td>
<td>0.90</td>
</tr>
</tbody>
</table>

Thus by taking efficiencies of various transmissions into consideration, power rating of the electric motor required for the machine-tools could be determined. Knowledge of power rating of motor is of great help for the designers during designing the machine, for the technologists during writing technology for the operation and for the maintenance engineers for proper care and maintenance and technological upgrades.
Chapter 2: DRIVES

2.1 SELECTION OF DRIVE

There are three main parameters to be considered while selecting a drive: type of drive controller, load and speed range. Torque requirement is the most important step in the selection of drive for any application.

The most common industrial drive is the AC induction motor. Normally induction motor run at fixed speed, in a fixed direction and draw high starting torque. Self- excited synchronous motor is also used where rotor is required to rotate at the same speed in synchronisation as the stator field. So in conventional machines, with these types of motors, a gearbox is required for reduction as well as changing to the desired speed range.

For infinitely variable speed additional mechanisms have to be incorporated such as PIV etc. for mechanical system or special electrical/electronic control to vary speed of the motor itself. Both DC and AC motors of specific design could be used depending upon the application for this purpose.

While selecting the motor following factors should also be considered:

- Permissible over loading
- Variable loading cycle ratio = \( \frac{\text{Cutting time}}{\text{Cutting time} + \text{idle time}} \times 100 \)

  For normal motor it is less than 60
- \( \frac{\text{time(switch on)}}{\text{time(switch on)} + \text{time(switch off)}} \times 100 \)

  Standard motors are available with this ratio as 15, 25, 40 and 60
- Must be capable of rotating in both directions as well as braking electrically.
- The higher the accuracy required at the work piece, the higher must be the pulse count of the controller used.
- Stall torque i.e. permitted continuous torque in the feed range
- Maximum operating speed for rapid traverse
  - Speed-torque diagram, showing the ranges of speed and torque in which a motor can be operated
    Motors for main spindle drives need to deliver constant power output over a wide speed range. The governing factor for the motor size is the torque. Therefore any reduction in the base speed with the required power output remaining the same would necessitate a larger motor. The short time current overload capacity must be up to two times the rated current of the motor.

  Requirement of spindle drives and feed drives are different. Feed drives are usually constant torque drives where as spindle drives are constant power drives and require much higher speeds than the feed drives. Some of the general characteristics important for a machine tool drives are given below.
i) High dynamic response. Toque requirement during cutting operations continuously varies depending upon the hardness and characteristics of the material. As soon as hard spot comes, cutting speed will drop down unless it is compensated by the corresponding acceleration in a short time. So the motor must develop very high torque with in a very short time. So
   a) Rate of rise of current capability (ds/dt) should be very high.
   b) Inertia of the motor should be low.
   c) Construction of the motor and lamination's assembly should be such that it does not get twisted at high torque.

ii) High degree of dynamic balancing

iii) Armature reaction should be properly compensated by the use of compensation winding. In case of motors for machine tool applications, it is necessary to use compensation windings with in the shoes of the main field to homogenise it. This will eliminate the chances of shifting of neutral zone and increase the stability of the motor.

2.2 VARIABLE SPEED MOTOR AND DRIVE MECHANISM

Speed regulation is possible in both AC and DC motors. To control the motion, drives, controllers and feedback devices are required depending upon their types and characteristics.

**Fig.2.1 Arrangement for motion control**

**Drives:** Amplifiers are necessary in many applications to transmit power to either the motors or actuators. Not all systems will need a power amplifier (e.g. an induction motor can utilise the power supplied directly by commercially supplied mains), however almost all motors need or can accept power from an amplifier for accurate control.
**Motion Controllers:** Most often motion controllers are used in positioning applications. Motion controllers can provide either an analog or digital signal to an amplifier in order to increase control over the system. Motion controllers often involve some sort of feedback from the actuator. The main purpose of the motion controller is to create a position reference signal. Most systems require position feedback in order to accomplish this task. Many motion controllers are loaded with other features to accomplish a variety of other tasks simultaneously. Some such features consist of digital and analog I/O control, operator interfaces, and communications support. A diagram of a basic motion controller is shown below.

![Diagram of a basic motion controller](image.png)

_{Fig 2.2 Basic motion controller_}

The profile generator is used to establish the desired position signal. The desired position is derived from information fed to the communications via some interface such as a host computer. This information may involve a series of precise and complicated motion commands, depending on the flexibility of the controller. The position converter is used to convert the signal sent by the position feedback device to an actual position signal. Now we have an actual and desired position signal. These two are compared, and an error signal is produced that will be used to determine the position signal output.

**Sensors/Feedback Devices:** Many systems require an action upon the occurrence of another action. The primary action can be detected by a sensor. There are a huge variety of sensors depending on what you need to detect (objects, pressure, light, colour, width, dimension, component integrity, sound, etc). Some situations may require continuous updating of a certain event; a feedback device may be the proper solution.
2.3 D.C. MOTOR AND DRIVES

The D.C. Motor contains two main parts, the field winding and the armature. The voltage applied to the field winding produces a magnetic field and this field interact with that produced by the current carrying armature circuit. The interaction develops torque, producing the rotation of the armature. The stator-windings are usually supplied by an alternate single phase DC source, leaving the DC drive in charge of the windings of the rotor. Since the DC brushed motor uses brushes to automatically commutate itself, this drive does not need complicated circuitry to electronically commutate the motor as is the case with the brushless motor. The torque $T$ of the motor is proportional to the strength of the field flux $\phi$ and the armature current $I_a$. The speed of the DC Motor $n$ is dependent on armature voltage $V$ and field flux.

$$n = k \frac{V - I_a R_a}{\phi} \quad \text{and Torque} \quad T \propto \phi I_a$$

Where $K$ is a constant and
- $R_a$ is the armature circuit resistance.
- $\phi$ is the field flux.

Thus the speed of the motor can be controlled either by varying the voltage applied to the armature or by varying the field flux when the rated armature voltage and the rated field voltage are both applied. The speed of motor rotating at its rated value is called base speed. By varying the armature applied voltage; the speed of the motor could be controlled from zero to full speed (base speed) infinitely at constant torque.

The motor provides full torque even at fractions of a RPM as neither field nor armature current is varying. In case of field control, the variation of speed results but the torque also varies. So it is normally termed as constant power arrangement. Thus by field control, speed variation is possible only at above rated speed up to the range of 1:3 & above base speed by reducing the field voltage. A combination of both the arrangements is generally provided in main drive systems of machine tools.

For changing speed of rotor, voltage applied to the armature could be varied either by putting variable resistance in series with the armature winding or by changing value of input voltage. Former method is not economical because the armature current passes through the variable resistance resulting in large heat losses. For changing value of input voltage initially Ward-Leonard systems were being used. With the advancement in electronics, the thyristered drives are preferred now a day.
WARD LEONARD SYSTEM

This is also known as the generator-motor system applied for changing the speed of D.C. motor (Fig 2.3). It consists of AC motor 1 coupled with a DC generator 2. The D.C voltage generated is applied to the armature of the main device D.C motor 3. By varying the excitation of the DC generator field 5, voltage applied to the motor is varied. The field execution of the generator can be controlled by another self-excited D.C generator or by amplidyne 6 or by means of SCR circuit. The speed of main drive DC motor can be increased above the rated speed by controlling its field winding 4 by the similar method as mentioned for the field winding of the DC generator. The disadvantage of the system is that several machines are required and so response is not very good.

Fig 2.3 Ward Leonard Control
THYRISTOR SYSTEM

A rectifier is a device that converts an AC signal to a positive varying signal. Rectifiers utilize diodes to permit flow in only one direction. If a signal such as this were used to drive a DC motor, there would be no variable control over motor torque or speed. The controlled rectifier drive resolves this problem by replacing the diodes with thyristors. These solid-state devices can be used to regulate what percentage of the sine wave signal is permitted through the device. The thyristor does not allow any current to flow through it, until a small voltage is placed on the gate; this is known as firing the circuit. By regulating the time period at which the thyristor is fired, one can regulate the mean voltage.

Thyristor power supply system provides a variable DC Supply from Single Phase or Three Phase AC input by controlled rectification. It is also called Silicon Controlled Rectifier (SCR) Converter system. The Thyristor is a three terminal device with an Anode, a Cathode and a Gate. The conduction of current through the device cannot take place until a positive firing pulse of appropriate voltage and pulse width is applied to the gate with respect to the cathode. At this time the SCR fires and will continue to conduct until the anode voltage is removed or reversed in polarity. The gate cannot control the SCR once it is fired. A very small gate power, of the order of milliwatts, is sufficient to control kilowatts of power through this device. Firing angle is measured in electrical degrees from the zero cross over of the AC voltage to the point at which firing pulse appears in the single-phase system. In three-phase system, firing angle is measured from the intersection of the two adjacent phase voltage waveforms. It has been observed that by advancing the firing angle, the average DC voltage at the load terminal goes on reducing. In the wave form where anode becomes more negative than cathode, firing pulses do not fire the SCR. Usually three types of SCR power supplies are used. These are single-phase half wave, single-phase full wave and three phase half wave types.

Fig 2.4 Single Phase, half wave, bi-directional drive

i) Single Phase Half Wave System:-

![Fig 2.4 Single Phase, half wave, bi-directional drive](image-url)
The system consists of two inverse parallel-connected SCR’s A11 and B11. Forward motion is achieved by conducting through A11 while reverse direction of rotation is controlled by the conduction of B11.

In this system voltage contains lot of harmonics causing unnecessary heating of the motor. The form factor of the voltage wave is very poor and the DC motor would have to be derated to nearly 50% of its rated value. Poor form factor causes torque pulsation at low speeds, limiting the speed range of the system. So such system has its application only in fractional horsepower motors.

ii) Single Phase Full Wave System:

For correcting positive error, SCR’s A1Z and A2Z conduct to supply positive voltage at the terminals of the motor for its forward motion for negative errors, SCR’s B1Z and B2Z conduct to rotate the motor in reverse direction. The response of the drive is better than the half wave system. The form factor is also better and the motor would be rated to approximate 75% of its rated value. Single-phase full wave finds application in medium sized machine tools.

Fig 2.5 Single Phase, full wave, bi-directional drive
iii) Three Phase Half-Wave System:-

The system contains three pairs of inverse parallel connected SCR's in each phase of the three-phase system with neutral as the return lead.

Thus 3 voltage pulses are applied to the motor per cycle of the input voltage so the form factor is much better and the motor derating can be typically 10%. System response is fast, thus it is possible to provide corrective power to the motor 150 times per second.

iv) Three Phase Full Wave Rectifier:-

Full wave rectifier is far superior to half wave rectifier; hence 6 pulse drives are the best and give very high form factor approaching unity.

This means the ripples in the output current will be reduced to zero and chances of unnecessary motor heating will be minimised.

SCR drives are available for single quadrant as well as for four-quadrant operation. With single quadrant drives, the motor can drive only in one direction and cannot brake. Four quadrant drives are able to drive and brake in both the direction of rotation. The performance of SCR drive system can be improved by using inductors as it smooths out the current flowing into the motor. Thus the form factor is improved which helps in extending the speed range of the drive because the motor can operate more uniformly at low speeds without torque pulsation. The frequency response of the system is improved as inductors act as a storage power; also the inductors protect the positive and negative bridges being short-circuited.
Working Of SCR-DC Servomotor System:

On the DC servomotor the compensating winding, the smaller diameter of armature and lamination of the yoke ring may be clearly seen. Good acceleration feature is obtained by exceptionally low moment of inertia of the small rotor. Slow speed servomotor is permanent magnet D.C. motor with shunt characteristics. They have high torque available at low speeds. It is always preferred that the tachogenerator used for velocity feed back is mounted on the armature shaft without any coupling.

The block diagram of a typical single-phase full wave SCR drive kits is shown in the figure. Pre amplifier amplifies the difference between the velocity command voltage from the control unit/regulator and the actual velocity of the motor as transuded by the tachogenerator feed back voltage. A ramp is generated synchronised with the AC supply for subsequent comparison with the amplified error voltage. Comparator consisting of two transistorised differential amplifier compare the ramp and the amplified error voltage. The signal is now fed to the pulse generator.

The main part of the trigger pulse generator is the blocking oscillator circuit, which produces a train of pulses for SCR triggering. Further circuitry is included to assure that only that SCR is fired whose anode is having the positive half cycle of the applied AC voltage at a particular point of time. A bias current is usually circulated in the motor via the chokes where zero velocity command is present. This helps in keeping the motor stiff and reducing the dead band. The amount of bias current circulated is usually 10-20% of the rated motor current. Bias current is adjusted to get more number of trigger pulses when the motor is stationary.

![Block diagram of typical SCR servo drive system](image)

*Fig 2.7 Block diagram of typical SCR servo drive system*
Transistor PWM DC Converter
If you have an application with a changing load and want accurate speed regulation, you should be sure to use velocity mode, which requires a velocity feedback device. The chopper drive varies the mean output voltage by switching the DC supply on and off with the use of one or more transistors. The arrangement of the transistors determines the type of chopper drive being used. The most common form of switching technique is PWM (pulse width modulation) and many drives are simply referred to as PWM drives.

![Fig.2.8 Signals in chopper drive](image)

The voltage signal supplied to the motor of a chopper drive can be seen in red and the current signal in black. As you can see, the current ripple and thus the torque ripple is very low. By increasing the frequency of the switching, the ripple will continue to decrease. There are basically two reasons for using a chopper drive rather than a controlled rectifier drive to control your DC brushed motor: 1) If your application warrants the use of DC supply, or 2) you need precise speed control and/or positioning control. The torque ripple and speed characteristics are improved over that of a typical DC variable speed device. For positioning applications, a chopper drive works great with a motion controller.

In case of transistor pulse width modulated (PWM) amplifiers, power-switching device is transistor instead of SCR. So it is also called transistor choppers. A rectifier feeds from AC into DC bus. The transistors are used in switching mode at frequencies 1 kHz to 10 kHz pulse width modulated and DC voltage become proportional to the pulse width.

Thus modulating pulse width could vary speed that is proportional to armature voltage. Four quadrant of an amplifier is possible in such case. It means-
(a) Forward running - quadrant I
(b) Reverse running - quadrant II
(c) Forward braking - III
(d) Reverse braking - IV

![Fig 2.9 Chopper control, one quadrant operation](image)
Braking Methods

The braking of D.C. motor is carried out either by regenerative method or by dynamic method or by reverse current. Regenerative braking is accomplished by means of a shunt circuit rheostat, operation of that causes the armature speed to drop to a minimum. Here motor begins the mains. The motor is brought to a full stop by being disconnected from the mains.

In dynamic braking, armature is disconnected from the mains, while exciting current is on and closed through the Ballast resistor or rheostat. Changing over the direction of current in the armature circuit does reverse current braking.
Constant Torque Applications

Armature voltage controlled DC drives are constant torque drives. They are capable of providing rated torque at any speed between zero and the base (rated) speed of the motor. Horsepower varies in direct proportion to speed, and 100% rated horsepower is developed only at 100% rated motor speed with rated torque.

Constant Horsepower Applications

Certain applications require constant horsepower over a specified speed range. As an example, the motor could provide constant horsepower between 50% speed and 100% speed, or a 2:1 range. However, the 50% speed point coincides with the 50% horsepower point. Any constant horsepower application may be easily calculated by multiplying the desired horsepower by the ratio of the speed range over which horsepower must remain constant. If 5 HP is required over a 2:1 range, an armature controlled drive rated for 10 (5 x 2) horsepower would be required.

Another characteristic of a shunt-wound DC motor is that a reduction in field voltage to less than the design rating will result in an increase in speed for a given armature voltage. It is important to note, however, that this results in a higher armature current for a given motor load. A simple method of accomplishing this is by inserting a resistor in series with the field voltage source. This may be useful for trimming to an ideal motor speed for the application. This provides coordinated automatic armature and field voltage control for extended speed range and constant HP applications. The motor is armature voltage controlled for constant torque-variable HP operation to base speed where it is transferred to field control for constant HP-variable torque operation to motor maximum speed.

2.4 VARIABLE SPEED AC DRIVES

With the development of electronics and introduction of microprocessor based control, application of AC drives is becoming more popular now a day. Two types of AC motors are normally used; induction motor and synchronous motor. In addition, Brushless AC and DC motors are also available now.

Induction Motor

In case of induction motor, voltage applied to stator induces current in the rotor i.e. one magnetic field is set up on the stator and a second magnetic field is induced on the rotor. Induction motor consists of stator with lamination and turns of copper wire, and a rotor constructed of steel laminations, with large slots on its periphery stacked together to form a squirrel cage rotor. The slots are filled with conductive material, either copper or aluminium and are short-circuited themselves by conductive end rings.
Synchronous Motor

The main difference between the synchronous motor and the induction motor is that the rotor of this motor travels at the same speed as the rotating magnetic field. This is possible because the magnetic field of the rotor is no longer induced. The rotor either has permanent magnets or dc excited currents, which are forced to lock into a certain position when confronted with another magnetic field.

The rotor construction enables this type of motor to rotate at the same speed, in synchronisation as the stator field. There are two types of synchronous motor; self excited and directly excited. The former type has a rotor with notches, flats or teeth on the periphery. These teeth are also called salient poles. They create an easy path for the magnetic field to follow, thus allowing rotor to lock in and run at the same speed as the rotating field. During operation the rotor lags a small distance behind the stator field.

Applications where constant speed is necessary or where two or more motors need to be in synch are ideal for the synchronous motor. Besides direct commercial line power source, there are other options to obtain different varieties of control. Inverters or as AC variable speed drive can be used to perform excellent speed control, without the need for tacho-feedback as in the induction motor. By keeping the load within the load rating specs, one should never have a problem.

In the case where the load gets too high, the rotor may fall out of synchronisation. This problem has been solved through self-synchronous control. With the inverter, the frequency is supplied by the amplifier. With closed loop self-synchronous control, the location of the shaft position is relayed back to the amplifier through an encoder device. Because the amplifier knows the position of the motor, the rotor will never fall out of synchronisation. By adding the commutation device, the synchronous motor can now be called an electrically commutated motor or a brushless motor, and the drive that controls it a brushless amplifier.

Brushless Motor

When a synchronous motor is excited directly, as with permanent magnets, it is called a brushless motor. The motor is a cylinder of permanent magnet alloy. The north south poles are, in effect, the design's salient teeth and prevent the motor from slipping. The motor is driven by selectively applying power onto their windings depending on motor position feedback, such as hall sensors, encoders or resolvers.

This is a self-synchronizing device that has characteristics of both AC and DC motors. The construction of the motor is more like an AC motor than a DC, but it has a commutator, which is usually a feature of the DC motor. If it is matched with a sine wave drive, which produces a sinusoidal signal, it is considered an AC motor. If it is matched with a trapezoidal drive, which produces a trapezoidal signal, it is considered a DC motor.

This motor is basically the same as an AC self synchronous motor. Windings of the stator are placed in slots throughout the periphery similar to the AC induction motor. These windings produce a magnetic field, which creates a force on the permanent magnets of the motor.
All self-synchronizing devices need a commutator. A commutator is a device that informs the drive of the location of the rotor, so that the supply can energize the next phase to assure smooth motion. The brushless motor has a commutator; however, it is an electrical commutator rather than the traditional mechanical commutator of the brushed motor. Electrical commutation is achieved with the help of a position sensor, usually either a resolver, an incremental encoder or hall sensors. In the case of the non self-synchronous motor, a voltage is supplied to the motor windings without information of the rotor position. This can lead to the rotor falling out of synchronization without the drive or motor being aware. This is not a concern of the self-synchronized device.

By controlling the brushless motor with a sine wave drive or trap drive, excellent torque and velocity profiles can be achieved. The sine wave drive has the superior profile of the two. By adding a tachometer to regulate speed, these velocity profiles can be improved even further. The brushless motor has two advantages over the other DC motors: there is no mechanical contact which eliminates problems of wear, and because the windings are energized at the stator, heat removal is more rapid.

The brushless motor is also great for positioning applications. By either converting the resolver signal to a digital signal or adding an extra encoder, one can relay detailed position information to a motion controller. Many manufacturers sell brushless motors ready for this type of operation.

2.5 AC MOTOR-DRIVE CONTROLS

For higher speed operation AC motor is always preferred because brushes in dc drives are more prone to wear at high speed. Speed of an ac motor is determined by the following equation

\[ V = 120f/p \]

Where \( p \) is the number of poles and \( f \) is the frequency. The rotor reacts to the magnetic field, but does not travel at the same speed. The rotor speed actually lags behind the speed of the magnetic field. The term *slip* quantifies the slower speed of the rotor in comparison with the magnetic field. The rotor is not locked into any position and therefore will continue to slip through out the motion. The amount of slip increases proportionally with increases in load, thus if you need accurate velocity profiles, open loop induction motors systems are not the way to go.

It is evident from the above that motor speed can be changed by changing number of poles, slippage or AC frequency. Variation of speed by changing number of pole pairs is commonly employed method used in machine tools. It involves the application of multi-speed pole-change motor. Variation of motor speed by changing in slippage is achieved by introducing effective resistance in to the rotor circuit, which can be done only with wound rotor induction motor.
AC Induction Motor Drives

A sinusoidal voltage signal applied to the phases of an induction motor produces a rotating magnetic field in the stator. This rotating magnetic field induces a current in the winding of the rotor. The induced current then reacts with the rotating magnetic field and rotational motion is created. Traditionally, the sinusoidal voltage has been supplied by the commercial mains at 50 or 60 Hz. Unfortunately, the speed of the magnetic field is directly proportional to the frequency, so the operator has no control over speed. This problem was resolved with the creation of the inverter drive, which can supply the motor with an AC signal of varying frequency.

INVERTER CONTROL

In such systems three-phase AC power is fed to a full wave diode bridge rectifier. The output from the rectifier, smoothed by a filter provides the D.C. input to a transistor inverter bridge. The inverter transistors are switched to produce a sine wave of three-phase alternating current for driving AC motors. The switching of the inverter bridge to produce variable voltage and frequency output is controlled by an electronic regulator and waveform generator. For electronics and base drive unit, supplies are provided by a switch mode power supply (SMPS), which is also fed from the smoothened DC link, DC voltage.

Three-phase AC produced by inverter have pulse width modulated (PWM) waveform. The PWM controller converts the AC power source to a fixed DC voltage by a full-wave rectifier. The resultant DC voltage is smoothed by a filter network and applied to a pulse width modulated inverter using high power transistors. The speed reference command is directed to the microprocessor, which simultaneously optimises the carrier (chopping) frequency and inverter output frequency to maintain a proper volts/Hz ratio and high efficiency throughout the normal speed range.

![Fig.2.12 Block diagram for Frequency control](image)

The voltage applied to the motor is a pulsed approximation of a true sinusoidal waveform. This is commonly called a PWM waveform because both the carrier frequency and pulse width is changed (modulated) to change the effective voltage amplitude and frequency. The current waveform very closely follows the shape of a sine wave and therefore provides improved low speed motor performance, efficiency, and minimal motor heating. Signals from external control devices are processed by interface and control logic, and then by regulator to form an input to the waveform generator.
Basic drives for each transistor in the inverter bridge provide isolation and amplification between waveform generator logic signal levels and the power transistor. The system can also work via a microprocessor that manages a six-transistor inverter bridge by reading inputs and comparing them with selected torque/speed curves and other optional values. MPU decides if a change is required in output voltage or frequency, and duration of pulse width modulated signal accordingly. This value is sent to counter which sets the output high or low for the required time.

Counter output is split providing firing signal for a pair of transistors in the Inverter Bridge. Motor’s torque will be constant in the area where the volts per Hz (V/Hz) ratio are held constant where V is the rms value of voltage applied to the motor. A standard induction AC motor provide a speed regulation of 1.5 to 3% (slip of the motor) of the base speed. With the inverter, constant torque could be provided to base speed and constant power to 1.5 times base speed. The low-end controllable speed is about 300 rpm.

There are three main types of inverter drives for induction motors:

- Open loop Variable Frequency Drives
- Closed loop Variable Frequency Drives
- Field Oriented Control Drives

The amount of control increases as you go down the list. The first two options are sufficient for a majority of applications. The closed loop inverter is actually capable of excellent speed control. The one problem with the first two inverter drives is the slow response to changes in speed. This problem has been solved by the new but developed technology of the vector drives

**Open Loop Inverters**

Open loop inverters are also referred to as an open variable frequency drive and an open loop AC variable speed drive. The main difference between the open loop inverter and the others is that it does not have any form of velocity feedback. Without feedback, precise speed control on an induction motor is difficult due to the inherent slip of the motor. Since the synchronous motor does not slip, its speed can be controlled very accurately with an open loop inverter.

The main purpose of the open loop inverter is supply the motor with an AC voltage signal with a varying frequency proportional to the reference speed signal. In order to create an AC signal with a varying frequency, it is first necessary to convert the AC supply input to a DC signal. This is accomplished with a rectifier, usually a full wave rectifier. The DC supply is then sent to an inverter. An inverter along with the control circuits creates a switching voltage output in a manner similar to that of the DC chopper drive. The switching techniques use transistors to turn on and off the voltage signal at a high frequency. By varying the length of time that the voltage signal is on, the inverter creates an average voltage that resembles a sinusoidal curve.

The speed reference signal represents the speed of the rotating magnetic field which closely correlates with the no-load speed of an induction motor. But with increased load, the slip of the motor will increase causing the speed to decrease. For application with low loads or situations that do not require accurate speed control, the open loop inverter drive should work just fine. With increasing application demands, you may want to look to a closed loop inverter drive or vector drive which both incorporates some form of velocity feedback.
**Closed Loop Inverter Drives**

Closed Loop Inverter Drives is also referred to as a closed loop variable frequency drive or a closed loop variable speed drive. A basic schematic of the closed loop inverter drive can be seen to the right. It is very similar to the open loop version with the addition of the velocity feedback.

![Fig 2.13 Closed loop inverter drive](image)

This closed loop inverter drive is commonly used to control the induction motor, which requires an ac signal to produce motion. The speed of the motion is related to the frequency of the signal. In order to create an AC signal with a varying frequency, it is first necessary to convert the AC supply input to a DC signal. This is accomplished with a rectifier, usually a full wave rectifier. The DC supply is then sent to an inverter. An inverter along with the control circuits create a switching voltage output in a manner similar to that of the DC chopper drive.

The switching techniques use transistors to turn on and off the voltage signal at a high frequency. By varying the length of time that the voltage signal is on, the inverter creates an average voltage signal that resembles a sinusoidal curve. The current waveform created by this switched voltage represents a sinusoidal curve much more accurately than the voltage waveform.

The difference between the open loop and close loop inverters is that the speed reference signal represents the speed of the rotor rather than the speed of the rotating magnetic field or the no load speed. The speed reference signal is compared with the feedback signal and corrections are made for any error. In loaded applications, the speed of the rotating magnetic field is increased beyond the desired speed in order to compensate for induction motor slip.

The closed loop inverter drive is capable of precise speed control. In comparison with a DC speed control system, the inverter drive and induction motor system are reasonably similar in price, can handle higher speeds, and are more robust. Applications that do not require high torque at low speeds, such as variable speeds fans and pumps are great for the closed loop inverter drive.

Although the closed loop inverters are capable of most speed control applications, it does lack in one area, which is applications that involve high performance transient response. This basically means that the inverter drives are not capable of shifting from one speed to another very quickly, however advances in AC vector drive can handle this situation. One example when an AC vector drive would be necessary is on a lathe machine.
VECTOR CONTROL

With a brushed motor, it is possible to apply a current directly to the winding of the rotor through the brushes. This allows for direct control of the torque of the motor. Because of this feature, a control speed drive can quickly alter the speed of a dc motor from one steady state to another. On the other hand, the AC induction motor has no direct control over the currents of the rotor, so a rapid transient response from one steady state to another was a huge problem.

With appropriate feedback such as encoder, an induction motor can also be operated by vector control. Power section of vector drive is similar to that of inverter drive. This technology uses a PWM synthesized sinusoidal waveform to control speed. But the inverter firing control circuitry is complex in this case. Vector drive permits motor current and voltage to be varied independently. With quick instantaneous changes in the stator current, quick instantaneous changes occur in the rotor current. Very simply put, the vector drive manipulates the motor torque by indirectly controlling the current of the rotor through the stator windings. Calculating and quantifying the corresponding changes in the rotor current is the genius of the vector drive. To achieve vector control, the drive must monitor the current of the stator windings by closing the current loop. Some vector control drives also require position feedback from a shaft mounted encoder.

With this control tighter speed regulation approaching 0.01% of set speed may be attained. Vector control allows controllable speed ranges from about 5 times base speed to zero speed. The constant horse power range is about 3.5 times the base speed and full rated torque from rated speed down to zero speed. Induction motors with vector controls can be applied in high performance adjustable speed applications. For positioning control they can be used with proper positioning controllers.

![fig2.14_flux_vector_control](image)

**Fig.2.14 Flux vector control**

To emulate the magnetic operating conditions of a DC motor, i.e. to perform the field orientation process, the flux-vector drive needs to know the spatial angular position of the rotor flux inside the AC induction motor. With flux vector PWM drives, field orientation is achieved by electronic means rather than the mechanical commutator/brush assembly of the DC motor.
Firstly, information about the rotor status is obtained by feeding back rotor speed and angular position relative to the stator field by means of a pulse encoder. A drive that uses speed encoders is referred to as a “closed-loop drive”. Also the motor’s electrical characteristics are mathematically modelled with microprocessors used to process the data. The electronic controller of a flux-vector drive creates electrical quantities such as voltage, current and frequency, which are the controlling variables, and feeds these through a modulator to the AC induction motor. Torque, therefore, is controlled indirectly. This type of drive has following advantages.

- Good torque response
- Accurate speed control
- Full torque at zero speed
- Performance approaching DC drive

Flux vector control achieves full torque at zero speed, giving it a performance very close to that of a DC drive. To achieve a high level of torque response and speed accuracy, a feedback device is required. This can be costly and also adds complexity to the traditional simple AC induction motor. Also, a modulator is used, which slows down communication between the incoming voltage and frequency signals and the need for the motor to respond to this changing signal. Although the motor is mechanically simple, the drive is electrically complex.

**Direct Torque Control**

Field orientation can be achieved without feedback using advanced motor theory to calculate the motor torque directly and without using modulation. The controlling variables are motor magnetising flux and motor torque.

With DTC there is no modulator and no requirement for a tachometer or position encoder to feed back the speed or position of the motor shaft. DTC uses the fastest digital signal processing hardware available and a more advanced mathematical understanding of how a motor works. The result is a drive with a torque response that is typically 10 times faster than any AC or DC drive. The dynamic speed accuracy of DTC drives will be 8 times better than any open loop AC drives and comparable to a DC drive that is using feedback. DTC produces the first “universal” drives with the capability to perform like either an AC or DC drive.
**Brushless Controls:**

When driving a three phase brushless motor by energising two of the three windings at a time, simple Hall Sensor feedback can be used. This is called six-step communication system. It may be termed as DC brushless also. When a sinusoidal waveform is applied to the motor windings on a continuous basis, the term “AC brushless” is used.

Dynamic braking may be used with this control. Whenever a motor is stopped faster it starts behaving as generator and the power thus generated may be shunted through the external resistor and dissipated as heat. If the application has high inertial load, regenerative braking could be used in which power generated by the motor is fed back to the incoming power line saving energy.

Now also offered adjustable speed drive that replaces sine coded PWM with a flux linkage PWM Bridge. It employs both digital and analog techniques. The flux linkage approach takes advantage of the fact that induction motor behaves best when a circular rotating magnetic field is formed in the gap. Speed control is from 3-60 Hz (20:1) or from 3-120 Hz (40:1). Torque control is by voltage (v)/frequency (Hz) ratio selection with full starting torque at very low frequencies.

Higher performance requirements demanded from the drive in case of machine tool spindle drive are fast dynamic response, regeneration brake and wide speed range. A microprocessor-controlled transistor PWM converter with an induction motor forms a powerful spindle drive system.

Advantages of variable frequency drives:-

1. Compactness-wt is 35-65% less than that of DC motor
2. Brushless- so maintenance free
3. High protection standards-completely sealed construction
4. External motor cooling system
5. Stable speed control over wide rpm range-1:150 ratio .Spindle speed range with constant power is normally in the ratio of 1:4 over the base speed. Wide constant power range enables both high-speed cutting and low speed heavy operations.

Shockless starting and acceleration/deceleration Variable frequency AC drive can be used for high speed application (upto 5400rpm) from 1.5 Kw to 45 Kw capacity.

While selecting types of drive for a machine tool, various merits and demerits are considered depending upon the requirements of the application.
Advantages of AC drives

- They use conventional, low cost, 3-phase AC induction motors for most applications.

- AC motors require virtually no maintenance and are preferred for applications where the motor is mounted in an area not easily reached for servicing or replacement.

- AC motors are smaller, lighter, more commonly available, and less expensive than DC motors.

- AC motors are better suited for high speed operation (over 2500 rpm) since there are no brushes, and commutation is not a problem.

- Whenever the operating environment is wet, corrosive or explosive and special motor enclosures are required. Special AC motor enclosure types are more readily available at lower prices.

- Multiple motors in a system must operate simultaneously at a common frequency/speed.

- It is desirable to use an existing constant speed AC motor already mounted and wired on a machine.

- When the application load varies greatly and light loads may be encountered for prolonged periods. DC motor commutators and brushes may wear rapidly under this condition.

- Low cost electronic motor reversing is required. It is important to have a back up (constant speed) if the controller fails.
Advantages of DC drives

- DC drives are less complex with a single power conversion from AC to DC.
- DC drives are normally less expensive for most horsepower ratings.
- DC motors have a long tradition of use as adjustable speed machines and a wide range of options have evolved for this purpose:
  - Cooling blowers and inlet air flanges provide cooling air for a wide speed range at constant torque.
  - Accessory mounting flanges and kits for mounting feedback tachometers and encoders.
  - DC regenerative drives are available for applications requiring continuous regeneration for overhauling loads. AC drives with this capability would be more complex and expensive.
  - Properly applied brush and commutator maintenance is minimal.
  - DC motors are capable of providing starting and accelerating torques in excess of 400% of rated.
  - Some AC drives may produce audible motor noise, which is undesirable in some applications.
- Selection of motor and drives is thus made as per the services required from the machine being designed. Operation and maintenance staff should be well aware with the various drives for the optimum utilization of the machines tools and equipment.
Chapter 3 : REGULATION OF SPEED AND FEED

3.1 SELECTION OF SPEED

There has always been a need to adjust the speed and feed of equipment for each machining operation.

The rotating speed of the spindle for any job depends on the following formulae

\[ n = \frac{1000 \cdot V}{\pi \cdot d} \]

Where, \( V \) = cutting speed in M/min
\( d \) = diameter of job/cutter
\( n \) = rpm

The ratio between maximum and minimum output speed (\( R \)) is selected based on the application of the machine

\[ R = \frac{n_{\text{max}}}{n_{\text{min}}} \]

\[ n = \frac{V_{\text{max}}}{\pi \cdot d_{\text{min}}} \text{ and } n_{\text{min}} = \frac{V_{\text{min}}}{\pi \cdot d_{\text{max}}} \]

Where, \( V_{\text{min}} \) and \( V_{\text{max}} \) are the minimum and maximum cutting speeds
\( d_{\text{min}} \) and \( d_{\text{max}} \) are the smallest and largest diameter of jobs to be manufactured/cutter to be used
\( n_{\text{max}} \) and \( n_{\text{min}} \) are the maximum and minimum speed of spindle in rpm

\[ R = \frac{n_{\text{max}}}{n_{\text{min}}} = \frac{V_{\text{max}}}{V_{\text{min}}} \cdot \frac{d_{\text{max}}}{d_{\text{min}}} \]

In machine tools gear ratio variation is used normally in geometric progression.

Table 3.1 Gear ratio for speed variation in machine tools

<table>
<thead>
<tr>
<th>Type of m/c Tools</th>
<th>Speed ratio (R) or ratio of max. and min. o/p speed</th>
<th>No. of steps</th>
</tr>
</thead>
<tbody>
<tr>
<td>Planing machine</td>
<td>6 to 10</td>
<td>6 to 9</td>
</tr>
<tr>
<td>Turning machine</td>
<td>50 to 200</td>
<td>12 to 18</td>
</tr>
<tr>
<td>Milling and Boring machine</td>
<td>upto 400</td>
<td>upto 36</td>
</tr>
<tr>
<td>Presses</td>
<td>1</td>
<td>1</td>
</tr>
</tbody>
</table>
GEOMETRIC PROGRESSION:

If rpm varies from \( n_1 \) to \( n_2 \)

\[
\begin{align*}
    n_2 &= n_1 \phi \\
    n_3 &= n_2 \phi = n_1 \phi^2 \\
    & \vdots \\
    n_z &= n_1 \phi^{(z-1)} \\
    \phi &= \sqrt[n_z]{\frac{n_z}{n_1}} = \frac{(z-1)}{\sqrt[n]{R}}
\end{align*}
\]

In order to provide for a wide range of operating speeds together with adequate torque at lower spindle speeds, it is necessary to use gearboxes that enable the required spindle speeds range to be covered in a number of discrete steps. Following arrangements are applied in speed gearboxes for changing various speed steps.

3.2 GENERAL TYPES OF SPEED GEAR BOXES

There are a great variety of gearbox designs and can be classified according to their layout and the mode of speed change and the constructional arrangement. According to layout gearboxes can be classified into following two categories;

1. **Gear boxes built into the spindle head**:
   In most of the medium and heavy machine tools, these gearboxes are employed. The advantages of the layout are a more compact spindle drive, higher concentration of control, fewer housing type parts and less assembly work involving the fitting of joining surfaces. But such layout has certain distinct disadvantages. Due to compactness there are a possibility of transmission of vibration and heat from gearbox to spindle. It is difficult to employ a flexible transmission to the spindle along with toothed gearing.

2. **Gear boxes with a divided drive**:
   In case of small and medium size machines headstock and gearbox are designed as a separate unit connected by a belt transmission. In divided drive heat evolved by frictional losses or vibration developed in the gearbox are not transmitted to the spindle head. So maintenance problems are reduced to the greater extent and life of spindle is increased. The spindle runs smoothly at high speeds, so high surface finish and longer tool life in finishing operation.

   In some machine tools two different types of drives to the spindle are employed. Belt drive is used for finishing operation at high spindle speeds and gear drive for roughing purpose.
3.3 MODE OF SPEED CHANGE IN GEARBOXES

Speed changing methods depend upon the frequency with which the speeds are to be changed and duration of the working movement. Where speeds are to be changed frequently, mechanism of friction clutches are provided for rapid change while machine is running. In general it is provided for such machine where machining time for each operation element is small, so speeds of gearboxes are required to be changed quickly without stopping the machine.

i.) Gear Boxes with Change Gears:

Speeds are changed by changing the gears of a group transmission between adjacent shafts with a constant centre distances. In most of the lathe machines horizontal boring machine etc., change gear are provided for threading operation. As speed change is carried out manually by replacing the gears, it is impossible to make conflicting engagement, so no interlocking devices are needed. Requirement of total number of gear is less as the same gears can be placed in reverse order. The principal drawback is that a great deal of time is lost in changing speeds. Another drawback is that the cover enclosing them is difficult to seal properly without a gasket. Housing with flanged walls are used and oil catch rings are provided to prevent oil leakage's, in change gear arrangement.

To reduce the time loss change gears may be installed on splined shaft or on tapered shaft journal with woodruff key. Change gear may be fixed in axial direction on the shaft by quick acting splined collars with locking devices or by C washers which can be readily removed to one side after slightly loosening the clamping nut or screw. The size of the nut or screw is such that it can easily pass through the bore of the change gears.
ii.) Gearboxes With Sliding Gears

In general double cluster gear, triple cluster gear or a set of two or three gears mounted on one another, is dependent upon the availability of the space for gear shifting on the shaft. Width of the gear, having lesser width out of the two and gear of triple cluster gear, decides the positioning of the gears on other shaft.

For fig.3.2d it is necessary to have \((r_1-r_2) \geq 2m\) or

\[
\frac{mz_1}{2} - \frac{mz_2}{2} \geq 4
\]

The advantage of sliding gear transmission is that they are capable of transmitting high torque, being comparatively small in radial dimension. The gear wheels, in speed transmission, relating the spindle, are only in mesh. This prevents the other gears of gearbox from wearing out. Spur gears are commonly used in sliding gear mechanism.

But speed changing is quite complicated and involves the disengagement of the gearbox drive, braking the gearbox shaft, slow rotation for shifting sliding cluster gear in and out of engagement. It cluster gears are shifted while they are rotating at faster speed or if two transmission of a single group between adjustment shafts are simultaneously engaged, break down may occur. So, interlocking devices are provided to prevent such conflicting occurrences.
iii.) GEAR BOXES WITH CLAW CLUTCH:

In claw clutches the forces of engagement are distributed more uniformly and over a greater number of working surface. Claw clutch requires axial movement for engagement or disengagement. They permit helical herringbone gears to be employed in the drive and can be shifted with less effort than that needed to shift cluster gear.

On the other hand the claw clutches will not stand the change over of highly contrasting speeds on the run. The clutch teeth or jaw may be broken if engagement is made with a large difference in the rotational speeds of clutch members. There are also inherent power losses and wear due to the rotation of idle gears. It is therefore necessary to disengage the drive and to slow down the shafts carrying the members. Jaw clutch is generally used in combination with sliding gears in an arrangement that exclude idle rotation of gears.
iv) Gear Boxes with Friction Clutches

Friction and magnetic clutches provide quick smooth engagement of gears on the run under load. The arrangement permits helical and herringbone gears to be used in the gearboxes. There are some disadvantages of the above system:

a) Power losses and wear due to rotation of idle gears.
b) Large axial and radial dimensions in the transmission of high torque.
c) Reduction of mechanical efficiency because of friction in clutches when disengaged.
d) Generation of heat and its effect on the spindle unit.
e) The needs for frequent adjustment to avoid slipping.

Generally electromagnetic friction clutches are used to have remote control.

v) Synchromesh Unit:

Synchromesh arrangement is a modification of the gearbox with claw clutch. Here sliding claw clutch and the corresponding gear pinions are provided with friction cones.

The synchronising action takes place during the initial movement of the claw clutch lever when the friction cones are pressed together. Thus in changing down, the force of friction between the surfaces increases the speed of the drives pinion of shaft until it is equal to the main shaft speed. A continued movement of the gear lever then engages the claw clutch. By this means noiseless changing can be obtained without declutching or stopping the system.

3.4 STEPLESS VARIABLE SPEED DRIVE:

The use of infinitely variable speed unit has proved its worth in countless industrial applications. For best performance of machine tools requiring stepless speed changing with precise control following types of units stand unmatched.

VARIABLE SPEED DRIVE WITH TWO PAIR OF ADJUSTABLE CONICAL SHEAVES

The casing of the drive houses driving and driven pulley, each sliding on surfaces coated with molykote to resist wear, corrosion and stick. The controlling hand wheel or other device operates the linkage to give movement of the driving pulley and the driven pulley responds automatically. Opening or closing the driving pulley cones causes a corresponding drop or rise of the wide V-belt and the spring loaded driven pulley maintains the belt tension thus providing a step less speed variation. When low speeds are required, the shaft on which the driven pulley is fixed is coupled with a reduction gearbox.

In case automatic control of the speed is required, a servo unit consisting of an electric motor and gear unit can be attached to regulating shaft (in place of hand wheel). Such units are supplied with limit switches to make them operative only in the prescribed range.

Normally these drives are manufactured upto a capacity of 4KW with an infinite variation of a maximum speed ratio of 1:8 from the lowest rpm. of 0,41 to a maximum rpm of 4080. Higher capacity drive in combination with a reduction gearbox or planetary gearbox can also be used to provide variable spindle speeds.
PIV GEAR DRIVE

It is positive infinitely variable drive, which is coupled with a conventional induction motor to obtain step less variation of output speed. In the system an endless chain is mounted on bevel disc pairs by sliding the discs in axial direction, the distance between two discs of each pair is varied. The control system is made such that while one disc pair slides close together, the other disc pair moves away from each other. With this chain running diameter of two disc pairs could be varied.

Depending upon the power transmission applications, various types of chains are used.

(i) Ring roller chain: It consists of steel links with cradle joints joined with each other. Over these links reliable ring rolls are fastened. This type of chain is suitable for chain speed up to 25 meter per second and 37.5 K.W. power.

(ii) Rocker pressure pin chain: In this type of chain, cradle joints transmit force over the bevel disc pairs. The relative light design makes it possible to have chain speed up to 30 meter per second. Such chain can transmit power up to 27.5 K.W.

(iii) Cylinder roller chain: Each chain member of this type of chain contains two rotating rollers supported with each other. Advantage of such chain is that the speed could be regulated even when the discs are in stationary condition. It can transmit power up to 18 KW with the maximum speed of 20 meter per second.

(iv) Lamination chain: All the three above mentioned chains run over smooth hardened and ground disc pairs. But the individual member of the laminated chain contains axially displaceable laminated sheets, which make it to mesh with teeth cut on bevel discs.

PIV drives with the first three types of chain have smooth hardened and ground disk pairs. Power transmission between spherical faces of the chain elements and the smooth surface of the conical disks relies on lubricated metal to metal traction.

For power transmission, axial squeeze is necessary between chain and conical disks. Cam arrangement and other mechanism produce this. The axial load is carried to the chain via pressure rollers and movable disk on one side, and fixed disk, supporting rollers and shaft on the other. Axial squeeze load is supported inside the rotating shaft without loading the housing bearings. When running idle, a compression spring provides adequate contact pressure.

Fig 3.4 View of typical PIV drive
Control Devices

In order to take maximum advantages of infinitely variable drives, control devices are necessary. The devices, which allow pre-setting of speed, are always preferred.

Following types of control are generally available:
1. Hand control
2. Mechanical control
3. Electro mechanical remote control
4. Hydraulic servo control
5. Pneumatic control

What ever system is adopted, axial movement of discs is achieved either with the help of lead screw - box nut or with the cylinder piston arrangement:

a) Screw cut on the adjusting spindle meshes with the box nut fixed on the adjusting blocks on each end of which one disc pair is mounted. Thus when spindle end of adjusting blocks come closer, the other end move apart.

b) In the hydraulic servo system, a pump incorporated into the drive feeds the oil to the hydraulic control device. The control valve regulates the oil into the rotating cylinders of disc assemblies. The hydraulic support pressure provides an additional axial load and controls the axial displacement of the movable disks. In addition, the pressurised oil is needed to provide forced feed lubrication to the moving parts of the drives. The servo hydraulic allows preselecting out put speeds when the drive is at stand still.

A geared motor can drive adjusting spindle or the control knob also. Slipping coupling additional switches and potentiometer are available to provide remote control to the unit.

Fig 3.5 Servo hydraulic control of PIV drive
**Selection Procedure**

Following points should be taken into consideration while selecting PIV drive for certain system:

- a) Smallest pitch circle Diameter of gear / sprocket = 2.5d
- b) Smallest pitch Diameter of v belt pulley = 4d
- c) Smallest pitch Diameter of flat belt pulley = 6d

**PIV DRIVE WITH PLANETARY GEARBOX**

In some design PIV system is joined together with planetary gear drive. With this the large portion of the power could be transmitted over the planetary gears, while the rest passes through the embracing PIV drive. There are two main advantages of the system.

1) The system as a whole is compact and light in design. The power is transmitted through more number of parts though which force and torque could be transmitted.

2) Larger range of gear ratio could be achieved i.e. higher speed range is available on output shaft.

**STEP LESS BALL DISC DRIVE**

For step less drive with small power, rolling drive is more economical and compact as compared to PIV. The ball disc drive can transmit upto 3 kW. With the help of this drive, constant torque upto above zero rpm could be obtained. Input shaft and output shaft is connected with their individual discs.
Axes of rotation of the discs are at a distance "e" with each other. Between these two discs, a steel ball cage is fitted, which transmit the torque from one disc to the other. The position of the ball cage is adjustable with the help of a screw. Speed of output shaft will be zero, when axis of input disc and that of ball cage coincide with each other.

Ball cage is eccentrically placed with respect to output disc. The balls roll off the output disc in this condition and take another track after each revolution. Necessary thrust on the discs are provided with the help of disc springs. The discs can be adjusted in stationary condition. The advantage of the system is that the adjusting force requirement is quite low and speed transmission is uniform.

Fig 3.7 Stepless ball disc drive
3.5 HYDRAULIC DRIVES FOR STEP LESS REGULATION:

The usual hydraulic system for spindle drive is based on a constant speed electric motor driving a hydraulic pump, which then supplies hydraulic oil under pressure to drive a hydraulic motor. The pumps and motors may be of either the fixed or the variable displacement type. The closed circuit is most suitable for frequently reversible speed and feed drive where jerking or stick slip motion is not permissible. For varying the speed of the hydraulic motor following two methods are commonly used.

(i) Varying the discharge of the pump.
(ii) Varying incoming quantity of oil to the hydraulic motor.

This is analogous to the Leonard method of electrical circuit. That is why it is termed as "hydraulic Leonard rule".

The advantage of hydraulic systems incorporating both variable delivery pumps and variable displacement pumps are:

a) Wide range of steplessly variable speeds.
b) High rotating rigidity
c) Possibility of repeated and sudden reversal
d) No backlash
e) Transmission of high power and torque's

As excessive heating of oil takes place in this process. So an effective heat exchanger should be provided for cooling of the oil.

Fig 3.8 Hydraulic circuits for stepless speed regulation
HYVARI DRIVE

It is a compact hydraulic drive comprised of an axial piston pump and hydraulic motor housed in a single body. Input shaft (1) drives the pump (2), which delivers the pressurised oil to the axial piston motor (3) through a control disc located in the middle between pump and motor. Finally rotary motion of the motor is transmitted to the driven shaft (4) through rotary housing (6) fitted in the stationary body (5).

Function of the system can be described as follows:

If in a hydraulic pump with stationary pump housing, pump delivery at the outlet is blocked, oil pressure would theoretically increase till a few parts get damaged. But here pump housing is mounted on bearings and so it is free to rotate with the same speed as that of its rotor. In this process a part of the oil delivered by the pump is passed in to the hydraulic motor. The quantity of oil flowing to the hydraulic motor inlet could be regulated.

The regulation of pump discharge rate and input rate to this motor could be achieved by changing angles $\alpha_2$ and $\alpha_3$ respectively of their swash plates. Both the swash plates are connected together through linkages and can be swung together. The whole unit is always full of oil. There are three different operating conditions:

1. Swash plate of pump is a right angle ($\alpha_2 = 0$) while that of motor is fully inclined - pump does not deliver oil so output (driven) shaft does not rotate,
2. Swash plate of motor is at right angle ($\alpha_3 = 0$). The torque will depend upon the rotor of the motor. The output shaft rotates through the reaction torque over the pump swash plate. Speed of input shaft = speed of the output shaft.
3. Both motor and pump with maximum inclined swash plate:
   a) Hydraulic energy of the system acts on motor torque over the rotating housing (6)
   b) Mechanical energy acts as reaction torque over the pump swash plate and rotating housing (6)

Thus in the system a part of work energy is used for mechanical work while the remaining part is used for hydraulic purpose. So, overall efficiency is better than that of other systems.
3.6 VARIABLE SPEED GEARED MOTORS:

Electric motors supply maximum power only at base speed. So to obtain optimum results variable speed motor is required to be operated at higher speed and suitable speed reducer has to be fitted with it to get desired speed. Planetary gear box and cyclo drive reducers are more compact type for the higher speed reduction ratio.

PLANETARY GEAR BOX:

Planetary gearing consists of a system of gears in which two coaxial gears are connected by gears called planets mounted on a carrier which is also coaxial. Planetary gearbox is used in machine tools for speed / feed drive either as fixed ratio drive e.g. reduction gearbox, mullet stage gearbox or as a differential drive to increase speed range in combination with variable drive system.

According to the gearing arrangement planetary gearing are classified as simple planetary gearing system and compound planetary gearing system Simple planetary system has four basic elements:

1) Sun gear (S)
2) Annulus gear (A)
3) Planet gear carrier
4) Planet gears (P)

In compound planetary gearing, two or more simple planetary system is combined such that two members of each train are connected to two members of another train. In this case internal gear could be replaced by external gear. In fixed ratio drive, one of the three coaxial members is locked and other two are used as input or output member as the case may be. In a differential drive all the three coaxial members are allowed to rotate with any one of them as the driver and the other two as driven or vice versa.

Simple Planetary Gear Train:

General equation for simple planetary gear train when the system is used as a differential drive is:

\[
n_S = n_C\left(\frac{z_S + z_A}{z_S}\right) \pm n_Az_A \frac{z_S}{z_S}
\]

![Fig 3.10 Simple planetary gear train](image)
Negative sign is used if annulus gear and carrier rotates in the same direction where \( n \) indicates the speed (rpm) of the respective gears while \( Z \) denotes the number of teeth. To drive equation for fixed ratio drive, put speed \( n \) for the locked gear as zero for example if annulus gear is locked, general equation will be

\[
n_S = \frac{n_C(z_S + z_A)}{z_S}
\]

**COMPOUND PLANETARY DRIVE WITH ONE SUN GEAR AND ONE ANNULUS GEAR**

Two sets of planetary gears are mounted on a single planet carrier. One set (\( P_1 \)) meshes with the sun gear while the other set (\( P_2 \)) meshes with the annulus gear. Gears of both the sets are fixed together as cluster gears. General equation for differential drive is given below.

Here also negative sign is used when annulus gear and carrier rotate in the same direction and positive sign is used if annulus gear rotates in a direction opposite to that of carrier. Equation for fixed ratio drive could be divided by putting speed \( (n) \) of the locked gear as zero in the above equation.

\[
n_S = n_C\left(1 + \frac{z_Az_{P1}}{z_Sz_{P2}}\right) \pm n_A \frac{z_Az_{P1}}{z_Sz_{P2}}
\]

*Fig.3.11 Compound planetary gear*
COMPOUND PLANETARY DRIVE WITH TWO SUN GEARs

In this case, two sets of planetary gears mesh with the two-sun gears respectively. In other words, the other set of planetary gears meshes with the second sun gear \((S_2)\) in place of annulus gear. So no annulus gear is there in this type of gear train. General equation of transmission for differential drive is:

\[
n_{S1} = n_C \left( 1 - \frac{z_{S2} z_{P1}}{z_{S1} z_{P2}} \right) \pm n_{S2} \left( \frac{z_{S2} z_{P1}}{z_{S1} z_{P2}} \right)
\]

Fig 3.12 Compound planetary drive with two sun gear

Hear positive sign is used for same direction of rotation for the second sun gear \((S_2)\) and carrier gear. It second sun gear \((S_2)\) rotates in opposite direction, negative sign is used. For fixed ratio drive the speed of locked gear may be put as zero.

In the same way general equation of transmission could be divided for the system where two annulus gears are used in place of two sun gears.

EQUAL BEVEL GEAR PLANETARY SYSTEM

In this type of gear train, 4 similar bevel gears are arranged in such a way that two acts as sun gears and the other two as planet gears.

Fig 3.13 Equal bevel gear planetary system
In system (i) F is connected to the planet carrier. The general equation of transmission when the system is used as a differential drive is:

\[ n_E = 2n_F \mp n_D \]

Negative sign is used when F and D rotate in the same direction and positive sign is used when D rotates in a direction opposite to that of F.

In system (ii) D is connected to planet carrier. So the equation of transmission will be:

\[ n_E = 2n_D \mp n_F \]

Negative sign is used for F and D rotating in the same direction and positive sign is used when F rotates in a direction opposite to that of D. In the above equations by fixing any one of the members i.e. by putting corresponding speed (n) equal to zero equation for fixed drive relationship could be derived.

In Fig 3.14 reduction of speed in the 2nd stage is obtained through planetary gearing by shifting the annulus gear to the fixed position with the help of jaws thus increasing the torque to the required level.

ADVANTAGE:

Planetary gearbox has following advantage:

1) Compact construction with possibility to obtain high transmission ratio.
2) Coaxial input and out put shaft.
3) Load sharing between several pinions.
4) Low flywheel effect.
5) Silent running
6) High efficiency
7) Complete balance of static forces of gear train
8) Reduced sensitivity to shock loading

Fig 3.14 Two stage Planetary speed gearbox for


There are essentially four major components in the cyclo gear box.

i) High speed shaft with eccentric bearing.

ii) Cycloid discs.

iii) Ring gear housing with pins and rollers.

iv) Slow speed shaft with pins and rollers.

As the eccentric rotates, it rolls the cycloid discs around the internal circumference of the ring gear housing. As the cycloid discs travel in a clock-wise path around ring gear, discs themselves turn in a counter-clockwise direction around their own axes. The teeth of the cycloid discs engage successively with the pins of the fixed ring gear, thus providing a reverse rotation at reduced speed. The number of cycloid teeth on the cycloid disc determines the reduction ratio. There is one less tooth per cycloid disc than there are rollers in the ring gear housing, which results in the reduction ratio being numerically equal to the number of teeth on the cycloid disc.

Therefore for each complete revolution of the high-speed shaft the cycloid disc move in the opposite direction by one tooth. The rotation of the cycloid discs is transmitted to the slow speed shaft via the pins and rollers projecting through holes in the cycloid discs. Reduction ratios from 9:1 to 85:1 are available from a single stage. Higher ratios can be obtained from multiple stages. As there is no sliding friction, efficiency of the drive can be as high as 95%. Cyclo gearboxes have the capacity of withstanding high short term overload peaks without damage because 65% of the cycloidally shaped teeth are engaged simultaneously with the ring gear rollers. Since the mass moment of inertia is very low the cyclo-drive responds quickly to acceleration, deceleration and reversing torque.
3.7 MAINTENANCE OF SPEED GEARBOXES

Proper care and timely maintenance of the speed gearbox is necessary to get desired service from the equipment in which it is installed. Defects experienced are tabulated below along with the probable causes and method of rectification. This could be utilized as guidelines during the trouble-shooting of equipment.

Table 3.1 Generally occurring defects in speed gearboxes and their rectification

<table>
<thead>
<tr>
<th>S.No</th>
<th>DEFECTS EXPERIENCED</th>
<th>PROBABLE CAUSES</th>
<th>METHOD OF RECT.</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.</td>
<td>While switching on, at one of the speeds, it is not possible to give motion to one shaft while the shaft can rotate at other speeds.</td>
<td>Two or more speeds are engaged simultaneously</td>
<td>To adjust the blocking mechanism provided to prevent simultaneous engagement of several speeds.</td>
</tr>
<tr>
<td>2.</td>
<td>The motor shaft rotate but the shafts in gearbox do not rotate.</td>
<td>The key of the coupling between the electric motor and the shaft of the speed box is damaged.</td>
<td>Fit a new key away</td>
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<tr>
<td>3.</td>
<td></td>
<td>a) The spline shaft has burs.</td>
<td>Take out shaft and clean.</td>
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<td></td>
<td></td>
<td>b) Clearance between cluster gear and shaft is not sufficient.</td>
<td>Grind the shaft, if necessary</td>
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<td></td>
<td></td>
<td>c) Lubrication is absent</td>
<td>Lubricate</td>
</tr>
<tr>
<td>4.</td>
<td>During work speed box gets heated beyond 50°C</td>
<td>a) Oil level is not sufficient in reservoir.</td>
<td>To fill oil up to the level.</td>
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<td></td>
<td></td>
<td>b) Side clearances in one or more gears are absent.</td>
<td>Check gears as per drawing.</td>
</tr>
<tr>
<td>5.</td>
<td>On shifting a particular gear for changing speed, the speed is not changed.</td>
<td>a) Key connecting the level with the shaft on which it is mounted, is sheared.</td>
<td>To fit a new keyway.</td>
</tr>
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<td></td>
<td></td>
<td>b) Fork for shifting gear, is broken.</td>
<td>Change the fork.</td>
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<tr>
<td></td>
<td></td>
<td>c) The keyway in the lever is not properly slotted.</td>
<td>Dismantle the speed-changing lever to weld the incorrect keyway and cut a new one.</td>
</tr>
<tr>
<td>6.</td>
<td>The speed gets discharged by itself during the working of speed gear box.</td>
<td>a) Gears are not fully engaged.</td>
<td>Provided full meshing by lengthening or shortening the levers and fork for shifting gears.</td>
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<tr>
<td></td>
<td></td>
<td></td>
<td>Bore the holes for shafts on</td>
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<tr>
<td><strong>b)</strong></td>
<td>The shafts are not properly fitted in the gearbox. This results in the generation of axial forces, which shift the spur gears.</td>
<td>H.B.m/c, providing the required accuracy of axis.</td>
<td></td>
</tr>
<tr>
<td></td>
<td>c) Excessively heavy speed changing lever and weak spring in the fixer of the lever. In such a case, the lever by its own weight puts off the speed.</td>
<td>Decrease the wt. Of changing lever put a stiffer spring in the fixing device.</td>
<td></td>
</tr>
<tr>
<td></td>
<td>d) Fixing spring of lever may get loosened due to vibration in the machine.</td>
<td>Use stiffer spring remove, cause of vibration.</td>
<td></td>
</tr>
<tr>
<td>7.</td>
<td>During reversing output shaft stops.</td>
<td>The reversing mechanism is defective or has been wrongly adjusted.</td>
<td></td>
</tr>
<tr>
<td></td>
<td>The reversing mechanism is defective or has been wrongly adjusted.</td>
<td>Regulate reversing mechanism, change the worn out parts if necessary</td>
<td></td>
</tr>
</tbody>
</table>
|  8. | Peeling of working surface of teeth. | a) Fatigue in the material.   
|   | b) Oil of lower viscosity | Change the gear  
|   | a) Fatigue in the material. b) Oil of lower viscosity | Change the oil. |
|  9. | Burr on the working surface of teeth. | Gear drive is working without lubrication. |
|   | Gear drive is working without lubrication. | Provide proper lubrication. |
| 10. | Correct profile of teeth is quickly lost. | Dirt, abrasive material or metallic dust has gone inside. |
|   | Dirt, abrasive material or metallic dust has gone inside. | Clean. |
| 11. | Wear in the gear and pinion, breakage of teeth. | a) Natural wear b) Over loading of the gear drive. c) Foreign matter has fallen into gear drive. |
|   | a) Natural wear b) Over loading of the gear drive. c) Foreign matter has fallen into gear drive. | Change gear & pinion To ensure from over loading. |
| 12. | One side of teeth is damaged. | a) Gears have been changed while m/c was in running condition. |
|   | a) Gears have been changed while m/c was in running condition. | Check interlocking Device if any |
| 13. | Increased noise is coming from the drive, accompanied by impact. | a) The centre distance between the gears is more than the calculated one. b) Big side clearance. |
|   | a) The centre distance between the gears is more than the calculated one. b) Big side clearance. | Decrease the centre distance. Make correction in the drive at the same time in the speed ratio. |
Chapter 4: FEED GEARBOXES AND MECHANISMS

4.1 ELEMENTS OF FEED MECHANISMS

With the exception of a few processes the feed movement is usually rectilinear. It may be either dependent (turning, drilling) or independent (Milling) of the cutting movement. Feed may be either continuous (Turning, drilling milling etc) or intermittent (planing, slotting, grinding etc). In the latter case it is better to call it setting movement.

As far as the feed drive is concerned, the power requirements at the cutting edge are relatively low. Although the more complicated feed drive gear boxes often have a relatively low efficiency, usually it is not found difficult to control the power which has to be transmitted. However stiffness and rigidity against vibration of the driving elements are important, especially if the stroke of a feed or setting movement has to be within fine limits.

In the case of dependent feed movements, their speed must be kept within given limits of accuracy in relation to the speed of the cutting movement. By this not only a uniform surface pattern is maintained (e.g. in turning, drilling) but also certain pitch accuracy when this is required (e.g. thread cutting).

Feed mechanism consists of the one or more of the following separate elements:

1. Feed gearbox including Device for engaging the feed mechanism. Jaw clutch, sliding gear or friction clutches are arranged in the feed train where feed motion is transmitted to the working zone.
2. Rapid traverse mechanism
3. Feed reversing device
4. Periodic/intermittent motion mechanisms

4.2 TYPES OF FEED GEAR BOXES:

1. Change gears: Feed gear boxes with change gears on fixed position shafts are used in machine tools requiring infrequent change of feed.

2. Gear boxes with sliding gears: This type of gearboxes is more suitable for frequent feed changing and is therefore widely used. They can transmit high torque and can operate at high speed, without idly rotating mated gears. Draw back is that helical gears cannot be used.
3. **Draw key types:** In this type, a number of gears are mounted on a shaft. Any one of these can be connected with the shaft by a draw key that can be axially moved to engage one of the gears through a radial spring operated movement. However, a certain play between the movable draw key and key way in the shaft as well as slot is necessary and unavoidable. Moreover, load transmitting of draw key is also small. So this drive can transmit only relatively small torques.

There are some distant advantages of the system such as
(i) compact arrangement
(ii) feasibility of arranging 8-10 transmission
(iii) possibility of using helical gears with exact transmission-ratio
(iv) control of all the engagements of gears with single lever.

4. **Feed gearbox with gear cone and tumbler (Norton type):** It is the compact arrangement of the gear block and require only (K+2) numbers of gears for K transmission. There is a possibility of very fine stepping and only those gears are in mesh which is actually required to transmit the torque. Weakness of the system is lever carrying intermediate gear, due to this Norton type gear box are used only for low power transmission.

5. **Meander Type:** It differs from Norton type only that instead of only two gears meshing at a time, there are two gear blocks and all the gears are engaged and rotate continuously and motion is tapped from one of the gear out of the block. Gears are either heavily loaded or lightly loaded according to the tapping point gear. Zo (Figure 4.1) tap motion from one of the gear matching pair an shaft I & II.

---

**Fig 4.1 Gearing arrangement**

a) Draw-key type  
b) Norton type  
c) Meander type
6) **Clutch Type:** Greater power can be transmitted by clutch type in which axially controlled positive e.g. dog clutch or friction clutches are employed. Clutch type drives are particularly suitable for press elector gear boxes, because the clutch required to be operative for a particular output speed can be set ready for engagement while drive is still working at a previous output speed. At the moment of speed change, only the mechanism engaging the clutch is put into operation and the gear sets producing the desired transmission ratio are engaged. Scheme of a feed gearbox in heavy vertical boring machine is shown in the figure 4.2.

![Fig 4.2 Typical clutch type feed gearbox](image)

Feed motor 1 is a variable speed D.C. motor. Gears remain permanently engaged and feed is further transmitted either through E.M. clutch 2 or clutch 3. Thus depending on gear ratio two different steps of feed are obtained. For rapid traverse clutches 2 and 3 are disengaged and motion is transmitted from AC motor. Direction of feed to vertical or horizontal motion of tool-head is selected with the help of electromagnetic clutch 4. Electromagnetic clutch helps in quick engaging and disengaging of the motion by remote control.
4.3 BACKLASH-FREE GEARING ARRANGEMENT

Backlash-free gearing arrangements are used in the gearboxes fitted with variable speed motor only.

In fig.4.3 (a), a block of two gears, one mounted over the hub of the other with the eccentric sleeve for turning, are in mesh with the third gear. After the eccentric is turned for adjustment of the backlash, both the gears 2 and 3 are fastened together.

Fig 4.3 (b) depicts a composite helical gear (consisting of gear 6 and 8) mounted on shaft 4. Backlash is adjusted by placing half rings of appropriate thickness between these two gears.

In fig 4.3 (c), a spring washer 5 is located between gears 1 and 4 which displace gear 4 and turns gear 3 until its teeth come into engagement with the axially fixed gear 1 to eliminate backlash.

Fig 4.3 Backlash-free gearing arrangement
In fig 4.3 (d) backlash is taken up with compression springs 8 located in the recess made in gear 2 having one end with pins 3 fixed in gear 4, and with the other end against pins 7 fixed in gear 2. In this arrangement gear 4 turns with respect to gear 2, and backlash is eliminated.

Backlash free gearing arrangement increases loading on gears causing additional power consumption and early wear of the gear profile. So where rapid and feed motions are to be transmitted through same gear train, a system has to be provided to ensure backlash-free motion during feed and with backlash motion during rapid movement.

This could easily be achieved by preloading the pair of helical gears during feed drive and releasing during rapid motion. Preloading could be achieved by axially shifting the driving shaft making two helical pinions turn in apposite direction to eliminate backlash between driven gear and two meshing helical pinions. The arrangement shown in the fig.4.4 is for turning/c-axis feed drive of the table of vertical boring machine. Switching from turning drive to c-axis rotary feed drive is achieved by switching over from double pinion drive with power distribution to preloaded double pinion drive.

![Fig 4.4 Arrangement for backlash-free feed drive and with backlash drive](image-url)
4.4. TIMING BELT DRIVE

These drives are also called positive belt drives or toothed belt drives. Timing belt are endless flat belts with cross wise projections (teeth) on their inner surface that engage the teeth of the pulleys, transmitting motion without slippage. Timing belt drives combine the advantages of belt drives and those of chains and gears.

They can handle speeds from 5 to 50m/sec, offer a speed ratio of up to 12 and can transmit a power of up to 100KW with the 98% efficiency. The belts are made of rubber with a chloroprene base or polyurethane rubber. The load carrying layer is steel cable cored or glass fiber cored. For better wear resistance the teeth are sometimes covered with nylon or similar fabric.

The basic parameter of the drive is the module (m) or pitch (p)

\[ m = \frac{p}{\pi} \]

The teeth are of trapezoidal shape

Tooth height = 0.6 to 0.9

Narrowest part of tooth = (1 to 1.2)m

Max. Belt velocity = 25 meter/sec for m=2

= 30 meter/sec for m=3

= 38 meter/sec for m=4

= 40 meter/sec for m=5

Min. no. of teeth on smaller pulley = 16 to 20

(Greater the velocity more the no. of teeth)

No. of teeth on engagement along the arc of contact should be more than or equal to 6. If it comes less than 6 increase the center distance. The belt module is selected in accordance with the power to be transmitted.

<table>
<thead>
<tr>
<th>Power in K.W</th>
<th>Up to 0.4</th>
<th>0.4 to 3</th>
<th>3 to 5.5</th>
<th>5.5 to10</th>
<th>10 to 22</th>
<th>over 22</th>
</tr>
</thead>
<tbody>
<tr>
<td>Module in mm</td>
<td>2</td>
<td>3</td>
<td>3</td>
<td>4</td>
<td>4</td>
<td>5</td>
</tr>
<tr>
<td>Torque at smaller Pulley in N-m</td>
<td>2-4</td>
<td>49</td>
<td>190</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Specific tangential force</td>
<td>5</td>
<td>10</td>
<td>15</td>
<td>35</td>
<td>15</td>
<td>35</td>
</tr>
<tr>
<td>Module in mm</td>
<td>2</td>
<td>3</td>
<td>4</td>
<td>7</td>
<td>10</td>
<td></td>
</tr>
<tr>
<td>Pitch in inch</td>
<td>1/5”</td>
<td>3/8”</td>
<td>1/2”</td>
<td>7/8”</td>
<td>11/4”</td>
<td></td>
</tr>
<tr>
<td>As per DIN Standard</td>
<td>XL</td>
<td>L</td>
<td>H</td>
<td>XH</td>
<td>XXH</td>
<td></td>
</tr>
</tbody>
</table>

The allowable specific tangential force \( f \) is determined by the following equation.

\[ F = f_0 \times C_s \times C_{sr} \times C_{ip} \]

\( C_s \) is the service factor tabulated below.
**Table 4.2 service factor C_s**

<table>
<thead>
<tr>
<th>Types of drive Assembly</th>
<th>AC- Star delta star</th>
<th>DC- Shunt wound</th>
<th>AC- direct on line start</th>
<th>DC-Series &amp; compound wounded</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>1 shift</td>
<td>2 shift</td>
<td>3 shift</td>
<td>1 shift</td>
</tr>
<tr>
<td>(Light duty) e.g. Agitators Fan, pumps compressor</td>
<td>0.6</td>
<td>0.55</td>
<td>0.5</td>
<td>0.75</td>
</tr>
<tr>
<td>(Medium duty) e.g. Agitators of mixer of variable density, machine tools, Rotary compressors.</td>
<td>0.55</td>
<td>0.5</td>
<td>0.45</td>
<td>0.7</td>
</tr>
<tr>
<td>(Heavy duty) Elevators reciprocating pumps, hoist, heavy duty conveyor</td>
<td>0.5</td>
<td>0.45</td>
<td>0.4</td>
<td>0.65</td>
</tr>
</tbody>
</table>

C_s is the speed ratio factor and used only in case of step up drives i.e. where driven pulley runs at higher RPM.

**Table 4.3 Speed ratio factor C_s**

<table>
<thead>
<tr>
<th>n1/n2 = speed ratio</th>
<th>1 to 0.8</th>
<th>0.8 to 0.6</th>
<th>0.6 to 0.4</th>
<th>0.4 to 0.3</th>
<th>0.3 &amp; less</th>
</tr>
</thead>
<tbody>
<tr>
<td>C_s</td>
<td>1</td>
<td>0.95</td>
<td>9</td>
<td>0.85</td>
<td>0.8</td>
</tr>
</tbody>
</table>

C_i is factor for ideal pulley or guide-pulleys
- = 0.9 for one ideal pulley.
- = 0.8 for 2 ideal pulleys.

**Belt width**

The required belt width \( b = \frac{F}{f - A} \)

where \( A = \frac{qv^2}{g} \)

\( F \) is total tangential force = \( \frac{102PK}{v} \)

Where \( g = 9.81 \text{m/sec}^2 \)
- \( v \) = velocity of belt m/sec.
- \( q \) = mass of 1 m of a belt of 1cm width in kg/cm.

Dynamic load factor \( K = 1 \) for normal loading
- = 1.1 for moderately variable load
- = 1.25 cons variable load up to 200%
- = 1.5 shock and irregular load up to 300%
The force acting on the shafts from the drive is
\[ Q = (1 \text{ to } 1.2)F \]

Pitch diameter of pulley \[ d = \frac{pz}{\pi}, \text{ where } z = \text{ no. of teeth.} \]

Addendum circle diameter \[ da = d - 2\delta + k \]

Where \( \delta \) = distance from the bottom of the tooth spaces to the load carrying layer = 0.6 for XL and L series
\[ = 0.8 \text{ for Heavy series.} \]

\[ k = \text{belt deformation correction factor.} \]

\[ = \frac{2}{10^3} \lambda Z \frac{F}{b} \]

Where \( \lambda \) = yielding of the belt material mm/kgf.

| Table 4.4 Yielding of the belt \( \lambda \) |
|---|---|---|---|---|---|---|---|
| m | 1 | 1.5 | 2 | 3 | 4 | 5 | 7 | 10 |
| \( \lambda \) | 7 | 8 | 9 | 14 | 6 | 8 | 11 | 16 |

Selection of timing belt drives: Following steps should be followed in selecting the drive.

a) Speed ratio  
b) Max. belt velocity.  
c) Power to be transmitted  
d) Calculate total tangential force on the belt  
e) Selection of belt pitch/module  
f) Determine min. no. of teeth of pulley.  
g) Pulley selection  
h) Center distance and belt length.  
i) Calculate belt width after determining specific tangential force.  
j) Pulley bores  
k) Pulley flanges.

d0: Pitch circle diameter of pulley; z: Number of teeth on smaller pulley; 
n: Speed of smaller pulley; \( F_t \): Force required to adjust the tension; \( F_v \): Static tension in the belt; \( F_r \): Testing force for verification of tension; \( h \): Deflection of belt under force \( F_r \); h: Center distance;

METHOD OF VERIFYING TIMING BELT TENSION

\[ \text{Fig 4.5 Elements of timing belts} \]
4.5 RAPID TRAVERSE MECHANISM

The structure of rapid traverse mechanism is determined by the properties of following main elements:

a) Traversing element of feed drive.

b) Drive of the rapid traverse train.

c) Device for joining the rapid traverse and working feed train.

Cam mechanisms as the traversing element enable the rate of feed to be varied within a single cycle and reversal without need of rapid traverse train.

Rapid traverse train is powered either from a high-speed shaft running at constant speed at the beginning of the drive train of the machine or from a separate electric motor, a reversible motor if necessary.

Where rapid motor and feed motor are separate, both may work simultaneously. So some safety must be provided otherwise two separate sources of motion will damage the mechanism. Overrunning clutch, epicyclic system or differential is incorporated with the feed gearboxes for this purpose.

a) **Free wheel coupling or over running clutch:** Here motion from feed gearbox is transmitted only through overrunning clutch 1, while rapid motor 2 is directly connected with the driven unit. Overrunning clutch protect from transmitting motion of rapid motor to the feed gearbox.

![Fig.4.6 Protection thru over running clutch]

b) **Epicyclic system:** Motion from feed gearbox drives worm and worm-wheel. Worm-wheel acts as arm of epicyclic system on which two planetary gears are mounted. One planetary gear meshes with a sun gear transmitting feed motion to the driven unit. The other planetary gear meshes with a gear of rapid motor, receiving rapid motion from motor and transmitting it to the driven unit (whole system acts as simple reduction gear).

It means system works as epicyclic system when rapid motion is transmitted taking advantage of self-locking property of worm wheel (worm wheel can not drive worm and protect feed gear box from motion from rapid motor)

During feed motion speed of \( S_2 = \) speed of worm wheel

\[
= 1 - \frac{N_s}{N_{A1}} \times \frac{N_{A2}}{N_s}
\]

During rapid traverse speed of \( s_2 = \) speed of rapid motor

\[
= \frac{N_s}{N_{A1}} \times \frac{N_{A2}}{N_s}
\]
In some machines differential gears are used in place of over running clutch, as differentials are better suited to carry the high inertia loads that occur in reversing large masses in rapid traverse motion.

![Diagram](image)

**Fig.4.7 Epicyclic system for connecting rapid traverse mechanism**

When feed motor I is switched on, feed motion is transmitted to output shaft A through simple reduction gearing since the shaft of the planet gears is held stationary by self locking of worm gearing.

Transmission ratio (feed) = \[ \frac{Z_1}{Z_2} \times \frac{a}{b} \times \frac{c}{d} \times \frac{Z_3}{Z_4} \times \frac{Z_7}{Z_8} \]

When electric motor II of the rapid traverse drive is switched on,

Transmission ratio (rapid) = \[ \frac{Z_9}{Z_{10}} \times \frac{Z_{11}}{Z_{12}} \times \frac{2x}{Z_7} \times \frac{Z_8}{Z_8} \]

In this case bevel gear Z5 of the differential is fixed, the planet gears roll around it and the transmission ratio of the differential is 2.

In most of the machine tools braking devices are used in the feed drive to avoid over travel after disengaging rapid traverse if the traversing elements are not of the self braking type.

### 4.6 REVERSING MECHANISM:

In the operation of machine tools and other Equipment, reversal of at least certain motion is required regardless of the system used, a reversing device should satisfy following main requirement:

a) It should be capable to transmit variable torque.

b) Component of reversing device should not wear out with the inertia forces caused by reversal.

c) Loss of energy due to reversal should be small.

Application of electrical reversing is quite common in machine tools. But electrical or hydraulic reversal can not be applied in all the cases. In a mechanical system, reversing device should have a brake to absorb the kinetic energy and a clutch for acceleration in opposite direction. The clutches and brake of reversing mechanism should have controls with an inter-locking feature which makes it impossible to switch over from one clutch to the other without applying the brake with the help of single lever or push button.
SPUR AND HELICAL IDLER GEAR REVERSING MECHANISM:

Motion is reversed through one idler gear for one direction of rotation and by direct engagement of the driving and driven shafts or through two idler gears for the other direction of rotation. It is carried out by:

a) Sliding gears.
b) Sliding double clusters of identical gears.
c) Gears in constant mesh and engaged by clutches or sliding key.
d) Tumbler gears brought into engagement by swivelling them about a stationary axis.

Generally bushings are press fitted into idler gears, that Run freely on their axles to prolong their service life. Double cluster gear, sliding gears and cluster gears are mounted on spline shafts.

Jaw clutches and less frequently gear clutches are used to switch over the reversing unit in the feed mechanism of lathes, vertical turret lathes and milling machines. It is a good practice to make provision for lubricant supply from inside the clutches to reduce heating and wear of the friction surface from slippage.

**Fig. 4.8 Spur and helical idler gear reversing mechanisms**
BEVEL GEAR REVERSING DEVICES

Reversing devices consisting of bevel gears are used in working feed and rapid traverse. The main advantage is that it is equally applicable for any relative position of driving and driven shaft. But bevel gears are required of comparatively large over all size in transmitting high torque as compared to spur idler gear mechanism. Angular velocity of driving and driven shafts can be the same in both direction or different. For the later case the construction of mechanism is more complicated.

Idler gear and clutches are usually mounted on the reversible shaft. Rotation is reversed by the engagement of jaw or friction clutches or by shifting bevel cluster gears. Electromagnetic clutches are also being used.

Fig 4.9 Bevel gear reversing device

PLANETARY GEAR REVERSING MECHANISM

With the help of planetary gear any transmission ratio can be obtained along with the reversal of the speed. It is applied in the machine tools where a large reduction is required to affect working feeds in conjunction with rapid return traverse.
WORM GEARING REVERSING MECHANISM

Shaft I denote the driving shaft and II denote the driven reversible shaft. There is a third shaft having worm wheels of different hand than the worm gearing of driving shaft. Worms of both the gears are mounted a single shaft. The controls of clutches are so designed with an inter-locking feature that their simultaneous engagement is not possible.

NAPIER MOTION

It converts rotary motion of gear into rectilinear reciprocating motion or rotary reciprocating motion which may be of ram, slide table, cradle etc. later one is used in bevel gear generator.

The driving element of the mechanism is a bevel gear rotating at constant speed in one direction. The driven element is a pair of parallel gear racks connected at their ends by two half gears or two concentric segment gears connected at their ends by the same half gears. Meshing may be internal or external. At constant speed of the driving gear, the reversing mechanism provides uniform rectilinear reciprocating motion of the driven unit of the machine over the section corresponding to meshing of the gears with the racks and reversal in the sections of path corresponding to meshing of the gear with the end half gears.
4.7 MECHANISMS FOR PERIODIC (INTERMITTENT) MOTION

Certain machine tools like planers, shapers, slotters, grinders and other automatic machines require intermittent motion for feed and indexing. Intermittent motions are effected in modern machines by various types of (1) cam mechanisms (2) Mechanisms incorporating over running clutches (3) Ratchet gearing mechanisms (4) Geneva wheel mechanisms (5) Electric hydraulic and pneumatic mechanisms.

Over running clutches are generally applied where first driving link of the train has a reciprocating motion. Driving link is generally, comprised of a rack meshing with a pinion mounted on the shaft through an over running clutch. When this link moves in one direction, the over running clutch provides a rigid and positive kinematic-linkage between pinion and shaft. When the link (Rack) moves in opposite direction, the clutch is disengaged and the linkage is eliminated.

RATCHET MECHANISM

Ratchet gearing is especially convenient in cases when the time allotted to the displacement is limited. A pawl periodically turns a ratchet wheel with external or internal teeth through a definite angle. The ratchet wheel is linked kinematically to a power screw, which traverses the table, slide etc. Rotary periodic motions can also be accomplished by means of ratchet gearing.

In one full stroke (back and forth) of the pawl, the ratchet wheel can be turned through an angle as large as 80° or 100° but in most case angle does not exceed

The motion can be varied a) by changing the angle of swinging movement of the link that carries the pawl b) by covering ratchet wheel teeth over a part of the arc. c) by automatically lifting the pawl out of engagement during part of the stroke.

The ratchet mechanism of table feed of a shaper is shown in the figures 4.12a. The crank wheel 6 is fitted on the drive shaft giving motion for the stroke. The crank wheel has a T slot into which fits a crank pin 7 fixed with a nut. The crank pin can be moved along the slot by hand, thus changing the crank radius. A connecting rod 5 is linked to the pin 7 at one end and to a lever 2 at the other. This lever can turn on a shaft 4 on which the ratchet wheel 1 is keyed. A ratchet pawl 3 fitted to the lever 2 is pressed against the ratchet wheel by a spring so that its end engages one of the teeth.

The radius of lever 2 is greater than the maximum radius of the crank pin so that in turning the crank pin pawl makes two swings both sides for each revolution of the pin. When moved back, the pawl-end slides freely over the ratchet teeth without imparting any motion and engage the ratchet in direction of motion only. Motion of ratchet wheel is further transmitted to the useful table feed through lead screw. The feed is reversed by pulling the pawl by head and turning through 180°. If turned through 90° the pawl will remain in the disengaged position. Feed is adjusted by changing the crank pin radius, which change the angle of pawl swing and with it the number of ratchet wheel teeth engaged. Spring of pawl should be so adjusted, that it does not slide over the teeth of the ratchet wheel during forward motion.
Beside above-mentioned spring loaded pawls, throw over pawls are frequently used in feed system of planer, slotter etc. Rotation from feed motor is transmitted through worm gearing to the main shaft of feed gearbox. Drum, on which friction clutch (bend type) is mounted, is keyed on shaft. The friction clutch is linked thorough a pin to disk which carries the pawl of a ratchet mechanism.

Fig 4.12 Ratchet mechanisms
As the table reverses from the return to forward strokes a command is transmitted to the feed gearbox motor. The feed motion continue until the expanding strip of the friction clutch runs against a fixed stop and this engages the clutch This stops the feed motion. Then a trip dog operates the table reversing limit switch and also switches off the feed drive motor.

At the moment of table reversal from the forward to the return stroke, the feed mechanism is reset or “charged”. Feed motor rotate in the opposite direction. The pawl slips over the ratchet wheel teeth and the ratchet wheel remains stationary. Feed drive motor is switched off. Now feed mechanism is ready for a new feed motion.

Some machines have another mechanism for changing the feed rate. Pawl swings with a constant angle. The ratchet wheel is provided with a blind 1, which can be turned to close off some of the teeth of the ratchet wheel. Depending on the amount through which the blind is turned, the pawl will slip part of the time on the blind, engaging a different number of teeth.

**GENEVA WHEEL MECHANISM:**

Device with a constant angle of Geneva type gearing are mainly used in indexing Periodic rotation of turrets, spindle carrier and table in multiple spindle automatic machine tools etc. If a transmission with a variable ratio is introduced in the kinematic train between the Geneva wheel and the indexed components of the machine tool, the angle of rotation of the component is varied though the Geneva wheel rotates periodically through a constant angle. The driving elements are in the form of a lever, pinwheel gear or worm wheel carrying the pin tooth, which can be a roller mounted either directly on the pin or on the needle rollers. In some cases a ball bearing of suitable diameter, mounted on the pin, serves as the pin tooth. The driven element is made either as a solid part in the form of a wheel or disk or it is assembled of separate sectors or strips fastened to the part to be indexed in such a manner that spaces between them constitute the slots of the Geneva wheel.

![Geneva Wheel mechanism](image_url)  
*Fig 4.13 Geneva Wheel mechanism*
Chapter 5: MECHANISMS FOR RECTILINEAR MOTION

The most common mechanisms for converting rotary motion into linear motion are cam, cranks, eccentrics, and rack and pinion \worm rack and screw nut.

5.1 CAM MECHANISM

Most of the time cam mechanism is employed to transmit periodic motion. Rotating motion is converted into straight movements. In gearbox these cams are fitted on the shaft to actuate small reciprocating pump for lubrication of gears. But in some automatic machine tools cam mechanisms are employed to transmit main motion also. Rotated cam transmits motion through roller and the sector gear lever to the rack of the carriage, which produces the reciprocating motion according to the cam profile.

![Cylindrical cam converting rotary motion into rectilinear movement](image)

Fig 5.1 Cylindrical cam converting rotary motion into rectilinear movement

Cylindrical cams are also being utilized for converting rotating movement into rectilinear movement. Motion is transmitted according to the groove provided in the cylindrical cam. In the figure groove of the cylindrical cam guide a shifter for linear movement of cluster gear to the other position of meshing with another gear for change of speed.

5.2 CRANK MECHANISM:

A crank mechanism is used for converting translational motion into rotary motion and vice versa. It consists of a fixed shaft; a crank or arm attached to it a connecting rod and a sliding block. When the sliding block reaches the left or right hand dead Centre, the cranks and connecting rod lie in one line. A heavy flywheel mounted on the crankshaft ensures constant direction of rotation. The flywheel brings the crank out of the dead centre by force of inertia. The crank radius is always shorter than length of motion.
To ensure counter balances smoother operation crank throw weights attached to the web of crank. The shaft and crank pins are made from good quality carbon or alloy steel. Small crankshafts are forged, stamped or cast. Larger crankshafts are made composite i.e. the crank webs are made separately from the shaft and are then joined to it by pressing or screwing.

Fig. 5.2 Crank mechanism

Big end of the connecting rod incorporates the crankshaft and small end the piston or cross-head. If the big end embraces the crankshaft journal it is made split, but if it is attached to the projection pin of the crank it is made solid. The connecting rod is either stamped or cast from steel. Crankshaft bearings are commonly built up of two holes and provided with split bushes. The bearing bushes are made of cast iron or steel with babbit lining. If the connecting rod end is solid, a bronze or babbit lined steel bush is pressed into it.

In a crank shaft, wear affects its main journal and crank pins, which grow thinner, lose their original shape and develop taper and out of roundness. Lines, scores and some times cracks appear on the journals and pins. Shaft may bend, the seats of flywheel may wear and oil throwers deteriorate. Out of roundness and taper should not exceed 0.01 - 0.02 mm. The permissible run out of the shaft on the middle journal is 0.03-0.05 mm.

5.3 ECCENTRIC MECHANISM

It is widely used in machines and presses to convert rotary into reciprocating motion. The split type eccentric mechanism consists of a circular disk (eccentric) fitted to a shaft by key. The distance between the axes is the crank radius. A split clip fastened by bolts encloses the eccentric disk. The clips coupled to a connecting rod whose fork is hinged via pin to the slide, which receives the reciprocating motion.
In some cases eccentric mechanism may have two eccentrics. The internal eccentric mounted on a shaft is enclosed by an external eccentric. The external eccentric can be turned and fixed in any position, thus changing the eccentricity and the length of slide motion. Eccentrics are made from cast iron or carbon steel with the internal surface of the clip lined with babbit metal. The clearance between the eccentric and the clip is adjusted by placing shims between the mating surfaces of the clip. These shims are removed as the clip friction surface wears off. The connecting rod is adjusted in length by a coupling nut. This adjustment determines the final position of the slide at the end of the stroke.

When assembling the eccentric it must be ensured that the axis of the eccentric shaft is perpendicular to the slide guides. Lack of perpendicularity will bring about intensified wear of the slide guide and the friction surfaces of the eccentric and clip.

Fig 5.3 A typical split type eccentric mechanism
5.4 LINK GEAR

Link gear is also a sort of crank mechanism and widely applied in shaper and slotting machines. The ram slide, on which the carriage and tool are mounted, is hinged by means of link to slotted arm. The other end of the arm is linked to a pin on which it can swing freely. Arm swings because of reciprocation of a slide block in the slot of arm. Slide block is mounted in the guide way of link gear, which is rotated by a flywheel fixed on drive shaft. The position of the slide block on the link gear decides stroke length of ram slide. Slide block adjustment is affected with the help of a pair of bevel gear. Larger the radius of a circular arc described by the slide block more will be the strip length. Bevel gear pair drives the lead screw, which make the slide block move with the help of a box nut fitted in it. The slide block and the guide ways are ground and lapped together to obtain smooth movement without jamming.

Wear affects the slotted arm, sliding block, slide with pin, slide motion screw and nut and link gear. The surface of the slot of arm can be corrected by milling and scrapping if it is worn by more than 0.3 mm. Scrap the opposite surfaces of the slot to make them parallel with each other with the max permissible deviation of 0.03 mm. Axes of holes of both the end of slotted arm should be parallel with each other and to the slotted walls.

Guide ways of the link gear are corrected by scrapping. The slide block is repaired by first turning the pin and then scrapping the slide base. Squareness of pin to base is checked. Now scrap the side inclined surface of the slides. Ensure parallelism with a tolerance of 0.02 mm over the entire length of the surface. The slide should not rock when running in the gear slide ways.
5.5 RACK AND PINION DRIVE:

It is used in drilling machine to convert rotary motion into downward movement of the spindle with hand feed. Rack is fastened with the sleeve in which the spindle rotates. Rack and pinion drive is employed for transmitting considerable power for instance in main drive of planer and slotter.

Formulas for determining the torque

\[ a = \frac{v}{t_p} \quad \text{[m/s}^2\text{]} \]

\[ F_u = m.g + m.a \quad \text{(For lifting axle) [N]} \]

\[ F_u = m.g.\mu + m.a \quad \text{(For driving axle)[N]} \]

\[ T_{2\text{req.}} = \frac{F_u.d}{2000} \quad \text{[Nm]} \]

\[ T_{2\text{perm.}} = \frac{T_{2\text{table}}}{K_A.S_B.f_n} \quad \text{[Nm]} \]

Where \( T_{2\text{table}} \) is the torque as per table of the rack –pinion arrangement in the manufacturer’s catalogue.

The condition \( T_{2\text{perm.}} > T_{2\text{req.}} \) must be fulfilled.

Table 5.1 Load factor \( K_A \)

<table>
<thead>
<tr>
<th>Drive</th>
<th>Type of the load from machine to be driven</th>
<th>Uniform</th>
<th>Medium shocks</th>
<th>Heavy shocks</th>
</tr>
</thead>
<tbody>
<tr>
<td>Uniform</td>
<td>1.00</td>
<td>1.25</td>
<td>1.75</td>
<td></td>
</tr>
<tr>
<td>Light shocks</td>
<td>1.25</td>
<td>1.50</td>
<td>2.00</td>
<td></td>
</tr>
<tr>
<td>Medium shocks</td>
<td>1.50</td>
<td>1.75</td>
<td>2.25</td>
<td></td>
</tr>
</tbody>
</table>

Safety coefficient \( S_B \)

The safety coefficient should be allowed for according to experience (\( S_B = 1.1 + 1.4 \)).

Life- time factor \( f_n \)

Considering of the peripheral speed of the pinion, the lubrication and the stiffness of the pinion support.
Table 5.2 Life-time factor $f_n$

<table>
<thead>
<tr>
<th>Bearing distance*</th>
<th>1 X tooth width</th>
<th>2 X tooth width</th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Lubrication</strong></td>
<td>Contin. Daily Monthly</td>
<td>Contin. Daily Monthly</td>
<td></td>
</tr>
<tr>
<td>m/sec</td>
<td>m/min</td>
<td>0.85</td>
<td>0.95</td>
</tr>
<tr>
<td>0.5</td>
<td>30</td>
<td>0.95</td>
<td>1.10</td>
</tr>
<tr>
<td>1.0</td>
<td>60</td>
<td>1.00</td>
<td>1.20</td>
</tr>
<tr>
<td>1.5</td>
<td>90</td>
<td>1.10</td>
<td>1.50</td>
</tr>
<tr>
<td>2.0</td>
<td>120</td>
<td>1.25</td>
<td>1.90</td>
</tr>
<tr>
<td>3.0</td>
<td>180</td>
<td>1.35</td>
<td>1.95</td>
</tr>
<tr>
<td>5.0</td>
<td>300</td>
<td>1.40</td>
<td>1.98</td>
</tr>
</tbody>
</table>

* Distance from centre of pinion to centre of adjacent bearing.

To avoid backlash in the drive two pinion (one mounted on the other) are used so that one pinion tooth face is touching one side of tooth gap in the rack and the other pinion tooth is touching the other side on the tooth gap during feed movement. To achieve the same the two pinions are given motion in the opposite direction. For rapid motion the two pinions are given the rotation in the same direction to reduce loading. The movement of a lathe carriage is also due to travel of a pinion over a rack.

![Fig5.6 Showing the two pinions during rapid motion](image)

Advantage of the rack pinion drive is that it has a large transmission ratio. Rack travels a distance equal to the length of pinion pitch circle for each pinion revaluation. But transmission ratio is not uniform because errors in gearing affect the velocity of rack pinion. It is difficult to maintain uniform slow travel in the feed drive in the feed drive of a high precision machine tool with a pinion and rack. Rack and pinion drive has no self-braking properties. So it presents difficulties when it is used for vertical positioning movement. But this drive is some time used in parallel with allied screw drive.
Large rack and pinion is made of cast iron or of medium carbon steel tempered to hardness BHN 230-260. In the feed mechanism diameter of pinion is made as small as possible to reduce the torque on the pinion shaft and to shorten the reduction train of feed drive. So the pinion is, some times, made of alloy steel. Surface contact pressures frequently deform the teeth of unhardened racks. Induction hardening of teeth reduces this tendency. Long racks are made of two or more sections. The rack is located with dowel pins and secured to the corresponding part of the machine with screws.

**MOUNTING INSTRUCTION**

**Racks**

To make it possible to link our standard racks to from any desired length, the teeth are cut so that there is half a tooth gap at each end of the rack. The opposite diagram shows how rack 1 and rack 2 can be brought into the correct pitch position. Fitting aids with teeth cut in the opposite direction are available for linking helical-tooth systems. In order to ensure optimal fit we recommend the assembly of racks with predrilled mounting holes in angle-profile sections and to copy the holes on assembly.

**Gear and/or rack pairing**

The two pitch lines, in the case of gears the two shafts, must be parallel. The centre distances and centre position tolerances are in conformity with the quality requirements of DIN 3964. The mode of operation and the determination of the flank backlash can be individually adjusted by adapting one of the two drive elements accordingly. The following reference values for the flank backlash are applicable to hobbed gears:

- For small wheels and modules 1 to 2.5, 0.1mm
- For medium-sized wheels and modules 3 to 4, 0.2mm
- For large wheels and modules 5 to 8, 0.3mm

If high-load pairings are used, it is advisable to check the contact reflection under load.

---

![Assembly of rack segments](image-url)
5.6 WORM AND RACK DRIVE

The distinct features of worm and rack drives are:

a) Low transmission ratio.
b) Smooth motion
c) Lower efficiency than ordinary rack and pinion drive.

Types of worm rack drives:
1) The drive that has point contact between the worm thread and the rack teeth, and is used mostly for drives of auxiliary motion.
2) Worm and nut type drive with the axis of the worm arranged at an angle to the rack axis. The teeth of the rack resemble those of a worm wheel and the type of engagement is the same as in ordinary worm gearing.
3) Worm and nut type rack with a parallel arrangement of the worm and rack axes. The type of engagement is like that of a short lead screw and partly enveloping nut. In such arrangement the outside diameter of the gear in the worm drive must be less than the root diameter of the worm. In some machines special type of worm is used on which gear teeth have been cut in addition to the worm thread.

In the feed drive of heavy machine tools hydrostatic worm-rack drive popularly known as 'Johnson drive are used now a days. In this drive a short thread lead-screw type worm 9 is applied. The worm is not enclosed by the nut from all the side but from one side by the wormrack 5 fastened with the saddle 4. Hydrostatic principle is applied between worm and wormrack. Pressurised oil line 6 supply oil from hydraulic pump to the pockets in the area of meshing with the worm only so that oil consumption remains low. In the axial direction worm is controlled by the hydrostatic axial bearings 1. The worm-drive is mounted in an enclosed opening in the bed and motion is transmitted thru a gear teeth 8 in built with worm outer surface and meshing with a gear mounted on shaft 3.
Material used to make the worm and rack should have good antifriction properties because much sliding motion is involved in the operation of these drives. Worm is usually made of casehardening steel, which is then carburised and hardened. The rack is made of antifriction cast iron. In some cases bronze worm is used as it is easier to replace worm than the long rack.

5.7 LEAD SCREW AND NUT DRIVE

In general rotary motion of the screw is converted into translational movement of the nut. These are also mechanisms in which rotation of a screw is converted into translational movement of the same screw. In this case nut is fixed. Table motion of milling machine and compound slide movement of lathe machine is example of this type. In some machines lead screw is fixed at the two end of the slide and box nut rotate and impart translational motion to the lead screw. In machines like planners, lead screw is fixed to the column and it can have neither a translational nor a rotational motion. Box nut rotates as well as moves in straight line over the lead screw. Special feature of lead screw nut drive are given below.

a) When single start thread is used, slow motion can be obtained in the feed drive due to low transmission ratio of screw nut drive.

b) Exceptionally smooth and highly accurate motion is produced due to strictly constant transmission ration.

c) Lead screw nut drive is used for accurate positioning and vertical movement because of its self-braking capacity.

d) Due to sliding friction, lead screw nut drive has low efficiency so it can not used in main drive.

In order to ensure accurate operation and to reduce wear following points must be checked.

a) The run out of the lead screw thread in reference to the screw journal should be limited.

b) Axis of the lead screw bearings should be parallel to the corresponding ways.

c) Lead screw should have no axial run out in its rotation.

d) Lead screw and its nut should be strictly coaxial.

Standard for accuracy stipulates the maximum permissible pitch error, half angle of thread error out of roundness of the thread of pitch diameter and run out at major Dai.

Permissible pitch error $= \pm 0,12 \text{ mm}$

Cumulated pitch error for 20mm length $= \pm, 018 \text{ mm}$

$100\text{mm} \quad = \pm, 025 \text{ mm}$

$300\text{mm} \quad = \pm, 035 \text{ mm}$

in the whole length $= \pm,080 \text{ mm}$

Permissible error in half angle of thread

for pitch 3.5 $= \pm, 30'$

$6.10 \quad =\pm 25'$

$12.2m \quad = \pm 20'$

Lead screw is generally made of medium or high carbon steel with 0,15% to 0,5% lead. Lead nuts are usually made of tin bronze or antifriction cast iron. To make disengagement easier, sometimes split nuts are used in place of solid nut. Generally trapezoidal threads, having 30° angle, are used in lead screw. Some Times Square threads are also adopted.
Lead Screw Bearings

Bearings should be such that it do not allow excessive axial and radial run out of the screw which may lead to pitch error in the thread being cut. In most of the cases, screws are fixed axially only in one support with the help of double thrust bearing so that heating of the screw does not result in dangerous thermal stresses. For light and medium size screws, sliding bearings are more frequently used than ball or roller bearing because with bushing it is easier to obtain small run out and these have smaller over all size. In order to reduce bending deformation of lead screw various methods are applied.

a) Using bush bearing with a higher length to bore dia, ratio raises the rigidity of the bearings.

b) Additional supports are provided for longer lead screw. The support is a sleeve of sufficient length whose bore is an exact fit on the major diameter of screw thread. Sleeve travel together with nut.

c) For long and heavy lead screw hinged support is provided which is pushed away by saddle as it travels by or support which only partly envelop the screw.

Back Lash Adjustment:

After running for a certain period, threads of screw and nuts are worm out creating clearance between the threads of the two. In order to adjust the clearance in the thread and to eliminate back lash, lead nuts are made of two sections ane section is fixed stationary on the slide while the other can be adjusted axially. For clearance adjustment one of the following methods is applied:

Fig 5.9 Two section nut with a wedge for backlash adjustment

Wedge- Wedge 3 is fitted between two nuts. It may be of washer type. For adjustment loose screws 1 fixing two nuts and then screw 2 responsible for wedge movement is tightened. With the help of wedge movement two nuts are kept apart till their threads are completely in contact with the threads of lead screw.
**Set nut** - Axially free lead nut has threads on the outer dia on which set nut ‘a’ is mounted. By tightening the set nut lead nut is taken out till its thread face strikes with the thread of lead screw.

![Fig.5.10 Backlash adjustment by means of set nut](image)

**Springs** - For automatic adjustment of backlash smaller lead springs are fitted between two-lead nuts, which keep them apart according to the play between lead nuts and lead screw.

![Fig. 5.11 System for auto-adjustment of backlash with the help of springs](image)
**Hydraulic pressure:** Hydraulic pressure is used in milling machine adopted for climb cut milling. When the same machine is used for conventional milling or during rapid traverse movement, a valve connects cylinders at both ends of the nut, thereby removing load on the nut.

![Hydraulic system for backlash adjustment](image)

*Fig 5.12 Hydraulic system for backlash adjustment*

**MAINTENANCE OF LEAD SCREW NUT DRIVE**

Defects experienced in the lead screw box nut pair. Their causes and rectification are tabulated below.

<table>
<thead>
<tr>
<th>Defects experienced</th>
<th>Cause</th>
<th>Rectification</th>
</tr>
</thead>
</table>
| 1. In certain section, nut meshes only when considerable force is applied. | a) Screw is bent  
b) Scratches on the screw resulting from careless operation during assembly or disassembly of the unit. | a) Rectify bent in the screw press and check the beating.  
b) Clean the scratches. |
| 2. The lead nut moves over the screw up to a definite distance and then stops. | Pitches in the lead screw and nut are not equal because of improper manufacture. | Make a new lead nut or lead screw matching with the pitch of the other. |
| 3. During rapid traverse a lot of vibration and noise appears. | Absence of lubrication in the nut. | Provide lubrication. |
| 4. In the dismantled condition nut move freely on the screw of nut. But lot of force as required for fitting during the assembly of the two. | The axis of the screw is not the same as that. | Dismantle the nut and scrap the place where nut is fitted in order to have the same axis. |
5.8 HYDROSTATIC TRANSMISSION OF LEAD SCREW AND NUT

In heavy machine tool hydrostatic system is used for lead screw nut transmission. Here oil is fed under pressure between threads of lead screw and box nut through holes/capillaries 4. It reduces wear and increase efficiency of the transmission. Throttles 3 are therefore provided for hydraulic adjustment. Oil is supplied under desired pressure by the pump 1 thru pressure filter 2.

![Hydrostatic leadscrew-nut transmission](image-url)

*Fig. 5.13 Hydrostatic leadscrew-nut transmission*
Hydraulic transmission of lead screw and nut, practically do not have any clearance because due to hydraulic pressure clearance disappears and highest uniformity in motion can be obtained. Hydrostatic lead screw is preferred over ballscrew because of its higher damping property and backlash free motion.

Hydrostatic pockets are provided in the flanks (side surfaces) of the treads of thru nut where pressurized oil is supplied thru capillaries. Return oil is taken back from the top and bottom most surfaces of the threads. Oil is supplied thru axial deep holes from the face of the nut and thru short holes to the various hydrostatic pockets. Hydrostatic leadscrew-nut is having longer life. There is no chance of wear, so it could be called maintenance free system for the rectilinear motion.

*Fig. 5.14 Oil pockets for Hydrostatic leadscrew-nut*
Chapter 6: BALL SCREWS

6.1 INTRODUCTION

Ball Screw is a development of rolling guides over a lead screw. Their main applications are in feed drive and measuring devices. Positive characteristics of ball screws are ...

(i) Very good efficiency because of low friction of the order of 0.004 coefficient of friction.
(ii) No stick-slip motion.
(iii) Longer life because of least wears.
(iv) Can be preloaded to achieve backlash free motion.
(v) Outstanding spring rigidity i.e. it comes back to its original position.

Fig 6.1 View of internal arrangement of a ballnut
6.2 CONSTRUCTION OF BALL SCREW AND NUT

The ball screw assembly consists of the screw with threads of either semicircular or gothic profile; with similar internal threads on the nut with precision balls inter spread in between. Both the nut and screw should be made of such steel that hardness of 60 Rc could be achieved on working surface. So either low carbon alloyed steel nitrided up to 0.5mm depth or high carbon (1% carbon) chromium steels, fully hardened are used. Bearing steel and alloyed structural steel containing 1% Si and 1% Cr may also be used for nut. The material is case hardened. Normally screw shaft is manufactured from EN19 (induction hardened) nut from EN362 (carbonized) and balls from EN 31 steels of British Standards or equivalent.

The semi-circular profile has a ball radius to thread radius ratio of 0.95 to 0.97. The difference of radius is necessary to provide sufficient contact area.

But larger the difference in radii, larger will be the difference in velocities of their contacting surfaces. Also it will result an increase in clearance. So, the nuts used with semicircular profile are generally of the split type to facilitate for axial displacement for preloading purpose.

Intersecting arcs form the gothic profile. It has an additional advantage of permitting transmission with an interface. So that preloading can be achieved merely by using balls of slightly larger size. To permit unlimited linear motion, the balls must be continuously recirculated within the nut. Ball transfer may take place either internally by means of cross over raceways within the nut body or externally by means of ball return tubes.

**Internal re-circulation:** A special retainer which connects two adjacent threads of nut together, serves as the return channel, it forces the balls to circulate within a single thread pitch or after a no. of turns. In the case where ball returns after each turn, the nut has 3, 4, 5 or 6 circuits. In the later case, as the balls reach to the end of the nut thread, they are lifted off the load-carrying track by a transit piece and are returned via the axial cross over track back to the start of the thread. This system has an advantage of extra smooth running, short nut length and high load carrying capacity because of larger no. of balls, while the advantages of former system is the uniform, symmetrical distribution of the return mechanism ensuring a uniform load distribution and small overall dimensions.

*Fig.6.2 Arrangement for internal circulation in ball nut*
**Fig 6.3 View of internal circulation in ball nut**

**External Circulation** The balls recirculate in a closed loop. They enter the nut at the leading end roll through the track between the screw and nut carrying the load and at the trailing end of the nut are taken out and led through the return tube to re-enter at the leading end. Advantage of the system is that balls flow smoothly without jerk. But extra care is required to be taken to eliminate the chance of failure of the return tube.

**Fig. 6.4 View of external circulation in ball nut**
6.3 LOAD CARRYING CAPACITY

Load carrying capacity of ball screw is calculated from its static load capacity $C_0$ and dynamic load capacity $C$ which could be directly derived from pulling force. For static loading pulling force $Q$ can be expressed as

$$Q = P \cdot \sin \alpha \cdot \cos \lambda \cdot Z_r$$

Where $\alpha$ = angle of contact of ball from the place perpendicular to the axis of rotation.
$\lambda$ = Helix angle of thread
$Z_r$ = actually load balls i.e. total number of balls minus balls in the return passage.
$P$ = Static load acting on a single ball

$$\leq 2000 d_b^2 \text{ Newton}$$

Where $d_b$ = diameter of balls

Taking $\alpha = 45^\circ$ and $\cos \lambda = 1$
(Assuming small helix angle and $Z_r = 0.7$ considering that only 70% balls are loaded at a time.)

$$Q = 1000Z d_b^2 \text{ Newton}$$

For dynamic loading pulling force $Q_d = Q/K$
Where $K$ is a dynamic loading coefficient and determined by

$$K = C_d \frac{(60h.n.f)^{1/3}}{10^7}$$

Where $h$ = desired service life in hours
$n$ = average rpm of power screw
$f$ = number of loading cycle during one revolution of the screw, it my be taken as half the number of balls accommodated in a single thread.
$C_d$ = Coefficient generally taken as 0.9

$$SO \quad Q_d = \frac{Q}{K} = \frac{1000Z d_b^2}{k} \text{ Newton}$$

6.4 SELECTION OF BALL SCREW

The lead is a decisive factor affecting the drive torque required and the load carrying capacity that demands the maximum possible ball diameter. Further the following factors should be considered in the selection of the ball screw assembly required for a given application:

1. SUPPORTING METHODS

No special bearings are required for ball screw assemblies. Bearings as for plain spindles will suffice. At least one bearing must be capable of taking up axial thrust forces. Where high stiffness is required, the combined radial and thrust bearings should be used.

Since both the allowable axial load and the allowable rotational speed depend upon the method of mounting and supporting the length of ball screw, much care could be used if high accuracy is required. There are four methods of mounting ball screws.
a) **F-O Arrangements**: One end is radially supported and axially fixed while the other end is free. This arrangement is more suitable for small length of ball screw rotating at slow speed.

b) **S-S Arrangement**: Both ends are only radially supported. The arrangement is widely applied for medium speed rotating ball screw.

c) **F-S Arrangement**: One end is both radially supported and axially fixed while the other end is only radially supported. Where high accuracy is required and the screw is rotating at medium speed, this arrangement could be applied.
d) **F-F Arrangement**: - Both ends are radially supported and axially fixed. This arrangement is necessary for achieving high accuracy in high speed rotating ball screw.

![Fig 6.8 F-F Arrangement ball screw assembly](image)

2. **PERMISSIBLE AXIAL LOAD OR BUCKLING**

   The shaft diameter of ball screw must be calculated such that the screw does not buckle even if it is subjected to the maximum axial compression load.

   \[
   \text{Bucking load } P_c = \frac{m\pi^2EI}{l_k^2} \text{ or } = m\left(\frac{d^2}{l_k}\right)^2 \times 10^4
   \]

   Where \( l_k \) = Maximum distance of load application point from the least supported end (mm)
   
   \( E \) = Modules of longitudinal elasticity 2.1 x 10 kg/mm²
   
   \( I \) = Minimum movement of inertia of the ball screw

   \[
   = \frac{\pi}{64} d_i^4
   \]

   \( d_i \) = Root diameter of ball screw
   
   \( m \) = Coefficient of mounting method
   
   = 0, 25 for F-O arrangement
   
   = 1, 0 for S-S arrangement
   
   = 2, 0 for F-S arrangement
   
   = 4, 0 for F-F arrangement

   Over and above this value 50% safety factor should be applied. Allowable tension/compression load \( P \) in Kgf for ball screws of different diameter \( d \) in mm are given below in table 6.1.

   **Table 6.1 Allowable tension/compression load**

<table>
<thead>
<tr>
<th>( d )</th>
<th>16</th>
<th>18</th>
<th>20</th>
<th>25</th>
<th>28</th>
<th>32</th>
<th>36</th>
<th>40</th>
<th>45</th>
<th>50</th>
<th>55</th>
</tr>
</thead>
<tbody>
<tr>
<td>( P )</td>
<td>21</td>
<td>28</td>
<td>32</td>
<td>54</td>
<td>66</td>
<td>82</td>
<td>11</td>
<td>137</td>
<td>184</td>
<td>221</td>
<td>285</td>
</tr>
<tr>
<td>( 10^2 )</td>
<td>( 10^2 )</td>
<td>( 10^2 )</td>
<td>( 10^2 )</td>
<td>( 10^2 )</td>
<td>( 10^2 )</td>
<td>( 10^2 )</td>
<td>( 10^2 )</td>
<td>( 10^2 )</td>
<td>( 10^2 )</td>
<td>( 10^2 )</td>
<td></td>
</tr>
</tbody>
</table>

   If excessive compression loading is a concern, the easiest solution is to use the most rigid end mounting. The next step is to select a larger diameter ball screw.
3. **ALLOWABLE ROTATIONAL SPEED**

When the rotational speed is increased, ball screw approaches its natural frequency leading to resonance and failure.

This speed is called the critical speed.

\[ N_{cr} = \frac{Md_i}{l^2} \times X \]

Where \( N_{cr} \) is the critical speed:
- \( d_i \) = root diameter of ball screw
- \( l \) = Supported length between two end bearings
- \( X \) = Safety factor (0.8)
- \( M \) = Coefficient of mounting method
  - \( = 40 \times 10^6 \) for F-0 support
  - \( = 120 \times 10^6 \) for S-S support
  - \( = 180 \times 10^6 \) for F-S support
  - \( = 270 \times 10^6 \) for F-F support

The critical speed \( N_{cr} \) from the bending vibration is influenced by the axial load. A compressive load reduces the critical speed; a tensile load increases the critical speed. Thus, tension loading is always the best condition.

Another consideration is that a ball screw is also limited by the D.N. value irrespective of its mounting method.

\( D.N = 5000 \) to \( 7000 \)

Where \( D \) = outer diameter of ball screw

\( N = \) Revolution per minute

4. **RIGIDITY OF BALL SCREW ASSEMBLY**

For precision machines, the rigidity of component elements is an important factor to increase the positioning accuracy by the feed screw and to remain stiff against the cutting force to minimize lost motion.

The rigidity of the feed screw system under load is given by

\[ K = \frac{P}{e} \text{ (Kgf/mm)} \]

Where \( P = \) total axial load applied to feed screw system

\( e = \) elastic deformation of feed screw system

Rigidity of the assembly depends upon

1. Rigidity of screw shaft \( K_S \) (Kgf/mm)
2. Rigidity of nut \( K_N \) (Kgf/mm)
3. Rigidity of support bearing \( K_B \) (Kgf/mm)
4. Rigidity of nut bracket and bearing housing \( K_H \) (Kgf/mm)

\[
\frac{1}{K} = \frac{1}{K_S} + \frac{1}{K_N} + \frac{1}{K_B} + \frac{1}{K_H}
\]
i) Rigidity of screw shaft ($K_s$)

$$K_s = \frac{A.E}{L} \quad \text{in case where only one end support is fixed}$$

Where $A = \frac{\pi d_i^2}{4}$ (di is the root diameter of ball screw)

$E = \text{Modulus of long elasticity} = 2.1 \times 10^4 \quad \text{Kgf/mm}^2$

$L = \text{maximum distance of load application of point (nut) from the fixed support.}$

$$K_s = \frac{A.E.L}{a.b} \quad (\text{incase where both end supports are fixed.})$$

Where $L = \text{distance between two fixed a & b = distance of nut from two ends}$

i.e. $(a+b) = L$

$Ks$ is minimum when $a = b = L/2$

Then $(Ks) \text{ min} = \frac{4A.E}{L}$

ii) Rigidity of Nut ($K_N$):

- Rigidity of nut is mainly dependant on the type of ball screw nut and its internal design. This data is mostly given in the manufacturer's catalogue. In most cases rigidity of the screw shaft ($K_s$) is much lower than the rigidity of the nut ($K_N$).

iii) Rigidity of support bearing ($K_B$):

- It depends upon the type of bearing arrangement. Normally highest rigidity can be obtained with the arrangement with angular contact ball bearings with the same amount of preload.

iv) Rigidity of Nut bracket and bearing housing ($K_H$):

- The rigidity should be improved as much as possible by examining the mechanical design of bracketaries.

5. DESIGNED LIFE

The life of a ball screw is expressed by number of revolutions (L) or number of operating hours (Lh) at constant speed which 90% of a representative sample of identical ball screw assemblies will attain or exceed before the first sign of material fatigue appears. The life could be calculated with the help of basic dynamic load rating ($Ca$) and the axial load ($Fa$).

$$\text{Life (L)} = \left(\frac{Ca}{f_w.Fa}\right)^3.10^6$$

Where $f_w$ is a load factor:

- $1-1.2$ for smooth motion w/o shock
- $1,2-1.5$ for ordinary motion.
- $1,5-2$ for motion with shock

To determine life of a double nut, first calculate $Fa$ for both the nuts individually. Then calculate as per following formula:

$$\frac{1}{L} = \left[\left(\frac{1}{L_1}\right)^{\frac{10}{9}} + \left(\frac{1}{L_2}\right)^{\frac{10}{9}}\right]^{0.9}$$

If a ball screw which is under and excessive load of a great impact load, a local permanent deformation takes place between the raceways and balls.
There is a possibility of practical problem if the permanent deformation is approx. $10^{-4}$ times the ball diameter. The load, which occurs under that condition, is called the 'basic standard load rating' $C_{oa}$. Axial load $F_a$ should be much less than this rating.

$$F_a \leq \frac{C_{oa}}{f_s}$$

Where $f_s$ is a static safety factor

= 1-2 for normal working condition
= 2-3 for impact or vibrating working condition

Life in hours ($L_{1H}$) = $L/6\pi n$ hours, where $n$ is rpm

Life in running distance in Kilometers ($L_S$) = $L.P$ where "P" is lead in mm.

Recommended life for machine tool is 250 Kilometers

6. ACCURACY

Ball screws are divided into accuracy's classes according to DIN 69051. Following definitions must be known before finalizing the accuracy:

i) **Target point of accumulated basic lead:** - Generally same as the accumulated nominal lead. Incase the pretension lead is given to the ball screw shaft or the expansion is anticipated due to thrust load and temperature variation, target point is established to adjust accumulated lead for plus or minus.

ii) **Variation:** - The variation of the accumulated actual lead against the overall threaded length. It is also called total relative lead deviation ($e$)

iii) **Variation (300):** - The above deviation in random 300 mm within thread length ($e_{300}$)

iv) **Wobble:** - Maximum single lead deviation ($e_{2n}$)

v) **Axial plan:** - Precision ball screws are usually used in a preloaded condition and have no axial clearance. However if preloaded condition is not suitable to the application, axial play can be provided. Accuracy standard for play (axial clearance) are given below in the table 6.2.

<table>
<thead>
<tr>
<th>Table 6.2 Accuracy standard for play</th>
</tr>
</thead>
<tbody>
<tr>
<td>Play designation</td>
</tr>
<tr>
<td>Play</td>
</tr>
</tbody>
</table>

Precision ball screws are classified into six grade based on the variation in lead i.e. CO, C1, C2, C3, C4, C5 and C7 depending upon the variation in microns tabulated below in the table 6.3

<table>
<thead>
<tr>
<th>Table 6.3 Classification of precision ball screws</th>
</tr>
</thead>
<tbody>
<tr>
<td>Accuracy grade</td>
</tr>
<tr>
<td>Variation(300)</td>
</tr>
<tr>
<td>Wobble (e2n)</td>
</tr>
</tbody>
</table>
In addition for each accuracy grade mentioned above total variation (e) against the overall length is also limited to certain prescribed value
\[ e = e_{300} \frac{L_G}{300} \]
Where \( L_G \) is overall threaded length

Positioning accuracy of the system = Accumulated or single lead accuracy of ball screw.
Lost motion = (Positioning accuracy in forward direction) - (positioning accuracy in backward direction)
Total deformation of system \( \leq 1/2 \) (amount of lost motion)

**6.5 PRELOADING OF BALL SCREWS**

An axial play takes place because of a small clearance between the ball screw and nut. An axial load causes an elastic displacement at the contact points between the balls and the ball grooves, resulting in a backlash. Compressing or tensioning of the two nuts against each other eliminates the inherent axial play of the screw assembly, increases its stiffness enhances the accuracy with which the nut can be positioned. This is accomplished by providing a preload.

The preload should be as high as necessary and as low as possible in order to achieve optimum efficiency and longest life. Since the preloading effect is three times the preload, the proper preload is \( 1/3 \) as great as the maximum axial load.

The most common method of preloading is to insert a spacer between two nuts. Tensioning preload is increased by thickening the spacer and compression preload is increased by decreasing the thickness of the spacer.

The radial clearance / preload of the single nut running on the ball screw can be adjusted via a screw through a slot of about 0.1 mm width. The maximum preload is 5% of the dynamic load capacity. The stiffness of these two systems is roughly the same.

Fig 6.9 Typical Preloading arrangements
6.6 INSTALLATION AND MAINTENANCE OF BALL SCREW ASSEMBLY

The nut and bracket structure should be such that the ball screw is not under twisting load. For that parallelism between ball screw and linearly motion guide should also be maintained. When mounting a ball screw on a machine, care is to be taken that the nut is not separated from the screw. If this is unavoidable, use a sleeve, which is approximately 1 mm smaller than the root diameter of the ball screw, and remove the balls together with the nut without separating them. No balls should fall off and return tube is not damaged.

In case standard ball screw are valuable and additional is required for bearing journals and end fittings, following procedure should be adopted during machining of the same. Tape up vinyl packed nut at both ends so that the nut will not be moved when the shaft is rotated. Ensure that no chips get in. Ball screw should be fixed by centre as far as possible.

6.7 CARE & MAINTENANCE

Properly applied ball screw components change very little during its operating life. As a result there is no need for adjustment. Here are a few maintenance trips to be followed for the best performance.

a) **Lubrication**: Machine tool ball screws must be properly lubricated at all times. Either a good quality oil P.S. turbine oil of 38 to 90Cst is recommended or lithium soap based grease is preferred. Never use grease that contains graphite. It can built up within the nut and reduce internal clearance.

b) **Dirt protection**: In an environment which can involve dirt or foreign substances, it is necessary to hermetically seal the ball screw with a bellows or cover. A labyrinth seal can protect the nut only if there is no foreign substance expert dirt in the surrounding. The easiest way to clean a ball screw is to flush it with clean oil during operation.

c) **Preventive care**: Common problems and their solutions are listed below for better preventive care of ball screws.
   i) The cause of jamming is a “Key stoning” effect where one ball enters the return path improperly.
   ii) System inaccuracy and non-repeatability may be due to faulty assembly if growling or rambling noise is heard. If no noise, the problem may be in loose end bearings on the ball screw or in the control system.
   iii) Excessive drive torque may be due to faulty assembling of nut to screw ball nut to the machine to enable it to take its own alignment.
   iv) If backlash isolate the cause whether it is there due to loose end bearings or wear in the nut.
Chapter 7: SPINDLE UNIT

7.1 SPINDLE DESIGN AND SELECTION OF DRIVING ELEMENT

The operational movement of the spindle may be either purely rotational (Lathe machine, boring machine etc.) or rotational and axially sliding (drilling machine, boring machine some milling machines). Design of a spindle depends upon following factors.

a) Size of spindle, distance between supports, and size of the spindle bore.

b) Driving element (gear, pulley) and its mounting on the spindle.

c) Type of bearings.

d) Method of fastening of chuck, face plate etc or tool, which determines shape of the spindle end, e.g. threads or flange is provided at the spindle end of lathe machine. On the threaded end of the spindle, chuck or faceplate is fastened easily and quickly.

Flanged end of the spindle does not have threads; faceplate chuck is fastened with screw on the taper seating.

Driving element of the spindle depends upon speed of the spindle and amount of force to be transmitted. Gear transmission is simple and compact and can transmit considerable amount of torque. But due to pitch of the gear, high surface finish can not be obtained a grinding, jig boring machine, etc. In machines with varying cutting forces for example in milling machine, gear transmission reduces floating of spindle rotation and avoid dynamic over loading of gear box parts. So up to 150 to 200 rpm, teethed wheel transmission is applied.

Application of belt transmission requires a complex design. Better support is required to relieve the spindle from extra load. Pulley is to be mounted on (independent bearing) so that tension of the belt does not over load the spindle.

In spite of all above difficulties, belt transmission is preferred for high-speed spindle. Smooth rotation of the spindle provides highly finished machining. At dynamic loading, non-uniformity of spindle rotation is, however, higher because of less rigidity of belt transmission.

Rigidity of the belt transmission on be determined by, \[ C = \frac{R^3E}{L}A \]

C= rigidity of the transmission
R= radius of the loading pulley
A= area of cross section of belt
L= Length of the loading part of the belt
E= Modulus of elasticity

For flat as well as V belts, same formula can be adopted. To include loading characteristics, coefficient (k) for rotational force may be considered.

K=1 for smooth working (Lathe, drilling and grinding machine)
K=1.25 for sufficiently fluctuating load (Milling m/c gear hobber)
K=1.4 for stroke loading (Planer, slotter shaper)

With belt transmission, speed of 6000 rpm can be achieved when peripheral speed remains up to 60-100 m/sec. For Higher speed, as required in internal grinding machine, pneumatic (up to 10000 rpm) or elastic spindle (up to 15000 rpm) is used.
In high-speed electric spindle of internal grinding machine, motor 1 and spindle with grinding wheel 2 are balanced by weight 3 and 5. High precision bearings are preloaded and clearance is adjusted with the help of spring 4, and 6. Cooling fan 7 is provided to avoid overheating of the spindle.

![Grinding spindle in built with motor](image)

Fig. 7.1 Grinding spindle in built with motor

7.2 STIFFNESS

By providing independent bearings for driving element, the magnitude of transverse force acting on the spindle (due to action of belt tension or tooth pressure) can be reduced.

The displacement of the spindle caused by bending deformation may result in skewing in the bearing. This means that lack of stiffness may indirectly become the cause of additional pulsating transverse force and thus lead to vibration trouble. In order to understand the behaviour of a spindle, following force acting on it should be recognized.

1. Cutting force.
2. Pressure on pulley due to belt tension.
3. Weight of the spindle.
4. Disbalance of the rotating system of the spindle.
5. Centrifugal force acting on the balls of bearing.
6. Preloading.

Permissible deflection due to these forces should be within limit. Deflection should not be more than 1/3rd of the tolerance for run out at the spindle nose. Deflection and slope in other sections along the length of the spindle are limited by requirement of proper operation of the transmission and bearings.

Maximum permissible deflection $Y_{\text{max}} \leq 0.002L$

where, $L$ is the distance between the supports.

Max. Permissible slope angle $Q_{\text{max}} = 0.0001\text{radians}$ i.e. toothed gearing operates normally if the angle by which the gear axes are out of parallel does not exceed 0.001 radian.

For spindle on which the rotor of an electric motor is mounted, the maximum deflection between the supports is limited by $Y_{\text{max}} \leq 0.01\delta$

Where $\delta$ is the average width of the air gap clearance between the rotor and the stator of the built in motor.

In this last case, in addition to other forces, the load due to unilateral magnetic attraction is also taken into consideration.
7.3 BALANCING AND VIBRATION BEHAVIOUR

In unbalanced high-speed spindles, centrifugal forces may lead to deformation and rough running. It is therefore necessary to avoid or eliminate any unmachined and irregular shaped cast components, unilateral keys and holes etc. Play between the spindle and the torque transmitting elements (Gear wheels, chain sprockets, pulleys) may contribute to irregular torque transmission, high stressing of the keys and rough running.

In order to know the vibration behaviour of a spindle, its natural frequency must be determined. Angular velocity should never be nearer to the natural frequency to avoid resonance.

Critical angular velocity \( W_{cr} = W_0 \frac{\sqrt{Y_{11}}}{Y_1} \)

Where \( Y_1 \) = Deflection due to dead weight of the spindle.
\( Y_{11} \) = Deflection when spindle is rotating with a random angular velocity of \( W_0 \)

The following condition is usually laid down to eliminate the danger or resonance.

\[ \frac{W_{cr} - W}{W} \geq 0.25 \text{ to } 0.3 \]

Where \( W \) is the maximum angular velocity of the spindle rotation.

7.4 REQUIREMENTS FROM SPINDLE BEARINGS

Working of the spindle depends upon type of bearing. The following specific requirements are made to the spindle bearings of machine tools.

1. Accuracy of guidance (radial and axial) of the spindle: Only small clearance is permitted in spindle bearings in conjunction with high rigidity of the bearings. Beating of the spindle of average size machine tools should be within the limit of 0.01-0.03mm. For precision machine tools beating should be within microns only.

2. Spindle bearings should have long life. Rolling bearings having upto 500-hr. normal life may be used for spindle assembly.

3. Adaptability to variable operating condition, in maximum machine-tools: The spindle bearings are subject to various loads in a wide range of speeds, and with frequent starting and stopping.

4. Simple and convenient assembly, adjustment and disassembly etc.

Both sliding and rolling friction bearings are used in spindle in spindle supports.
7.5 SLIDING FRICTION BEARING

Material, for sliding friction bearings of spindle is selected after considering following properties.

a) Wear resistance.
b) Heat conductivity.
c) Coefficient of conductivity.
d) Coefficient of linear expansion.

Most commonly used material for sliding bearings are given below.

**Cast Iron**
Cast iron possesses poor running in or brisk in properties. So surface of cast iron bearing sleeve and the hardened journal of the steel spindle should be carefully finished. At low peripheral velocities, Cast iron bearings are capable of withstanding pressure up to 200 or 300Kgf/Sq.cm.

**Bronze**
Due to high cost of bronzes, some time steel or C.I.back is lined with a thin layer. Up to 5M/sec velocity bronze bearing can withstand specific pressure of 200Kg/CM2.

**Babbits:**
Babbits are used in the form of bimetal bushing in large bearings. Babbit bearings have good running in properties, due to which they provide excellent service in operation with an unhardened journal.

Sliding bearings as spindle support include following types:

**Non-adjustable bearings**
Such bearings are very rarely used and only where no wear can be expected over a long period of service.
b) **Bearings with radial and axial clearance adjustment:**

In such cases, bushes with cylindrical bore and external taper are used, the adjustability being obtained by the provision of slots. It is to be done very carefully as the bush is pressed into the conical housing with the help of two adjusting nuts (one on each side), the segments are deformed and the original circular bore takes on triangular or other shapes according to the number of slots. It means bearing ought to be rescraped after each adjustment. It is also important to make certain that the bearing bush can not collapse on to the spindle and clamp it.

For this purpose, sometimes, separating screws or shims made of leather or wood are usually provided. It would appear advantageous to place open slot at the top of the bearing, so that lubricating oil can not escape. However, radial load must not be directed against the 5

![Fig. 7.2 Sliding bearings for spindle with clearance](image_url)
Threads in the bush and adjusting nuts for axial displacement of bush must be rectangular threads to avoid collapse of bush on spindle due to radial load. In the figure 'b' shown above, bearing bushes with tapered bore is used for adjusting the play by axial displacement of the bush on a tapered spindle. Taper on the spindle is kept small (1:30 to 1:10) to minimize danger of pressing the bush on to the spindle at the time of adjustment.

Fig. 7.3 Sliding bearing with multiple wedge arrangement

c) Sliding bearings with multiple wedge arrangement

Automatic play adjustment is possible in such types of bearings. Bearings shown in fig 7.3(a) ensure high accuracy of rotation because the spindle is cantered by hydrodynamic pressure developed in oil wedges at several zones around the circumference. These wedge-shaped oil pockets are generated by using bearing made up of several self-aligning segments spaced equally around the circumference.

“Hydraulic” bearing in which a loose segment is held against the spindle with min. possible play by means of piston and spring is shown in the fig. 7.3(b). Oil is supplied via non-return valve to the piston, which thus prevent the lifting of the segment and with it the spindle. Another pump supplies the pressure oil for the bearing adjustment and also the lubricating oil for the bearing.
In the fig. 7.3(c) multi-block arrangement with several segments made of elastic material is shown. With this self-adjustment of clearance is possible with high precision.

d) **Hydrostatic bearings:**
These have provisions for supplying oil at considerable pressure to several pockets in the bearing. The oil flow out through the clearance between shoulders on the spindle at the end of the bearing. Hydrostatic bearings can operate under fluid friction condition even at the slowest speed of rotation.

The geometry and nomenclature of a cylindrical journal bearing with n pads are illustrated in fig. For journal bearings the optimum value of design pressure ratio is \( \beta = 0.5 \) as for other hydrostatic bearings. Other values of \( \beta \) will reduce the minimum film thickness and may reduce the maximum load. The following equations form a basic for safe design of journal bearings with any number of recesses and the three principle forms of flow control (refer to fig and table).

Load: \( W = p_f A_c \bar{W} \), Where \( \bar{W} \) is a load factor, which normally varies from 0.30 to 0.6 a better guide is \( \bar{W} = \frac{\lambda'}{2} \)

\( \lambda' \) =Dimensionless stiffness parameter from table.

\( \lambda' = \) Value of \( \lambda \) for capillary control and \( \beta = 0.5 \),

\( A_c = D \) (L-a)

Concentric stiffness: \( \lambda = \frac{P_f A_c}{C} \bar{\lambda} \)

Where \( C = h_o \) = radial clearance.

Flow-rate: \( Q = \frac{P_f C^3}{n} n \beta \bar{B} \)

Where \( \bar{B} = \frac{\pi D}{6an} \)

is the flow factor for one of the n recesses.

---

*Fig. 7.4 Hydrostatic bearing for spindle*
\[
\gamma = \frac{na(L-a)}{\pi Db}
\]
is a circumferential flow factor. If the dimension ‘b’ is too small the value \( \gamma \) will be large and the bearing will be unstable.

The recommended geometry for a journal bearing:

\[
a = \frac{L}{4}, \quad L = D, \quad b = \frac{\pi D}{4n}
\]

Journal bearings, which operate at speed, should be optimized for minimum power dissipation and low temperature rise for the same reasons as given under the previous paragraph headed ‘Plane Hydrostatic Pad Design’. Values of viscosity and clearance should be selected so that:

\[
\frac{nN'}{P_f} \left( \frac{D}{C_D} \right)^2 = \frac{1}{4\pi} \left( \frac{n\beta B}{A_f} \right)^{\frac{1}{3}}
\]

where \( N' = \) rotational speed in rev/sec

\[
A_f = \left[ \frac{\text{(total area)}}{2} - \frac{3}{4} \text{(recess)} \right] / D^2
\]

Recess depth=20 X radial clearance. Maximum temperature rise may be calculated as for plane

![Fig.7.5 Typical hydrostatic journal bearing](image)

e) Air bearings:

Air bearing can operate with aerodynamic pressure at high speed of rotation, or they are designed as aerostatic supports with large surplus pressure of air supply. Feature of air bearings is their lower rigidity as compared to hydraulic bearing and lower friction losses. Both factors are due to the fact that the viscosity of air is only 1/2000 that of light oil of 32 CST.
7.6 BALL AND ROLLER BEARINGS IN SPINDLE SUPPORT

Ball and roller bearings have not only the advantage of simple lubrication requirements even at the high speeds, but they also provide the possibility of obtaining almost complete elimination of play. However, the damping capacity of ball and roller bearings is less and their sensitivity to impacts load is greater than those of plain bearings. In spindle support, always costly accurate bearing is used. Rolling Bearings used normally in spindle units are shown in the figure 7.6.

It is preferable to separate the support of radial and axial forces. But design is simplified, in some m/c tools, by the application of taper roller bearings, which have a high load carrying capacity under both axial and radial forces. Ball bearings are less sensitive to small alignment errors. But in taper roller bearing, it is difficult to obtain the required running accuracy as this depends upon the manufactured accuracy with which the axes of several tapers (outer race, inner race and rollers), axis of spindle and that of bearing-housing coincide or intersect at one point. If the stiffness of the spindle is so low that the “Back bending effect” of centre bearing comes into full effect and third bearing is used to increase the stiffness, there is a danger that the axes of the rollers and the races will be displaced relative to each other. Moreover, if the two taper roller bearings are too far apart from each other, the bearing play as originally adjusted, increases with any rise in temperature and resulting expansion of the spindle. This difficulty will not come if double taper roller bearing is there in the front of the spindle.
Fig. 7.7 shows the spindle unit. The play of bearings of spindle is adjusted by means of nuts, pressing against the thrust ring. The pressure exerted by the thrust ring will cause the inner race of first taper roller bearing to shift towards the inner race of second bearing.

At the same time, the outer race of second forces the outer race of the first bearing for to move in axial direction and thereby reduce the clearance between the rollers and races of bearings, ensuring more accurate rotation of spindle.

As regards the lubrication of taper roller bearings, it is important to ensure that the pumping action of the rollers arising from their axial inclination does not oppose the direction of the oil supply. It is best to feed the oil in at the smaller diameter so that it is pumped by taper rollers towards the larger diameter, from their brought back to the tank.

Fig. 7.7 Spindle assembly with double taper roller bearings at front
7.7 PRELOADING OF SPINDLE BEARINGS

Preloading of the bearing increases its accuracy. This eliminates clearance between the bearing rings and the balls or rollers and, in addition sets unrealistic deformation that improves the total rigidity of the spindle unit.

Fig. 7.8 Preloading in typical bearing unit for spindle

Taper roller bearings or angular contact bearings installed in pairs, are preloaded by adjustments made during assembly or with the help of adjusting nuts provided on the spindle.

Fig. 7.8 displays the spindle bearings with double row roller bearing and a pair of angular contact ball bearings for high accuracy applications. Angular contact bearings face each other and preloaded at both the inner and outer races with the help of two separate well ground spacers. Take your measurements using two identical gauge blocks arranged between the widths of the rings, which are obtained by grinding the faces. Permissible non-parallelism of the ring faces is 0.005mm.

A special design of preloaded rolling friction bearings has found application in the supports of high-speed spindle. The double row bearing shown in fig.7.9 has a split outer ring. When the two halves of this ring are forced together, clearance is eliminated and a preload is produced. It is preloaded during the manufacture and assembly of the bearing by special built in device with a third row of small balls separating the two main rows of balls.

Fig. 7.9 Angular contact bearings with split race
On detecting diametrical or face play in a spindle mounted in rolling contact bearings, adjust at first the front roller bearing. To adjust diametrical play, release lock screw, give a few turns to nut, which bears against the inner race of bearing, and shift this race into the taper Journal of the spindle. The race diameter will slightly increase, and the diametrical play will be reduced.

Thrust ball bearings are mostly used to take axial load. Two thrust bearings as close as possible to each other at one of the supports to avoid the effects of excessive thermal deformation when the unit heats during operation.

![Fig. 7.10 spindle assembly with thrust bearings](image)

### 7.8 SEAL UNITS FOR SPINDLE BEARINGS

Spindle bearing should be sealed to prevent contamination or leakage of the lubrication. The seal units, consisting of cup or U, leather, plastic or rubber packing pressed to the shaft by a garter spring and enclosed into a metallic case, are commonly used for this purpose. Such packing type seals should embrace the shaft tightly and be correctly secured. A 0.1mm thickness gauge blade should pass with difficulty between the packing and the shaft easy passage indicates wear. As shown in the fig. 7.11a, in packing type seal, rubber or leather lip packing is placed in metal housing and secured therein with spring washer. A spring exerts a constant and uniform pressure upon lip pressing the rotating spindle. The packing is treated with a special chemical compound to increase its wear resistance.

Labyrinth seals:- Labyrinth seal(fig.7.11b) having no friction surface may prove expedient for high speed spindles. A normal clearance in the radial direction is 0.3-0.6mm and 1.5-3.0mm in the axial direction.
Fig. 7.11c shows some seal units for vertical spindles and shafts consisting of felt rings in combination with retainer rings and also piston ring type rings. Felt collar should be fitted against the shaft journals with a moderate tightness. A very tightly fitted collar will cause excessive friction and intensive heating of the shaft journal and the bearings.

7.9 REPAIR OF SPINDLE

Spindle must meet very high requirement. Coaxiality of the mounting journals should be within 0.01mm. The permissible taper of the journals is 0.01mm and out of roundness 0.003-0.005mm. Tapered bore of the spindle should be concentric with the journals. Permissible run out is 0.01-0.02mm per 300mm. End run out of locating faces with reference to the axis of rotation should not exceed 0.006-0.008mm. Bearing journals should be machined to a finish of V9 and more.

For spindle, axis of rotation is most important. For this journal seating 1 and 2 are machined by grinding and polishing. Accuracy for non-circularity of the journal should be with in 0.01mm and non-cylindrically 0.003-0.005mm. The same accuracy is required for the surface 3. Taper bore 4 and 5 of should be concentric with the journals; allowable beating 0.001-0.002mm at 300mm length.
Journals under bearing and seating place of rotary part like gear etc are worn out first. The same is true for bore surface 4. Theses surfaces must be checked for scratches and score marks. Wear affects the spindle thread 5 and keyway.
Table 7.1 Requirements in accuracy of spindle of metal cutting machine tools.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Allowance for Machine in mm</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Normal accuracy m/c</td>
</tr>
<tr>
<td>Ovality and taper of bearing journals</td>
<td>Less than 50% of allowance in diameter.</td>
</tr>
<tr>
<td>Spindle ovality</td>
<td>----</td>
</tr>
<tr>
<td>Taper(on the length of 300mm)</td>
<td>----</td>
</tr>
<tr>
<td>Allowance on diametrical size of journal</td>
<td>----</td>
</tr>
<tr>
<td>Radial beating of seating surface(center-ring journal tapered bore)</td>
<td>Less than 5-10.</td>
</tr>
<tr>
<td>Facial beating at diameter 120-180mm.</td>
<td>Less than 6-8.</td>
</tr>
</tbody>
</table>

Spindle on which the diametrical wear of the journals amount to 0.01-0.02mm, are corrected by lapping. Spindle journals worn out by more than 0.02mm, are ground and then lapped to the repair size. This method is used when the size of appropriate holes in bearings or other part can be altered accordingly.

If that is not possible and wear is more than 0.1mm, repair is carried out by building up a layer of chromium and those with greater wear by metal spraying or vibration arc welding.
Chapter 8: FRAMES, HOUSINGS AND SLIDEWAYS

8.1 INTRODUCTION

Housing or body parts of the machines are structure on which the main units of machine tools are assembled. Beds, column, carriage and cross rails, gearbox housings etc. care in this category. Body housing may be divided into two categories.

1. I group: Stationary housing is bed, columns, cross rail head stock are mounted at different positions and remain clamped during machining operations.
2. II group: Movable housing body like table, cross slide chuck, tool post which moves on the guide ways of bed or column during machining operations.

8.2 STATIONARY HOUSING

1. Bed: the configuration of a bed (Box, column etc.) is determined primarily by the arrangement of the ways on it for various units of the machine tools.
2. By the weight, dimensions and length of the stroke of the main units part.
3. By the necessity of housing various machines inside the bed.
4. By the necessity of providing various opening aperture etc. in the bed walls for assembly disassembly, in section, adjustment and lubrication of various mechanisms of the machine.
5. By rigidity against vibration etc.

In selection of feed & depth of out, that are permissible for the required machining accuracy and surface finish and specified tool life, depend upon the rigidity of the whole housing compiler.

Ribs connecting the walls or cast into the walls greatly affect the rigidity. Cross section of beds and column may of various profiles shown in fig. 8.1

Fig.(a) shows profile for column of milling drilling and others machines fig.(b) shows opened profile for horizontal bed figure(c) when two walls are joined by ribs in semi opened condition. Removal of large amount of chip necessitates the profile cross section as shown in fig. (d)

In beds mostly cast iron is employed welded steel beds are also being used in some machines to some extent reinforced concrete are also being used for the purpose of bed.

![Fig 8.1 Cross-section of bed and column](attachment:image.png)
1. **Housings of speed and feed gear boxes**
   Gear boxes fastened with the bed or column of the machine have box shape. The size and position of the mechanisms of drive and control determine its sizes. In some machines bed and speed gearbox are caste into one body. Unless rigidity of the machine so demands, it is not recommended to use such method because repair of guide ways of bed will become difficult. In machines like horizontal boring machine, gearboxes move on the bed or column guide. Proper clamping mechanisms are therefore provided to clamp it during machining operation. Housings of such gearboxes should have considerable rigidity and vibration resistant.

2. **Overhanging housing body**
   Cross rail in vertical boring machine, planner, arm in radial drilling machines are used to support tool base while knee of milling machine is employed for supporting job. Such overhanging housing body should have high rigidity while machining operation is carried out. Machining accuracy of machine tools depends upon the rigidity and vibration proof features of these parts. Rigidity of such moving parts as table, carriage or slide depends to a great extent upon the no. of joints or mating surfaces and their arrangement in respect to the acting forces. As a rule, the fewer the joints or mating surfaces, the more rigid the construction will be. Where operating conditions do not permit the reduction of no. of joints, constructing surfaces are enlarged in a direction approx. perpendiculars to that of the acting forces, to reduce the specific pressure and to make provision for firmly & reliability clamping parts which are to be stationary during operation.

   If a housing type part is traversed along vertical ways by a kinematics train which contains no self braking transmission, the part is balanced with a counter weight or spring to facilitate its setting and to prevent it from sliding down when it is unclamped.

### 8.3 THERMAL DEFORMATION OF BED AND OTHER HOUSING PARTS

Thermal deformation of the bed and column may be determined with the following formula.

\[ \nabla L = \sigma L t_a \]

- L = Length of the Bed
- \( \sigma \) = Coefficient of linear expansion = \( 10^{-5} \) /grad for cast iron
- \( t_a \) = Average temp.

Deformation \( X = \frac{\sigma L^2 \nabla t}{8H} \)

\( \nabla t \) = difference of aver. Temp. of the upper and lower surface of the bed

\( H \) = height of the bed.
8.4 SLIDE WAYS OF MACHINE TOOLS

Tool or the job travels in a straight line or a circle together with the units on which the tool or job is mounted. To guide the travel of the unit ways are provided. Either slide ways or antifriction ways are used. The principle characteristics of the ways are given below.
1. Accuracy of the travel
2. Durability
3. Rigidity.

Types of Sideways for Rectilinear Motion

Slide ways may be encompassed type (with apex upward) or encompassing type (with apex downward) as shown in fig. 8.2. Former type retains lubricants poorly than the later type. That is why encompassed types are generally used for comparatively a low traverse of a middle or a table advantages are that encompassed type are easy to manufacture and it has no tendency to accumulate type is easy to manufacture and it has no tendency to accumulate dirt and chips, so not necessary to provided shield or other protecting devices. Encompassing type slide ways are employed in M/c having units traversing at higher speed because it can retain lubricants in large quantity.

![Fig. 8.2 Principal types of slideways](image)

(i) **V ways**

V ways are difficult to manufacture but are capable of self adjustment. Clearance is automatically eliminated under the action of load. ‘V’ ways are made symmetrical as the load is directed vertically by the weight of the travelling unit and are more suitable for higher speed such as planners and grinding M/c. A deeper ‘V’ with small apex angle is preferred for precision machine-tools where loading is less.
In order to satisfy the requirement of unrestrained guidance, it is useful to combine one V with flat slide ways. Where two V slide are used it is practically improbable that all the four faces of the V are in perfect contact and carry the forces acting on them. Even in some M/c this arrangement is preferred because of reduced wear effect upon the working accuracy.

(ii) **Prismatic shaped ways (Triangular ways)**

Prismatic ways are used for symmetrical load and slow movement as in small revolving lathe machine larger face is located perpendicular to the direction of respective external load.

(iii) **Flat ways**

Flat ways are rectangular in shape so easy to manufacture. Closed type flat ways require devices for adjusting clearance and are used for high supporting forces for long slide ways. The location in horizontal and vertical directions is independent of each other because adjustment in one direction does not result in displacement in the other direction. Guiding surfaces for location in the secondary direction are suitably arranged as close together as possible in order to prevent skewing or jamming. Play adjustment or wear compensation is not automatic as in V ways so wedge or adjusting strips are provided.

(iv) **Dovetail ways**

These ways are distinguished for the small space they occupy and their comparatively simple clearance adjustment by means of a single taper or flat gib.

(v) **Cylindrical ways and circular ways**

Circular guide ways are simple to grind. They may be manufactured from light material and are easy to dismantle. Mostly such ways are used for revolving head in planomilling machine and several special attachments. Main shortcoming of the cylindrical ways is the low rigidity, which results from the fact that they are secured to the bed only at their ends. Besides, quite complex devices are required to adjust clearance in cylindrical ways.

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**Fig. 8.3 Circular or cylindrical guideways**

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**8.5 COMPENSATORS FOR ADJUSTING CLEARANCE**

Fitting the guides to the mating parts is laborious and precise operation special compensating devices are used to simplify the fitting of the friction surfaces and adjust the clearance if it is altered in the course of wear of the sliding surfaces. There are fixed compensating elements e.g. blocks and back plates and adjustable compensators which can be used to maintain the accuracy of a joint incorporating partly worn out members.

**BLOCKS OR BACK PLATES**

Both the ‘V’ and flat slides secure the vertical location only in the downward direction i.e under weight of the guided part. Forces and couple may occur which may tend to lift or tilt the moving parts. Therefore, holding strips or back plates are provided. These must
be carefully adjusted to ensure that the play in the vertical direction is not excessive. A groove is provided separating sliding faces from the fitting face so that fitters files or scraps only one or the other and knows exactly how far he has to go on each face.

In the fig.8.4a, a & b are the surfaces of the nonadjustable type back plate 1, which need scrapping or grinding according to the clearance in the other part. Adjustable plates 2 and 3 (Fig.8.4 a & b) which are fastened with saddle by pins, could be adjusted with the help of adjusting screws.

This type of designs have low rigidity, so are used only where frequent are not required. Compensating wear with the help of taper wedges (Fig.8.4c) has wider application. Taper of wedges may be from 1:40 to 1:100. Here surface of the saddle under the wedge has matching taper.

**Fig. 8.4 Methods for compensating wear of slideways**
ADJUSTABLE COMPENSATORS

Adjustable compensators may be of various constructions depending on the requirements.

(i) The clearances between the contracting horizontal surfaces, carrying the vertical pressure and were adjusted by flat gib 1 & 2 (fig.8.5a). The clearance between the vertical contacting surfaces, carrying the horizontal pressure $H_1$ or $H_2$ and constituting the guiding surfaces proper are adjusted by taper gib 3.

(ii) If the saddle or slide encompasses the contour of flat ways of the bed only on three sides (fig.8.5b), flat gib 1,2 secured to the slide ways are required. Scrapping will be required to compensate for were of horizontal faces. Some times to avoid scrapping thin shims are used. Flat gib of constant thickness is used here to adjust the clearance in the vertical contacting surfaces. In the course of its wear, this gib is adjusted forward by several screws 3.

(iii) A taper gib can also be used in the above case. Taper strip bears on its whole length and therefore, provides better conditions. The bearing area is independent of positional adjustment and with the usual taper of 1:40 or 1:100. Fine adjustments are possible. Care must be taken that the heavy mechanical advantage, provided by the wedge effect, does not create lateral stressing.

Fig.8.5 Adjusting clearances in flat ways
Tightening is to be done continuously taper strip must be prevented from undesirable longitudinal displacement, for instance under the effect of frictional forces, which may either loosen or tighten them. The provision of two adjusting screws (one at each end) or a stud with nut and lock nut may serve this purpose. Two adjusting strips are usually provided if it is necessary that transverse position of a slide is not effected by play adjustment. The longer the gib the less it is tapered. Various designs of regulating screws can be used for moving the gib in both directions. (fig.8.6b)

Shortcomings of adjustable taper gibs
(i) Taper gib increases the no. of joints between the mating parts.
(ii) They themselves possess low rigidity as a result; lower the rigidity of the unit in respect to compressive forces.

The effect of these shortcomings can be reduced by correctly locating the gibs and by providing means for clamping them tightly after making adjustments.
In clearance adjustment with a flat gib of constant thickness, the gib should be arranged so that its pressure is carried by the directly contacting faces of the ways. This means that the gib should be on the side or way opposite to the one to which load is applied. If a taper gib is employed, it may be on either side.

Fig.8.6 Adjusting clearances in dovetail ways & taper gib
The shape of dovetail slideways locates the guided parts horizontally and vertically both upward and downwards. Taper gib for long flat surfaces must be so designed that the wall thickness of either guiding or guided parts are not excessively weakened at the thicker end of the strip. In long dovetail slideways, the thickness of the strip may reach a value at which the stability of the guiding action is reduced because two points of action then lies too far away and even a small force may create a large tilting couple and create instability. Long slide ways are, therefore equipped with two taper wedges.

Better solution in the case of dovetail slide way will be to make use of wedged shape cross section in place of parallel cross section of the strip because the wedge effect of cross section counter acts the tilting couple.

Fitting and scraping should be very accurate so that the strip bears well on the sliding surfaces even after adjusting for wear.

### 8.6 RESTORING CLAMPING BLOCK AND WEDGE

Clamping blocks may be repaired by scraping grinding, planing or living checking periodicity the tightness of contact with the use of a marking compound. Measure the clearance with a 0.03-0.04 mm thickness feeler. A repaired block should fit snugly against the way, at the same time allowing table to move freely. The conjunction between the block and the bed should never be adjusted by slackening the screws.

Worn wedges are restored with the aid of straps welded to the thick portion of the wedge. The straps are then planed and fitted in place. This is done for the restoration of wedge for tables, saddles, carriages and slides. Laminated fabric of appropriate thickness is also a suitable material for linings, which are attached with carbonic or epoxide glue.

The surfaces of a repaired wedge should fit tightly against working surfaces of moving part and stationary face of the bed or addle. For this purpose, coat the ways of the part with a marking compound and install the wedge in place gently tapping its end. Then loosen the wedge by light blows on its thin end, using a soft drift pin, pull it out and scrap. A required wedge should be long enough to be able to compensate for the wear of the unit in the future.

For scraping wedge should be fixed in special fixtures (fig.8.7). The fixture consists of a place of angle iron 1. Stops 2 and 7 and last spring 5 stop 2 can be moved over the angle flange and secured in any place with bolt 3 and nut 4. After fitting the wedge on the flange snugly against spring 5, bring movable stop 2 to its other and fix it is place. Now tighten up nut 6 to clamp the wedge securely on the angle steel.

![Fig. 8.7 Fixture for fixing wedge for scraping](image-url)
With this device three-cornered scraper can be used which removes more metal than an ordinary flat scraper. Grinding may prove more efficient if done with the aid of a special fixture mounted on the magnetic table of surface grinding machine.

8.7 MATERIAL OF THE GUIDEWAYS

Wear on guideways is not always evenly by distributed over the full length of the fixed part. Its distribution depends upon the use of slideway by the shorter, guided parts in accordance with its relative position during various operations. This result in different wear conditions which together with the manufacturing inaccuracies and deformation of the M/c elements under load, reduce working accuracy of the machine. The following factors influence the wear of slide ways:
1. Material properties of the fixed and moving elements.
2. Surface condition of the slide ways.
3. Pressure exerted by the moving parts on the slide ways.
4. Dirt on the slide ways.

Table 8.1 Wear chart for sliding pair of different hardness

<table>
<thead>
<tr>
<th>Slide ways</th>
<th>Material</th>
<th>Wear</th>
<th>Material</th>
<th>Wear</th>
<th>Material</th>
<th>Wear</th>
<th>Material</th>
<th>Wear</th>
</tr>
</thead>
<tbody>
<tr>
<td>Upper</td>
<td>Unhardened CI HB 190</td>
<td>f₁ = 0.15</td>
<td>Unhardened CI HB 190</td>
<td>f₁ = 0.1</td>
<td>Hardened CI HRC 45</td>
<td>f₁ = 0.08</td>
<td>Hardened CI HRC 45</td>
<td>f₁ = 0.04</td>
</tr>
<tr>
<td>Lower</td>
<td>Unhardened CI HB 220</td>
<td>f₂ = 0.05</td>
<td>Hardened CI HRC 47</td>
<td>f₂ = 0.02</td>
<td>Unhardened CI HB 220</td>
<td>f₂ = 0.01</td>
<td>Hardened CI HRC 48</td>
<td>f₂ = 0.02</td>
</tr>
</tbody>
</table>

Table 8.2 Wear chart for sliding pair of different material

<table>
<thead>
<tr>
<th>Slide ways</th>
<th>Material</th>
<th>Wear</th>
<th>Material</th>
<th>Wear</th>
<th>Material</th>
<th>Wear</th>
<th>Material</th>
<th>Wear</th>
</tr>
</thead>
<tbody>
<tr>
<td>Upper</td>
<td>Unhardened CI HB 190</td>
<td>f₁ = 0.15</td>
<td>Bronze</td>
<td>f₁ = 0.1</td>
<td>Fiber longitudinal</td>
<td>f₁ = 0.15</td>
<td>Fiber cross</td>
<td>f₁ = 0.11</td>
</tr>
<tr>
<td>Lower</td>
<td>Unhardened CI HB 220</td>
<td>f₂ = 0.05</td>
<td>Unhardened CI HB 220</td>
<td>f₂ = 0.03</td>
<td>Unhardened CI HB 220</td>
<td>f₂ = 0.05</td>
<td>Unhardened CI HB 220</td>
<td>f₂ = 0.01</td>
</tr>
</tbody>
</table>

An experiment is conducted for sliding pressure of 10Kg/cm² pressure and velocity about 7M/min. The wear was measured after total length of about 7000 meter had been traversed. If the wear of upper (moving) element is f₁, that of lower (generally fixed) element f₂, then the total displacement in the direction normal to the sliding is the total wear(f₁ +f₂) is greatest if two elements of non-hardened cast iron on each other, the larger amount of wear f₁ accruing on the upper part. Total wear was less in the case of the sliding surface of the upper element being hardened and that of the lower are remaining unhardened.
The minimum wear occurs if both the surfaces are hardened. If the upper element is made of Bronze or plastic material, the wear of its sliding surface is little affected. However, the wear of lower surface (CI of 210-255BHN) is considerably less. The use of plastic sliding surface, of course, reduces considerably the danger of seizing.

Generally harder material is used for the stationary slide ways since their shape is copied in travel of the moving unit. Moreover, it is more difficult and expensive to repair the bed ways.

Steel slide ways, made of EN-18 hardened to 52-58 HRC by induction hardening, in the form of strips are either welded to steel bed, or they are secured by screws or bolts to a cast iron bed. Low carbon steels carburised and hardened upto 56-62RCH is also being used some times.

Laminated fabric strips are used in combination with cast iron for the slide ways of heavy machine tools where the comparatively low rigidity of the travelling unit leads to considerable non-uniformity in the distribution of pressure on the slideway surfaces. This may result in jamming with insufficient lubrication.

**DRAWBACKS OF LAMINATED FABRIC WAYS**

a) Low modulus of elasticity in comparison to that of steel.
b) Tendency to swell when they absorb all.
c) Low coefficient of thermal conductivity.

Seeing these drawbacks, it proves more advantageous to employ slideways with a thin polymeric coating applied by spraying, gluing on a thin film or some other method.

In certain case pads of Zinc alloy of Bronze are used on the slide ways. They possess good wear resistance, but are expensive and some times involve the use of critical materials.

**8.8 ATTACHED WAYS**

Now a day hardened steel attached ways are being widely used in m/c tools. After machining such ways undergo carburisation and hardening. Generally low carbon chromium steel (carburised 1.6-2 mm depth and hardened to HRC 56 TO 62) is used for ways of thickness of 4.60 mm and width 12-200 mm. Attached ways of longer length can be made in parts (of 600-1000mm length). In such cases end faces should be machined with proper accuracy so that there is no gap between two parts of the ways.

Then they are attached to the semi-finished surfaces of the cast bed in the form of strips. The design of fastening should be such that no damage is done to the working surface of the ways. It is better to fasten from underneath with screw. If design excludes this possibility, fastener should be used which does not violate the homogeneity of the working surface of the ways. After tightly screwing in the screws the heads are cut or broken off at the narrow neck and the remaining part of the screws are ground off flush with way surface. Ways secured with screws usually have an integral key, which relieves the screws of lateral loads, and considerably increase the transverse rigidity of the ways. Finally working surface of the ways is ground. The use of attached ways increases the service life of their friction surfaces 5-10 times.
Welded steel beds have the advantage over the cast iron beds in that, for the same rigidity they have higher wear resistance and require less metal.

8.9 GUIDEWAYS WITH PACKING PADS & GUIDEWAYS LININGS

In order to achieve maximum benefit and easiness during repair linings are recommended for slide ways. Guideways are ordinarily lined with laminated fabric (hylum or textolite), bronze or other copper alloys. Maximum effect can be achieved by application of anti friction material in heavy machine tools.

For feed movement (Working with lows sliding speed) and abrasive wear condition, hardened guides provide better wear resistance than guides padded with anti friction material. Pads from alloy or plastic material are attached with the guide of less length (on the slides of traversing unit). Matching unhardened guides of cast iron remains the same. Guides with bronze and particularly Cu-Zn alloy pads have low coefficient of friction with CI matching guides than CI-CI or textolite pair, but in abrasive condition it is less wear resistance. Guide with textolite & CI pair and bronze & CI pair has the same wear resistance properties. Pads of bronze or packing are used for the guides of carriage, saddle and tail stock of lathe machines, saddle of tool heads of vertical boring machines, saddles for column and table movement and spindle head of horizontal boring machine etc. Bronze is used for taper wedges and back plates, and also for the guideways of the unit working with extra load when inherent and contacted deformed layer of plastic may play the adverse role.

Zinc alloy (Al = 0-10% to 12%, Cu. 4 to 55%, Zn = remaining) is used for feed movement guides of heavy machine tools where guideways are well protected from dust and it is necessary to have uniform feed and increasingly accurate traverse (Table of plano-milling machine, gear hobbling machine and vertical guides of heavy machine tools.

Zinc alloy and textolite are used for guides of main motion working at high sliding speed (circular guides of vertical boring machine) textolite pads for flat circular guides.

Textile, bronze and zinc alloy are used for guides of main motion e.g. for guides of planer machine. Textolite pads are fixed on the guides with the help of epoxy glue. Generally pads of the length of 500-800 mm with the width of guide ways and 3-5 mm thickness (5 mm for heavy machine) are used. Textolite is mounted in such a way that their fibers remain parallel to the direction of sliding. For increasing strength of fixing, pads may be fastened with the pins of textolite or bronze after gluing the pads on the guide surface. Textolite pads having thickness more than 10 mm should be fastened with help of bronze-screw or textolite pins having fibers perpendicular to the plane of sliding.

Fig. 8.8 Fastening of attached ways
In the case of bronze and zinc alloy pads, guide surface is milled upto the depth of 2 mm in the middle of the surface where pad are to be fastened. It is done to increase the durability of the mounting. If the thickness of the pad is 5-6 mm, it can be fixed with the help of technological glue and by fastening by bronze screws only at the end of the pads.

These packing or pads are worn-out after working for some years. During repair it is required to be replaced by new one. In the case of combined guideways, it is necessary to determine the thickness of the packing to be fixed on the different guideways (V shaped & flat with the help of following formula).

\[ a = c \cos \alpha \]
\[ b = c \cos \beta \]

Surface finish of the guideways should be \( \nabla 6 \) for the heavy machine tools, \( \nabla 7 \) for medium type and \( \nabla 8 \) for precision machine tools. After fixing the pads it should be scraped according to the matching bed guideways. Then lubricating grooves are made up to the depth not less than 0.3 mm. Fastening screws must be well inside so that scratches do not came on the matching guide.

\[ \text{Fig. 8.9 Determining thickness of slideway linings} \]

\[ \text{Fig. 8.10 Slideways for circular movement} \]
8.10 SLIDE WAYS FOR CIRCULAR MOVEMENT

Circular ways are employed for the main cutting motion (Vertical boring machine) as well as for the work speed motion (Hobbing machine) and some times for handling motions.

Flat ways are used for circular movement of light and medium machine tools when a central bearing locates the axis to rotation of the table and flat circular ways carry only the vertical load.

If there is no central bearing a Vee type circular way, usually prism type ways is employed.

If the rotary table is of very large size, as in heavy vertical boring machines, two circular ways are used. This is done to reduce the vertical deformation of the table by providing intermediate surface in the ways.

Circular ways are partially relieved of their load in order to reduce the contact pressure and consequent wear by the provision of an additional adjustable antifriction thrust bearing and the delivery of lubrication under pressure between the working surfaces of the ways.

Oil is delivered under pressure between mating surfaces so as to produce an oil film over the full contact area. At higher speed liquid friction is created because of hydrodynamic effect. From pump through filter, oil is delivered into two types of pockets or grooves into open grooves for hydrodynamic pressure and closed groove for hydrostatic pressure at low speed. In open groove oil is delivered at less pressure in large quantity.

There is a throttle provided in the system, which helps in regulating hydrostatic pressure according to the weight of the job, being machined.

![Fig.8.11 Hydrodynamic lubrication system for circular guideways](image-url)
Load carrying capacity \( P = pF\alpha \)
\( p \) = Oil pressure
\( F \) = area of slide ways.
\( \alpha \) = Factor taking into consideration the drop in
Oil pressure in the clearance = 1/2 to 1/3

Higher will be the rigidity only when smaller is the clearance. Clearance depends on
micro regularities of the slide way surface with high quality scraping of 16 to 20 spots per
square inch (25 mm X 25 mm) area, min. design clearance of 15-25 microns can be
maintained.

For significant reduction of friction in guides, in recent years aerostatic pressure is
used. Air from the compressed air mains passes through a filter and pressure regulating valve
and enters at a pressure of 3 or 4 Kg/cm\(^2\) through aperture of small diameter (0.2 to 0.5 mm)

8.11 GUIDEWAYS PROTECTION

Open guideways are subjected to most intensive wear when metal dust and dirt mixed
with oil penetrates to the bed ways. A slight wear facilitates the ingress of dirt. The ways are
also worn down by chips pinched between the guide way surfaces. In practice three types of
protection are mostly used.

(A) The protection of slideways surfaces by providing covering devices.
(B) The insertion of a replaceable intermediate member between the guide ways surfaces.
(C) The use of seals and scrappers or wipers, which prevent the entry of foreign matter
   between the guiding and the guided surfaces.

Covering devices

Covering devices surround the otherwise open guiding elements and seal them off
hermetically. Such devices have to act telescopically, by becoming lengthened and shortened,
as required during the movement.

Covering belts may also be arranged above the guiding surfaces and these are either
belt in tension by spring-loaded rollers, or they cover the whole length of the fixed guiding
surface and are lifted off over the length of the moving part.

Protective device with belt tensioned from under the bed are recommended for
grinding machine and other machines where fine metals dust is produced. In the figure three
types of such devices are shown. In the first case belt is fastened on the face of the table and
moves on the rollers traversing from inside the bed.
In the second case tension in the belt is maintained by weight hanged on both side, which move in the ring or pipe.

In the third case two analogous schemes are shown in which belts are tensioned by load, which hangs the belt on both sides while their ends are clamped at same stationary part. Belt of 0.2-0.3mm thickness and 100mm width are generally used. It should be properly heat-treated. Some machine tools are protected by bellows made of cardboard covered with leather substitute.

Metallic strip covering

In some m/c a replaceable intermediate member is inserted between the guiding and guided surface. In drilling m/s a metallic strip is fitted on the guide of arm on which drilling hand is traversed. Metallic strip is clamped on the one end and fastened on the tensioning device at the other end. Whenever the strip is broken it can be easily replaced. The wear of C.I surface will be less as pressured acting on C.I surface is evenly distributed over a greater length.
Wipers

Dust may be kept off the ways by felt seals secured to the ends of slides, tables and other elements of the machine tool. However, the seals are rapidly contaminated with dirt and chips and act as a kind of lap which speeds up the attrition of the surface. This can be avoided by washing the felt at least once a week with pure kerosene. An unwashable felt seal is replaced.

The provision of simple felt seals is not available, as these are subject to wear and loss their effectiveness. It is better to combine felt seal with rubber seal or with some other suitable device.

Selecting the design of scrapper or wipers one should that these are simple easily replaceable and taps the ways for which these are provided. Some designs are given below.

The type shown in fig.8.14 is mostly used in Lathe machine. In the body of scrappers at the bottom side bronze piece 2 & 3 are fastened which throw out the chips. These bronze wipers are already machined and scraped according to guide ways. Front wiper bronze 2 act as knife and protect in a better way the entry of the chips dust etc. inside the scrapper. Three support of wipers of bronze are provided which increase thickness of the scrapper (not less than 5mm) to have sufficient rigidity under the real scrapper, just at the saddle, combined sealing is placed. Combined sealing is made from oil resistant rubber 4 and felt 5 compressed at the guide by plastic spring 6 through a small plastic piece 7. Spring is 3-4 mm compressed to create the pressure of 0.4 to 0.6 Kg/cm² between sealing and guides.

Fig.8.14 Wipers with bronze pads and scrappers
In order to ensure the required pressure between the seal and the guiding surface a leaf spring can be arranged between the cover strip and the seal itself. Mostly plastic spring is used (fig.8.15)

Fig. 8.15 Wiper with leaf spring
(ii)  Device shown in the fig.8.16 Provide excellent arrangement. Because here sealing is covered by clamping plate 3 fastened with screw 2 which can be very easily be taken out. It increases life of the guide by 1.5 times.

Fig.8.16 Wipers with clamping plate

(iii) Rubber sealing are much better than that of other material in cleaning the sliding surfaces. This along with felt is much effective. Sealing is mounted in rigid metallic cup type support from which rubber comes out about 3-4mm. Angle of the end of rubber piece touching bed guide is recommended to $5^\circ$-$10^\circ$. Seal should press guide with the force of .3 Kg at 1 cm length of sealing for sending oil resistant plastic rubber is generally used.

Fig. 8.17 Wipers with rubber sealing
Combined sealing of rubber and felt serves the purpose in the best manner in the guide of large machine tool. Rubber cleans best and felt absorbs abrasive materials, which usually go inside the lubricant. Width of sealing from felt should be more rather double than that of rubber. Pressure of felt sealing should be 0.4-0.6 Kg/cm². With this sealing wear is reduced to 5-6 times. Rubber and combined sealing require lubrication by automatic oil circulating system or through all gun etc. for lubrication of the saddle guide ways because oil resistant rubber cleans off the oil poured by oilier used in heavy lathe, boring and other machines.

Fig.8.18 Wipers with combined sealing

Bronze piece acting as chip cleaner is fitted before sealing to remove dirt and heavy metallic chips. Chip cleaners are made according to the shape of the guide prism. These chip cleaners protect sealing from chip getting inserted into sealing and damaging friction surface of guide. Light cast iron or dirt particle fallen on guides fill in micro-profile. Sliding chip cleaner does not clean the friction surface completely. So its effectiveness is not much. It is recommended for heavy m/c tools. In m/c with plastic or synthetic covered guide ways, wipers are not necessary.

For planer chip cleaners are used of the shape shown in the fig.8.19 It is better to provide spring loaded brass strip in addition to felt. These strips protect the seal.
8.12 GUIDE LUBRICATION

Proper lubrication reduces the chances of direct contact of the friction –surfaces and increases the average life of the rubbing part. Lathes and other revolving machine are provided with pump, which lubricates guide through distributors. It reduces wear effect by 20% for guides. For feed movement having lubrication system with low pressure (except hydrostatic and hydro-unloading system) cross lubrication grooves (perpendicular to the direction of movement) are recommended because longitudinal and diagonal grooves drastically depress hydrodynamic characteristics and increase frictional force which increasingly affects the oil exhaust. Longitudinal grooves is particularly harmful in the part of the width of the guide, where hydrodynamic pressure has its maximum value and is less harmful if groove is shifted at the edge of guide.
In planner it is advisable to provide support of (increasing pressure of) oil layer during lubrication of guide of the table. For this bed does not have flat base between ends of the guides, oil pressure is maintained because of the locking of oil with both ends of the base from which oil arrive in the hollow portion of the cross groove. Fig.8.21 shows the sealing pads fitted for this purpose. Part of the oil is delivered through hole in table and screw, with which sealing is fastened, in the space between face of the table and sealing. In the machine having flat base between edges of the guides, for building oil pressure, appropriate movement the thick flow of oil should remove chips and dust fallen on the bed guide.

![Fig. 8.21 Sealing pads at the end of slideways](image)

**Table 8.3 Lubrication grooves on guides**

<table>
<thead>
<tr>
<th>Delivery of lubrication oil</th>
<th>Guide of stationary unit</th>
<th>Guide of traversing unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>From the sides of traversing unit (from the hole in the guide of face of traversing unit)</td>
<td>Without groove</td>
<td>With cross groove in each for delivery of oil</td>
</tr>
<tr>
<td>From the holes of the surface of stationary unit</td>
<td>With holes along length</td>
<td>With cross grooves joined by longitudinal groove at one side of the edge of the guide</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>l/b</th>
<th>10</th>
<th>20</th>
<th>30</th>
<th>40</th>
<th>b in mm</th>
<th>Upto 50</th>
<th>50 to 100</th>
<th>100 to 200</th>
<th>Above 200</th>
</tr>
</thead>
<tbody>
<tr>
<td>6-10</td>
<td>1-4</td>
<td>3-8</td>
<td>5-12</td>
<td>7-20</td>
<td>A in mm</td>
<td>10-20</td>
<td>20-30</td>
<td>30-35</td>
<td></td>
</tr>
</tbody>
</table>
Length of guide of stationary unit $L$ should exceed the length of guide of traversing unit $l$ plus stroke of movement

$$L \geq l + \text{length of movement}.$$  

8.13 REDUCING LOAD ON GUIDES BY SPRING LOADED ROLLER

Such devices convert a portion of sliding friction into rolling friction. Spring-loaded rollers are mounted on the moving unit and share part of its load. This type of reduction of load on the guide provides lowering of wear and minimizes force of friction.

Along with the popular construction of providing rollers with saddle moving on hardened guide, following design is advised. In this roller is having the arc shape along its width. Cylindrical rod serves as the rolling path. Rollers have possibilities of self-adjustment in axial direction.

In cylindrical and surface grinders following method of guiding unloading may be used. On the bed guides oil pockets are provided. No. of pockets depend on the length of bed. In these pockets lubricating rollers are supported by the springs. Rollers fastened in the axle can move up and down along with column bush, which is mounted on the above said spring. This movement of the supporting column bush is restricted by screw. Design of the support depends upon the shape of the pocket. Tension of the spring should be such that 50-60% of the weight of the table is taken by these supporting rollers. Because of these rollers also the uniform and movement of the table of grinding machine may be achieved by the hand wheel.

![Fig.8.22 Unloading devices for slideway system](image-url)
Chapter 9: LOW FRICTION SLIDEWAYS

9.1 INTRODUCTION

Conventional guideway materials such as cast iron to steel, cast-iron to cast iron etc causes stick slip motion at low velocity because of their negative friction characteristics while seeking the solution it is concluded that-

1) stick slip is eliminated when the friction velocity characteristics has a positive slope.

2) when the difference between the kinetic and static friction is small, the elastic energy stored in the driving system will also be small and hence stick slip will be minimized.

To achieve these characteristics several methods are being followed in guideways system now a day.

9.2 NON METALLIC SLIDEWAY LININGS

As friction coefficient is higher between cast iron – cast iron as well as between cast iron – steel pair, it was thought to apply synthetic lining on one of the slide pair. Plastic like nylon, Perspex on cast iron give a frictional pair better than cast iron – cast iron pair even though their friction velocity characteristics curve has negative slope. It is because the plastic has anticorrosive and conformability properties. Plastics like phenolics and epoxies give positive slope friction velocity characteristics in a fair range of velocities. The former has that in a low velocity regime and the later has that in a high velocity regime (1.5-2 m/min). PTFE (Polytetrafluoroethylene) has a positive slope friction velocity characteristic at all the operating velocities encountered in a machine tool. PTFE has a lamellar structure from chain to chain branching. This gives important characteristics to the plastic such as-

i) Anti-sticking tendency
ii) Thermal stability
iii) Non solubility and lowest dry coefficient of friction
iv) Slide way does not wear in contact with PTFE.

Fig. 9.1 Diagram for velocity-friction relationship
v) PTFE improves dynamic rigidity of the machine tool because of good damping characteristics.

vi) Because of visco-elastic nature of PTFE, friction coefficient reduces with increase in load while running at higher speed.

vii) In the running in period friction increases with time and it stabilizes after the transferred layer is formed. Wear rate also reduces on subsequent sliding to a steady value.

viii) When sliding is restarted after stoppage for a longer period, friction value was observed to be higher than the steady state value due to adsorption of gas vapours, air etc in the PTFE.

As the adhesion tendency of PTFE is low, matching slideway should be a smooth surface. Fixing hardened ground steel strips on matching slideway are therefore recommended for lower coefficient of friction.

Further investigations revealed that bronze filled PTFE composite a material provides a good compromise between friction and wear requirements when employed under dry conditions. Among these, two types of plastic material are most commonly employed.
i) Thermoplastic material – such as Turcite B, multifil 426, fluon VB 60, Fluon Vx-2 etc.
ii) Thermosetting plastic – Like Ferobestos, SKC-3 etc.

It has been observed that Turcite B, a PTFE based thermoplastic and SKC-3 a thermosetting plastic are capable of working without stick slip at feed rates down to a value of 0.01 mm/min which is a common requirement of CNC m/c. Also these plastics can readily embed the foreign matter eliminating the possible of abrasion of slideway interface thus exhibiting higher wear resistance. SKC-3 exhibit higher wear resistance than Turcite B. The wear factor of dry cast iron is 0.47, which is approximately two times as high as the wear factor of dry Turcite B which 0.13 mm per 1000 Hrs. When lubricated with contaminated oil, the wear factor for cast iron is found to be approximately ten times as high as the wear factor of Turcite B.

The excellent compressive strength - only 1% deformation at 70 Kg/cm$^2$ pressure provides high load carrying capabilities in case of multifill 426 and Turcite B. At frequencies other than resonant frequencies the vibration amplitude of cast iron-SKC3 pair at different velocities is much lower than that of cost-iron-Turcite B pair. This is an added advantage towards maintaining good surface finish on the component and hence SKC-3 is preferable to Turcite B as far as damping behaviour and higher wear resistance is concerned. Other details and method of application of the two plastic materials are described below.

**TURCITE B**

It is homogenous materials made of Teflon, bronze and wear-inhibitors. The Teflon content makes it a self-lubricating material while the bronze provides excellent wear resistance. The static and dynamic coefficient of friction falls between 0.03 and 0.3 depending upon unit loading, velocity, lubrication and run in. Turcite B slide way perform best under a load between 0 to 56 Kg/Cm$^2$ because of little difference between static and dynamic friction, advantage of an easy and rapid adjustment procedure with reduced feed back time is achieved. Turcite B sheet of 1.6 mm or 3.2 mm thickness is bonded to the bottom surface of the sliding way by using special adhesive such as araldite and is cured for 24 hours at room temperature. Before bonding, it should be ensured that the surface is clean. Machined surfaces finish between 32 and 125 rms. is recommended for proper bonding.
SKC-3

Thermosetting plastic SKC-3 consists of a two-component epoxy based synthetic resin containing high-grade fillers. With appropriate drive system it is possible to achieve minute increments of as little as a few microns from rest.

SKC-3 coating is applied either by injection or by spatula after machining narrow confining edges of 2-mm wide and 2.8-mm depth to produce 3-mm thick SKC-3 coating. Before applying SKC3, the surface should be cleaned with acetone, SKC-11 is applied to counter guide surface and allowed to dry. The skc-3 coated slideway is then made to rest with its confining edges on the counter guide surface to form coating. Setting time is 24 hours.

9.3 ANTIFRICTION GUIDE WAYS

For heavier slides antifriction guide way elements are preferred to achieve accurate positioning accuracy. Here sliding friction is replaced by rolling friction, which has least difference between starting and running friction. Thus the stick slip motion is eliminated. The coefficient of friction in this case is considerably low and varies from 0.001 to 0.004. As a result, rapid slide response can be achieved with the actuators of lower power. However roller-slide units provide less damping and have a lesser load carrying capacity than the conventional slide ways. To improve the load bearing properties hardened steel strips are used in the matching guides. Sometimes hardened steel strips hardened upto 60-62 Rch are fitted on the bed guideways opposite to the bearing having rolling elements.

Fig.9.2 Typical antifriction slideway assembly

The main advantages are-
(a) The low friction, which does not practically depend upon the speed of travel.
(b) Precision movement and uniform slow motion.
(c) Longer service life as the coefficient of friction in antifriction ways is 20 times less than that in slide ways.

Following types of antifriction guideways are mostly employed.
NEEDLE ROLLER AND FLAT CAGE ASSEMBLIES

Most common arrangements available are of M and V shaped. For light load applications needle roller cages are used while for heavy loading roller cages are used. M and V guideways are always used in pairs. The seating surfaces of the guide ways are preferably ground. The main drawback of anti friction ways includes the lagging behind of rolling elements from the traversed unit.

During movement of the table, cage with balls also move some time. It moves about half a distance of the table movement. Because of this, the length of the cage with balls is in general shorter than length of the table by half of its movement. So in such design, movement of the table should not be large. Otherwise cage along with ball will hang out of bed in one direction and balls may come out due to weight.

For instance, the linear velocity of the centre of rolling element is only half that of the moving member.

\[ l_c = l_s - \frac{l}{2} \]

Where 
- \( l_c \) = length of the cage
- \( l_s \) = length of the stationary guideways
- \( l \) = length of travel of moving member

So the roller cage should be in the middle of the guide way when the moving member is in the centre position. It is therefore necessary to provide facility for return or recirculation of the rolling members for long distance travel wherever possible.
Open antifriction ways using balls or rollers find application in case when the main load constitute the dead weight of the travelling unit and varies only slightly during the machining operation. Closed antifriction ways provides means for preloading and have a much higher rigidity. Preloading is accomplished by taper gibbs or adjustable flat gibbs.

In some guide ways, ball bearings are used as shown in the figure 9.4. For cylindrical or circular ways, balls fitted in the cage, as shown in fig.9.5 are used. Here several rows of the balls are fitted in cylindrical cage or separator between hardened sleeve and cylindrical guideways.

To eliminate principal draw back of antifriction ways of the lagging behind of the rolling members, recirculation of balls or rollers are used in various construction. For longer traverse of the table, such system is used in which balls are placed between guide of beds and ways of the table without any cage. During movement of the table, balls fall in special enclosure which guide the balls to return again on guide ways.

TYCHOWAYS

Recalculating guideways popularly known as ‘tychoways’ are used for all types of straight guideways where long paths, low friction, accurate guidance and low wear are required. These bearings can be fitted in any desired position but generally arranged in pairs on the opposing guide surface and are preloaded against each other to achieve zero clearance.

Though stiffness is increased in this way but damping capacity is still less, so chances of vibration and chatter may be there. It has been established by experiments that the rigidity of roller ways approaches that of slide ways and may even be 3-4 times higher if they are suitably preloaded by taper wedge etc. The rigidity of ball ways is only from 40 to 50% than that of the roller ways if ball and rollers are of the same diameter. The surface roughness of the ways affects rigidity of antifriction ways. It is therefore, necessary to resort to lapping in manufacturing or repairing antifriction ways of precision machine tools.
Some times, it is therefore recommended to use combined guideways, Tychoways on the upper side to reduce friction and PTFE liner on the lower side for high damping capacity.

![Fig.9.6 Sectional view of Tychoway](image1)

**LM GUIDES**

The rollers are in contact with guideway machined on casting of the machine, so require accurate form to be machined hardened and ground to have a smooth texture. To reduce the problem of machining an accurate form on the bed of machine, hardened steel rails with special guide forms may be fastened to the casting of the machine. Special blocks, a pair along each guide rail, with recirculating balls can move along the rail. The balls provide the rolling motion. There is a line contact since shape on the rail is mating form of the balls. These guideway sets are precision elements. Tolerance is as fine as ±10 µ.  

![Fig.9.7 Typical slideway assembly with tychoways](image2)
The LM guides are available in ball type as well as roller type configurations. These are available in different length, which makes them suitable for different types of machines. To obtain a higher positioning accuracy and smooth running of guiding system, accurate mounting and locating surfaces are required. The parallelism is checked using standard fixture and dial indicators. The allowable value varies from 6-20 \( \mu \). The guiding elements should be supported on both the sides against the locating surface.

Fig. 9.8 Sectional view of typical LM guideway block

Fig. 9.9 Typical assembly with LM guideways

**Maintenance and Care of LM Guideways**

Before installation of LM guideways, foreign material and flashes on the mounting surface of the guide surface should be removed with an oilstone. A thin coating of oil should be applied on the resting surface of the guides and the mounting surfaces. Care is to be taken to prevent damage to the lip seal during assembly.
A thin sheet of spring steel should be used between the guide surface and the bearing block. There should not be error in parallelism of the rails and variation in the height of mounting surface. The preload should be one third of the operating load for dynamic stiffness of the guide. LM guides are provided with a lubrication nipple mounted on the end face of the carriage. Guides are available with grease lubricated for the life or with oil lubrication.

**Load Calculation for LM Guideways**

\[
P_1 = \frac{W}{4} + \frac{F}{4} + \frac{F_{xa}}{2c} + \frac{F_{xb}}{2d}
\]

\[
P_2 = \frac{W}{4} + \frac{F}{4} + \frac{F_{xa}}{2c} - \frac{F_{xb}}{2d}
\]

\[
P_3 = \frac{W}{4} + \frac{F}{4} - \frac{F_{xa}}{2c} + \frac{F_{xb}}{2d}
\]

\[
P_4 = \frac{W}{4} + \frac{F}{4} - \frac{F_{xa}}{2c} - \frac{F_{xb}}{2d}
\]

\[
P_1 \sim P_2 = -\frac{Wxh}{2d} + \frac{Fx\ell}{2d}
\]

- **Constant velocity**

  \[
P_1 \sim P_4 = \frac{W}{4}
\]

- **Acceleration**
\[
P_1 = P_3 = \frac{W}{4} + \frac{1}{2} X \frac{W}{g} \frac{V}{t1} X \frac{\ell}{d}
\]
\[
P_2 = P_4 = \frac{W}{4} - \frac{1}{2} X \frac{W}{g} \frac{V}{t1} X \frac{\ell}{d}
\]

- **Deceleration**

\[
P_1 = P_3 = \frac{W}{4} - \frac{1}{2} X \frac{W}{g} \frac{V_c}{t3} X \frac{\ell}{d}
\]
\[
P_2 = P_4 = \frac{W}{4} + \frac{1}{2} X \frac{W}{g} \frac{V_c}{t1} X \frac{\ell}{d}
\]

**CALCULATION FOR ANTI FRICTION GUIDE WAYS**

The sliding friction is a function of the load and friction.

Friction force \( F = \frac{\mu W}{f_r} \)

Where \( \mu \) - Coefficient of friction = 0.001 for steel

= 0.0025 for cast-iron

\( f_r \) - Factor for equivalent radius of rolling element \( \frac{r}{k} \)

\( r \) - Radius of rolling element in cm

\( k \) = 1.4 for open type slideways

= 2.8 for closed type slideways

\( W = P_y + W_t + W_w \)

\( P_y \) = Vertical component of cutting force

\( W_t \) = Weight of sliding table

\( W_w \) = Weight of work piece

Pulling force \( P = F + P_x \) Where \( P_x \) is the axial component of cutting force

Load carrying capacity of the antifriction ways is determined from maximum allowable stress in contact zone.
a) **Ball bearing guide ways**

Maximum stress in contact zone, \( p_{\text{max}} = K_1 \sqrt[3]{\frac{P}{d^2}} \) (N/cm\(^2\))

Contact deformation \( \delta = K_2 \sqrt[3]{\frac{P^2}{d}} \) (micron)

Where \( P = \) Force acting on a single ball
and \( d = \) diameter of ball

\( K_1 = \) Coefficient For load = 22x10\(^3\) for steel guide

\( = 16x10^3 \) for cast-iron guide

\( K_2 = \) deflection coefficient = 0.9 for steel guide

\( = 1.2 \) for cast iron guide

b) **For roller guide ways.**

Maximum stress in contact zone, \( p_{\text{max}} = K_3 \sqrt{\frac{q}{d}} \) (N/cm\(^2\))

Contact deformation \( \delta = K_4 q \)

Where \( q = \) load per unit length of roller (N/mm)

\( k_3 = 870 \) for steel

\( = 680 \) for cast-iron

and \( K_4 = 0.06 \)

For determining loading on each ball or roller, non-uniformity of contacting surface and eccentric loading must be studied. Generally for accurate m/c tool allowable stress in case of roller bearing guide ways are taken as 10-15KN/Cm\(^2\).

Thus with the help of the above formulae, load per unit length of roller can be determined after selecting suitable roller diameter.

The number of rolling elements (Z) in a guide ways using rollers can be determined from the expression –

\( 16 \leq Z \leq \frac{q}{4} \)

for ball ways \( 16 \leq Z \leq \frac{P}{3\sqrt{d}} \) i.e. Number of balls or rollers should be more than 16
9.4 HYDROSTATIC SLIDEWAYS

In the hydrostatic system the surfaces are separated by a film of lubricants supplied under pressure to one or more recesses in the slide way. From the pump oil is delivered under pressure through flow control valves with a restriction into pockets made in the ways.

When the two matching slide ways are made to approach each other under the influence of an applied load the flow is forced through a smaller gap. This causes an increase in the recess pressure across the lands surrounding the recess build up to balance the applied load. The ability to resist variation of gap depends on type of flow control.

Load capacity

The maximum mean pressure of a plane pad equals $\frac{1}{3}$ of the supply pressure.

1. Maximum mean pressure on the projected area of journal bearing and opposed pad bearing equals $\frac{1}{4}$ the supply pressure.

2. Stiffness $\lambda = \frac{W}{h_0} = \frac{\text{load}}{t_d}$ Where $t_d$ is the design film thickness.

Flow controllers in order of increasing bearing stiffness are as follows ...

1. Laminar restrictor
2. Orifice
3. Constant flow (fixed displacement pump or constant flow control valve with pressure compensation)
4. Pressure sensing valve.
5.

Fig. 9.12 Flow restrictors (capillaries) for Hydrostatic bearings

(Here capillary tube being fine, oil needs to be highly filtered)
Depending upon various design requirements different shapes of capillaries are available. Hydraulic System must provide pocket pressure \( p_0 \) at the design bearing clearance. Flow from the bearing recess at the design condition \( Q_0 \) must be known. When \( p_f \) is supplied pressure on gauge.

For capillary of bore diameter \( r \) and length \( l_c \)

Capillary resistance \( R_c = \frac{8\eta l_c}{\pi r^4} \)

pressure on gauge.

Capillary flow \( Q_0 = \frac{p_f - p_0}{K_c \eta} \), where \( K_c = \frac{128l}{\pi d^4} \),

(Resistance is adjustable)

Where \( l \) is length and \( d \) is diameter of the orifice and \( \eta \) is dynamic viscosity of oil

\( \frac{l}{d} \) should be greater than 100 and Reynolds no. \( \text{Re}<1000 \)

Orifice flow \( Q_o = \frac{\rho}{2(C_f A)^2} (p_f - p_0)^{\frac{1}{2}} \)

Where \( A = \text{cross sectional area of Orifice} \)
\( \rho = \text{Density} \)
\( C_f \) varies from 0.3 to 0.7 depending upon the Reynolds no. upto 100 \( \text{Re} \).

Bearing characteristics are also dependent on the pressure ratio \( \frac{p_0}{p_f} \). It is recommended to aim for 0.5 pressure ratio.

9.5 HYDROSTATIC GUIDING SYSTEM

According to Hagen poiseuille in case of laminar flow, the amount of oil \( Q \) passing through a parallel gap is given by following formula

\[
Q = \frac{\Delta p \cdot b \cdot s^3}{12\eta \cdot l} \quad \text{.........(i)}
\]

Where \( b = \text{gap width through which oil flow} \)
\( l = \text{length of gap} \)
\( s = \text{height of gap} \)
\( \eta = \text{Dynamic viscosity} \)
\( \nabla p = p_0 - p_1 \Delta p = p_0 - p_1 \)
As in most cases oil is allowed to escape out so $p_1=0$, then $\Delta p = p_0$

**Structure**

The hydrostatic grinding systems consist of pressure pockets whose boundaries are formed by gaps through which the pressurized oil drains away.

Equilibrium condition

$$ F = p_T \cdot A_T + \frac{p_T}{2} \cdot 2.1(b_1 + b_2) \quad (\text{ii}) $$

Where $A_T =$ pocket area

$$ F = p_T \cdot [A_T + l(b_1 + b_2)] \quad (\text{iii}) $$

$$ F = p_T \cdot A_w $$

Where $A_w =$ effective area.

Height of gaps resulting from equ. (i)

$$ S = \sqrt[3]{\frac{12\eta l}{2(b_1 + b_2)}} \cdot \sqrt[3]{\frac{Q}{T}} $$

$$ S = \text{Const.} \cdot \sqrt[3]{\frac{Q}{T}} $$

Rigidity C of guide ways = $dF/ds$.
Decisive for the determination of rigidity C of the guide ways is the flow.
Operating Principle
There can be two systems
a) One pump for each pocket
b) One pump for all the pockets.

a) One pump per pocket
Amount of oil = \( f(p_T) = \text{const.} \)
So height of gap \( s = \text{const.} \frac{1}{\sqrt[3]{p_T}} \)
Rigidity = \( \text{const.} \cdot p_T \cdot \frac{1}{\sqrt[3]{p_T}} \)

b) One pump for total system (all the pockets)
In this system a throttle is provided with each pocket to control the flow of oil in each pocket. For the laminar throttle \( \nabla p = \text{constant pressure drop across throttle} \)
Pressure drop across throttle = \( \nabla p = p_0 - p_T \)
\[ \frac{Q}{p_T} = \text{const.} \cdot \left( \frac{p_0}{p_T} - 1 \right) \]
From equ. 4
\[ S = \text{const.} \cdot \sqrt[3]{\frac{Q}{p_T}} = \text{Const.} \cdot \sqrt[3]{\frac{p_0}{p_T}} - 1 \]
Rigidity \( C = \text{const.} \cdot \frac{p_T^2}{p_0} \cdot \sqrt[3]{\left( \frac{p_0}{p_T} - 1 \right)^2} \)
This law is comparable with 'ohms law' of electricity. If 'R_T' is the oil pocket resistance then

\[ R_T = \frac{\Delta p}{Q} = \frac{12nl}{bh^3} \]

It is the resistance of throttle or capillary Installed in the hydraulic circuit.

\[ P_p - P_T = \Delta p = Q.R_K \]

While \( R_T = \frac{\Delta p}{Q} = \frac{P_T}{Q} \)

(as \( \Delta p = P_T - \text{atm. pressure} = P_T \))

\[ R_K = \frac{P_p - P_T}{Q} \]

\[ Q = \frac{P_T}{R_T} = \frac{P_p - P_T}{R_T} \]

Or \( \frac{P_p}{R_K + R_T} = \frac{P_T}{R_T} \)

Or \( \frac{P_K}{R_T} = P_T - P_T \)

Or \( P_T = P_T \left( 1 + \frac{R_K}{R_T} \right) = P_T \left( \frac{R_T + R_K}{R_T} \right) \)

Or \( P_T = \frac{P_T}{R_K + R_T} \)

Or \( P_K = P_T \frac{R_T}{R_K + R_T} \)

Actual area = \( A_R \)

Oil flow length = \( l \)

Flow \( Q = \frac{P_T bh^3}{12nl} \)

Load \( F = A_{eff}.P_T \)

Working area. \( A_R = lb = 2(B_e + h_e)l \)

Hydraulic pocket resistance \( R_T = \frac{P_T}{Q} = \frac{12nl}{bh^3} \)

Fig.9.15 Hydrostatic and Electrical circuits

Fig.9.16 Hydrostatic pocket & pressure
The calculations appear to be the same as used in electrical circuit where resistance is connected parallel to each other.

Where there are several guide ways supporting a saddle, two possibilities are there as shown in the figure above. Out of these two the (p) system having individual pump for each pocket has greater load carrying capacity. But the system with one pump with throttles for each pocket is more economical. In several cases for throttle, capillary tubes are used. Flow through capillary can be calculated by following formulas.
$$Q = \frac{\Delta p \Pi r_k^4}{8\eta l_k} \quad \text{And} \quad R_k = \frac{8\eta l_k}{\Pi r_k^2}$$

Where $r_k =$ Bore dia. of capillary tube.

$l_k =$ Length of capillary tube.

$\Delta p =$ less of oil pressure in passing through capillary.

From graph (1), (2) and (3) between oil gap 'h' and load coefficient 'p' it is clear that for the same load if oil viscosity is more 'h' (oil gap) will be more and so it is clear that average rigidity of the system is maximum when oil viscosity is minimum.

Depending upon the design, pressure could be applied on one side or both the sides of the slide ways.

**Fig.9.19 Hydrostatic guides without and with opposed pressure**

**Fig.9.20 Typical hydrostatic guide with opposed pressure**
9.6 AEROSTATIC GUIDEWAYS

The system is similar as that of hydrostatic except that here air viscosity is much less
\[ \eta = \frac{18}{1000} \text{ cst.} \] So gap 'h' is much less. Effectiveness of oil pocket could be worked out just by providing one air nozzle on each side.

Through nozzle flow rate should be \[ Q = \frac{3 \text{lit}}{\text{min}} \] for the pressure of 8 bar \[ p_s = \frac{8 \text{Kg}}{\text{cm}^2} \].

Here if we draw the graph between gap 'h' and load Coefficient 'p', it could be easily seen that rigidity is more then that in hydrostatic system. In the aerostatic bearings relative speeds could be much more because of low viscosity of air and low external friction for which especially high speed bearings with reduced loading are specified (identified).

**Advantage of Aerostatic System**
1. Extra ordinary reduction in friction.
2. Specified for highest speed.
3. Insignificant heating.
4. At very high speed lower temperature in the drive is ensured.
5. No chance of dirtiness through lubricant.
6. Simple design no need of labyrinth ring or oil seal.
7. Very simple.

**Disadvantage of Aerostatic system**
1. Low load-carrying capability.
2. Low damping.
4. Chances of corrosion in the material
5. Costly aerostatic unit.

**Air consumption and load carrying capability**

In order to increase load carrying capability special from of saddle plate(pockets) should be use

![Fig.9.21 Saddle pockets for aerostatic guideways](image)
When the gap is parallel to slide way, pressure flats down rapidly from nozzle towards circumference. This tendency is reduces when the gap is of concave form. So concave form gap enables for-higher load carrying capability. With the help of special design having elastic membrane, a shape of variable gap appears which allows the rigidity as high as infinitive. The upper surface of the membrane, where area is greater as compared to the bearing surface, develops the pressure such that \( P_{\text{gap max}} = P_{\text{chamber}} \).

Due to this the membrane takes a convex shape in the original sense such that it forms a gap of concave form. With proper balance between supply pressure and rigidity of membrane, the effective areas at the upper and bottom side of membrane become such that the bearing hosing does not displace with the increase of load to a wide range. Such condition may be named as fully compensated and rigidity appears to be infinitive.

There are two types of air bearing.
(a) Central recess bearing- air is fed from central orifice
(b) Annular recess bearing -air is fed from a ring of orifices.

There are two types of air bearing.

![Central and Annular recess air bearings](image)

**Central air bearings**

**Annular recess air bearing**

Less clearance -better stiffness

Load carried - \( W = c_t \pi r_0^2 p_t \)

\( p_t \) = supply pressure (gauge)

Stiffness \( k = 1.42W/h \) for pocketed orifice.

For central recess bearing \( \frac{d}{d_0} = \sqrt{\frac{h}{h_0}} \)

where \( d_0 \) = design orifice

\( h_0 \) = design clearance

\( d \) = actual dia orifice

\( h \) = actual clearance

For annular bearing with N pocketed orifice

\[ \frac{d}{d_0} = \sqrt{\frac{h}{h_0}} \cdot \frac{2}{\sqrt{N}} \]

For annular bearing with N unpocketed orifices

\[ d = \frac{d_0^2 h^2}{N h_0^3} \]

Stiffness = 0.95W/h
<table>
<thead>
<tr>
<th>Air pressure</th>
<th>(r_0/r_1=2)</th>
<th>(r_0/r_1=3)</th>
<th>(r_0/r_1=4)</th>
<th>(r_0/r_1=5)</th>
<th>(r_0/r_1=6)</th>
<th>(r_0/r_1=7)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1 bar</td>
<td>0.63mm</td>
<td>0.48</td>
<td>0.4</td>
<td>0.38</td>
<td>0.35</td>
<td>0.34</td>
</tr>
<tr>
<td>3 bar</td>
<td>0.85mm</td>
<td>0.7</td>
<td>0.6</td>
<td>0.57</td>
<td>0.54</td>
<td>0.5</td>
</tr>
<tr>
<td>5 bar</td>
<td>1.1mm</td>
<td>0.85</td>
<td>0.75</td>
<td>0.72</td>
<td>0.65</td>
<td>0.62</td>
</tr>
<tr>
<td>7 bar</td>
<td>1.25mm</td>
<td>0.98</td>
<td>0.83</td>
<td>0.75</td>
<td>0.7</td>
<td>0.68</td>
</tr>
</tbody>
</table>

Load carried \(c'_{L} \pi r_0^2 p_s\)

Where \(P_s=\) supply pressure of air

For annular type bearing load coefficient \(c'_{L} = 0.25\) for maximum stiffness and \(\frac{r_0}{r_1} = 2\)
Chapter 10: CLUTCHES AND COUPLINGS

10.1. CLASSIFICATION OF CLUTCHES AND COUPLINGS

The purpose of clutches and couplings is to transmit a torque between two coaxial shafts. Shafts and other components joined by couplings and clutches may be co-axial approximately co-axial or have axes inclined with respect to each other.

With respect to controllability, the couplings and clutches can be classified into the following groups.
1. Controllable clutches which can be manually engaged and disengaged
2. Self-acting clutches controlling the torque (safety clutch) direction of motion (over running clutches) speed (centrifugal clutches) etc.
3. Permanent couplings, which are continually engaged: rigid, compensating and flexible couplings.

Clutches are employed to engage and disengage shafts during their relative motion or at stand still. Shafts and other parts linked by clutches should be strictly collinear. Even slight misalignment deteriorates the performance of clutch and lead to their rapid failure.

Clutches are classified into following ways.
1. By kind of engagement - (a) friction
   (b) Jaw and toothed
   (c) Electro magnetic, fluid and power.
2. By form of friction surface - (a) Disk
    (b) Cone
    (c) Block
    (d) Band and spring
3. By mode of control - (a) Lever type
                         (b) Electromagnetic
                          (c) Pneumatic
                           (d) Hydraulic.

In all cases the actuating force is used to produce the clutch engaging pressure while spring pressure is used to disengage the clutch when the actuating force is released. Clutches that are manually controlled by lever and lever cam mechanisms, are applied for low and medium torques, when remote or automatic control is not needed. Clutch with hydraulic or pneumatic control mechanisms, are used for high torque when remote control is required. Hydraulically controlled clutches are not used at high speed. Electromagnetic controls are used for clutches when remote automatic and quick action controls are required.

Motion is transmitted from the manual control lever to a collar, which is shifted along the axis of the shaft. Most pressure mechanisms have operation levers. When the collar is shifting, its beveled surface engages one end of the operation levers whose other end applies pressure to the friction components of the clutch. Some time toggle system is also used for multiplying the engaging force. System of lock in the clutch is provided to eliminate the necessity of maintaining constant pressure on the operation levers after the clutch has been engaged. For very rapid frequency of operation such feature may be omitted.
In the clutch operation by air or hydraulic cylinders or by solenoids, pressure is applied to the mechanism through a sliding cam or spool on which is mounted a non-rotating sliding sleeve.

10.2. FRICTION CLUTCHES

The principle of a friction clutch is based on the developing of friction forces between the members of the clutch. Friction force can be regulated by varying the force with which the rubbing surfaces are pressed together. Smooth engagement of clutches is important to avoid high dynamic loads and noise in starting a machine. During the period of engagement (period of acceleration) and disengagement of the clutch, slipping may occur in the clutch. At steady motion, there is no slipping. Incidental slipping may occur at peak loads.

\[
\text{Acceleration period (t)} = \frac{I_e \times N_{\text{clutch}}}{F \times T_r}
\]

Where \( F \) is a factor equal to 308 in lb ft unit.
\( T_r \) is rated clutch torque
\( N_{\text{clutch}} \) = Speed of clutch in revolution per minutes
\( I_e \) = Effective inertia of load at prime mover
\( I_e = \text{Actual moment of inertia of load} \left( \frac{N_{\text{load}}}{N_{\text{clutch}}} \right)^2 \)
\( N_{\text{load}} = \text{Speed of load} \)

Heat Dissipation

A clutch will overheat and may burn out if it cannot dissipate heat it generates in slip. If clutch is constant.

Energy or heat input = \( h_p \times \% \text{slip} \)
\( h_p = \text{actual power input to clutch} \)
\( \% \text{Slip} = \frac{N_{\text{output}}}{N_{\text{input}}} \times 100 \)

It slip is frequent or cyclic

Heat input = 0.00017 \times I \times N \times f

Where \( I = \text{Total inertia of all driven parts} \)
\( N = (\text{Initial r.p.m})^2 - (\text{final r.p.m})^2 \)
\( f = \text{Number of applications per unit time} \)

POSITION OF CLUTCH

If clutch is installed just near the prime mover i.e. higher speed position, smaller clutch will be required. Generally clutches are used at this higher speed position. But in some cases clutch can be used at low speed position also, although larger costly clutches are needed but it will require less maintenance, down time and expenditure on spare parts. As heat dissipation corresponds to square of speed of the, at higher speed also too small clutch can not be used.
Upon instantaneous build up of the engaging force, the maximum torque in the drive reaches a value double that for which the clutch is adjusted. Therefore, provision should be made for gradual build of the engaging force.

**FRICTION MATERIAL OF CLUTCHES**

Friction material should satisfy following requirements:

a) A high coefficient of friction, retaining at wide range of velocities, temperature and loads.

b) Adequate mechanical and thermal strength.

c) Little wear and no scoring.

d) High heat conductivity for rapid dissipation of heat.

**Table 1 Friction material generally used**

<table>
<thead>
<tr>
<th>Material of friction surfaces</th>
<th>Operating condition</th>
<th>Coefficient of friction $\mu$</th>
<th>Unit pressure in kg/cm$^2$ (allowable)</th>
<th>Maximum operating in °C</th>
</tr>
</thead>
<tbody>
<tr>
<td>Hardened steel</td>
<td>Hardened steel</td>
<td>In oil</td>
<td>0.08</td>
<td>6-8</td>
</tr>
<tr>
<td>Cast-iron</td>
<td>C.I. or Steel</td>
<td>-do-</td>
<td>0.06</td>
<td>6-8</td>
</tr>
<tr>
<td>-Do-</td>
<td>Dry</td>
<td>0.15</td>
<td></td>
<td>2.5-4</td>
</tr>
<tr>
<td>Bronze</td>
<td>-do-</td>
<td>In oil</td>
<td>0.05</td>
<td>4</td>
</tr>
<tr>
<td>Pressed asbestos</td>
<td>-do-</td>
<td>Dry</td>
<td>0.3</td>
<td>2-3</td>
</tr>
<tr>
<td>Powder metal</td>
<td>-do-</td>
<td>Dry</td>
<td>0.4</td>
<td>3</td>
</tr>
<tr>
<td>-Do-</td>
<td>-do-</td>
<td>In oil</td>
<td>0.1</td>
<td>8</td>
</tr>
<tr>
<td>Steel</td>
<td>Sintered bronze</td>
<td>In oil</td>
<td>0.06 to 0.08</td>
<td>5 to 20</td>
</tr>
<tr>
<td>-Do-</td>
<td>-do-</td>
<td>Dry</td>
<td>0.12 to 0.18</td>
<td>5 to 8</td>
</tr>
<tr>
<td>-Do-</td>
<td>Ferrodo</td>
<td>Dry</td>
<td>0.25 to 0.45</td>
<td>2 to 2.5</td>
</tr>
<tr>
<td>-Do-</td>
<td>Fibre</td>
<td>Dry</td>
<td>0.2</td>
<td>3.5 to 4</td>
</tr>
<tr>
<td>Cermet</td>
<td>Hardened steel</td>
<td>Oil</td>
<td>0.1</td>
<td>6 to 8</td>
</tr>
<tr>
<td>-Do-</td>
<td></td>
<td>Dry</td>
<td>0.4</td>
<td>3 to 4</td>
</tr>
</tbody>
</table>

A special heat resistant material- Retinax- has been developed for operation under severe conditions (chiefly in brake units). The binding element of Retinax is phenol-formaldehyde resin modified by rosin, and with filler of baryta, asbestos and in especially severe cases, brass. Anti-seize compounds are also added to Retinax. Heating the working surface facilitates the formation of an efficient surface layer, which has high resistance to wear. The limiting values are: pressure $p \leq 60$ Kgf/cm$^2$ and velocity $v \leq 1000$ m/second, the temperature of the surface layer may be as high as $1000^\circ$C.

The friction elements of clutches that are to run in oil are made of steel with subsequent hardening or sulpho-cyaniding (simultaneous addition of nitrogen, carbon and sulphur to the surface of steel), as well as of combinations of materials: hardened steel on a cermet in the form of a lining.

The housings of clutches subject to sever thermal conditions can be made of bimetallic design with an aluminium base because thermal conductivity of aluminium is 5-8 times higher than that if cast-iron and has one-third its density. With respect to the shape of their friction surfaces, friction clutches can be classified as

- Disk clutches
- Cone clutches
- Radial clutches
DISK CLUTCHES

Disk clutches can have a large friction surface even with a small overall size. The force required for engagement is not very large because it consecutively applies pressure on the friction surfaces. Clutches may be of single disk and multiple disk design.

In the single disc clutch, the disk is keyed to on shaft and is compressed for engagement between two flanges keyed to the other shafts. These clutches are widely used in automobile for which clear-cut disengagement is a desirable feature.

A multiple-disk clutch consists of a housing, sleeve, a set of disks linked to the housing (outer disks), a set of disks linked to the sleeve (inner disks) and a pressure mechanism. In some case inner disks are directly mounted to the shaft in place of sleeve. In many cases there is no separate housing and the outer disks engage the body or hub member of pulley, gear or other component mounted on the shaft. Disks are linked to the housing and sleeve by means of parallel sided or involute spline joints. Frequently, the disks wear groove on the sides of the splines, which prevent smooth engagement by interfering with axial motion of the disks. For this reason, the surfaces of the splines must be properly hardened. Holes, slots and cuts in the disks make for better lubrication and cooling, reduce warping of heated disks and allow a smoother engagement. Clutches with frequent and continuous slipping employ disks with powder-metal facing and radial cuts. The spiral groove on the disks is intended to ensure their rapid cooling in a disengaged clutch and to facilitate the expulsion of grease on engagement. To reduce warping, the disks are provided with radial cuts. Generally not more than 8 to 12 disks are used to avoid chance of non-uniform pressure between them owing to friction on splines and to poor disengagement.

When the clutch is released, following clearances should be there between disks:
For metal disks- from 0,5 to 1 mm for single and two disks clutches and 0,2 to 0,5mm for multiple disks clutches.
For non-metallic disks- from 0,8 to 1.5mm, for single and two disk clutches and 0,5 to 1mm for multiple disks clutches.

It is more difficult to ensure proper disengagement on vertical shafts. Disks are sometimes made with different outside diameter in outer disk or different spline steps in inner disks providing steps on which they rest. When such a clutch is disengaged, each disk drops by gravity on to its own step.

The torque T that can be transmitted by a disk clutch can be determined with the help of the following equation:

\[ T = \frac{\mu}{f} n (r_o^2 - r_i^2) R_m n p \]

Where \( T \) = \( \frac{\text{power \ in \ KW}}{r.p.m} \)

Where \( r_o \) and \( r_i \) = outside and inside radii of angular

Friction surfaces ratio \( \frac{r_i}{r_o} \) usually ranges from 0,5 to 0,7

\( R_m = \frac{r_o + r_i}{2} \) mean radius of the frictional surfaces

\( \mu = \text{Friction coefficient} \)

\( n = \text{number of pair of friction surface} \)

\( p = \text{allowable unit pressure (lower will be pressure more number of disks)} \)

\( f = \text{margin coefficient for friction engagement} \)

The force required to press the disks together is \( F = \frac{f T}{R_m n \mu} \)
Disk disengagement can be improved by providing spreading springs or one half of the disks (for instance the inside disks) are not flat in free state (they are tapered or wavy). Such disks behave like springs themselves. To keep from making adjustments for wear too frequently, pressure mechanisms should have a definite amount of yielding. Sometimes disk springs are incorporated to increase the total amount of yielding in the system. Pressure mechanisms should provide for uniform distribution of the pressure both over the width of the friction surface and around the circumference.

The multi disk friction clutch shown in the figure 10.1 operates as follows. The gear wheel 2 takes rotation from shaft 20 through disk clutch though it is freely mounted on the shaft 20. Gear wheel 2 is fixed with the housing having slots on the rim to receive the lugs of friction disks 8(outer disks). Assembly is simplified by making the slots somewhat wider than the lugs and the diameter of the rim. The inner disks 9 are placed on three keys 22 fitted into the keyway of the shaft 20. A thrust collar 12 is fitted on the shaft 20 and can move along it. This collar is continually pressed to the right by spring 13. The other end face of collar 12 has several radial grooves, which house balls14. Theses balls are continually pressed against the internal surface of the engaging sleeve 15 by the action of the collar 12. Hence, when the sleeve 15 is moved to the left, its bevelled part will force the balls to move towards the shaft axis by rolling along the bevel of the adjusting bush 16. This will only be possible if the collar 12 shifts to the left, thrusting against the faces of the friction disk. Thus the friction clutch is engaged. It is stabilized in the engaged position by shifting the sleeves cylindrical portion. The force is adjusted by adjusting bush 16, which may be moved to the left, or right with the help of ring nut 19 screwed on the shafts 20. Arrester and lock is provided to fix the nut position.

![Fig.10.1 Friction disc clutch](image)

Disk clutches are periodically adjusted to achieve a tight contact without slipping and over heating in engaged position and free rotation in disengaged position. It should never be over tensioned otherwise it may result in slipping of disks overheating and wear in neutral position.
Mechanical clutches require means of adjustment to compensate for wear and to control the plate pressure. Tight adjustment increases the pressure and torque capacity but requires greater force on the operating lever. Since the pressure applied to the sliding sleeve lever or fork is an external force, there may be tendency to move shaft itself by force of reaction. So, positive means should be provided to prevent such movements.

Disk type pneumatic clutch utilize a built in piston, which may be in reality a piston or be a diaphragm or an expanding tube or bladder. The torque capacity of the air clutch may be varied over a wide range by varying the air pressure. Generally 6-8 kg/cm$^2$ pressure is required. In some cases even 1-2 kg/cm$^2$ may be used, where slipping conditions require a clutch relatively large in comparison with torque requirements to permit sufficient heat dissipation.

Pressure regulating values are used for varying air pressure supplied to the clutch. Air clutches require no mechanical adjustments, since the moving parts automatically take up any wear on the friction surface. Air pressure must be maintained continuously while the clutch is engaged. Air supply should be as free from moisture as possible. A slight amount of oil is desirable as lubricant for piston type design. For remote control operation, solenoid operated valves are used.

![Fig.10.2 hydraulically operated disc clutch](image-url)
HYDRAULICALLY OPERATED DISC CLUTCH

As shown in the figure 10.2, hydraulically controlled disc clutch contains moveable cylinder 1 and a bush 3 (keyed on the driving shaft) acting as piston. When oil is delivered to the corresponding hole in the shaft to the right side of the cylinder, the latter moves in to that side and compress set of disks 6 and 7 forcing them to the end plate 11. As a result driver body 12 starts rotating. When the oil enters to the left side it engages that side of the clutch by means of flange 2 and plate 5 resulting in rotation of guide body 4, releasing the former one. The piston is sealed by rings 9. When there is no oil in the clutch the cylinder is held in its intermediate neutral position by springs 8 and 10.

![Fig.10.3 Typical assembly of hydraulically operated disc clutch](image)

Typical assembly of an improved version of hydraulically operated friction disc clutch is shown in fig.10.3. The pressure force of piston is transmitted over an axial bearing to the running discs. In order to achieve higher working load, internal lubrication of discs are foreseen. Wear of discs are compensated through piston stroke. So adjustments of discs are not necessary. The clutch does not have rotating working cylinder, so supply of pressurized oil is simple and reliable. Also the pressurized oil does not have centrifugal force. The clutch becomes pressure free very fast, so it could be engaged and disengaged quickly.
ELECTROMAGNETIC FRICTION CLUTCH

In this type of clutch the friction surfaces are compressed by the force of attraction of an electromagnet. The advantages of electromagnetic clutch are remote or auto magnet control, quick operating, the possibly of adjusting the transmitted torque and the absence of unbalanced force. The main component of these clutches (as shown in fig.10.4) is electromagnetic system comprising the electromagnet frame, winding 4 and armature 1, and a set of friction disks 2&3. The coil is impregnated with epoxy resin, enabling the clutch to run in oil. The clutch is energized by direct current, which is supplied through a contact ring or slip ring 6, insulated from the casing by a collar 12, with brushes sliding over them. As the current flows through the winding, the armature is attracted to the frame and the friction surfaces are drawn together. The magnetic field arising, clamps the discs between the armature 1 and the casing 11. The casing rotates freely on the driven shaft together with a pressed in bush 10, and is retained against axial movement by a ring 9, secured with a lock screw. The maximum moment of friction of the clutch can readily be adjusted by varying the current intensity. When the current is switched off, the armature is drawn away by spring etc. and the friction disks are released. In general, one terminal of the coil is connected to the contact ring and the other terminal through body to the earth or through other contact ring to the brush.

![Fig.10.4 Typical assembly of electromagnetic friction clutch](image)

In a clutch with magnetically attracted disks, the annular magnetic flux pierces the disks and its circuit is closed through the armature. To prevent the magnetic flux from shorting through the disks, the latter have holes provided along an annular belt opposite the coil. The armature consists of two rings, one outer and one inner ring enabling it to bear more tightly against the disks if the latter are somewhat tapered. The clutch body and armature are made of soft steel or mild-steel. The disks are made of manganese steel with a hardness of 40-45 Rc.

Degree of wear of brushes and electromagnetic disks should be checked periodically. In order to check wear of brush, unscrew brush holder by one turn, if still contact is maintained the wear is with in limit.
In repairing the clutches, disks are ground or replaced so that the size of the new set corresponds to the distance between armature and housing. A greater distance will reduce magnetic attraction, and this may result in slipping and intensive wear of the disks. A worn armature should be ground on a surface-grinding machine until all traces of wear in the form of circular lines are eliminated. Disks assembly should be such that first disk rotate synchronously with coil. Inner ring of the clutch armature must not project with respect to the outer ring otherwise only the inner ring of the armature will operate.

For disk clutches, if it works with high frequency of engagement and disengagement, and at higher speed actual transmitted torque will be less than the calculated value by following factors.

\[ T_{\text{actual}} = T_{\text{des}} K_m K_v (1-K'_n) \]

Where, \( K_m \) = coefficient depending upon number of disks.
\( K_v \) = coefficient for speed

**Table 10.1 Coefficient depending upon number of disks \( K_m \)**

<table>
<thead>
<tr>
<th>No of outer disks</th>
<th>3</th>
<th>4</th>
<th>5</th>
<th>6</th>
<th>7</th>
<th>8</th>
<th>9</th>
<th>10</th>
<th>11</th>
</tr>
</thead>
<tbody>
<tr>
<td>( K_m )</td>
<td>1</td>
<td>0.97</td>
<td>0.94</td>
<td>0.91</td>
<td>0.88</td>
<td>0.85</td>
<td>0.82</td>
<td>0.79</td>
<td>0.76</td>
</tr>
</tbody>
</table>

**Table 10.2 speed Coefficient \( K_v \)**

<table>
<thead>
<tr>
<th>( v ) in m/sec</th>
<th>2.5 and less</th>
<th>3</th>
<th>4</th>
<th>5</th>
<th>6</th>
<th>8</th>
<th>10</th>
<th>13</th>
<th>15</th>
</tr>
</thead>
<tbody>
<tr>
<td>( K_v )</td>
<td>1</td>
<td>0.94</td>
<td>0.86</td>
<td>0.80</td>
<td>0.75</td>
<td>0.68</td>
<td>0.63</td>
<td>0.59</td>
<td>0.55</td>
</tr>
</tbody>
</table>

\( K'_n \) = coefficient signifying number of disengagement

If frequency of engagement is 50 per hour, \( K'_n = 0 \)
\( K'_n \) is taken as 0,01 for every extra 5 frequency. For example,

\[ K'_n = 0,01 \frac{150 - 50}{2} = 0,2 \]

If \( K'_n \) >0,5, special material will have to be used for heat dissipation.

**CONE FRICTION CLUTCHES**

One of the members of an ordinary cone clutch has an internal conical working surface and the other member has a mating external cone. The clutch is engaged and disengaged by axially shifting one of the members. The conical friction surfaces enable considerable normal pressure and friction forces to be produced with a relatively small engaging force.

To avoid self-engagement and to facilitate disengagement, the cone angle (angle between conical surface and shaft axis) is taken greater than the static friction. For metallic friction surfaces the cone angle is taken from 8° to 10° or more while for asbestos-base linings it is taken from 12° to 15° and more.
Cone friction clutches are used for engaging or disengaging two axially arranged shafts. They are also used as safety clutches. Double cone friction clutches are necessary whenever the motion has to be reversed. A cone friction clutch assembly is shown in the figure 10.5. The driven shaft 5 takes its drive from the driving shaft via gear gearwheels 1 and 19. These wheels are freely fitted on the shaft 5 and cannot impart motion directly to it. A bronze bush 6 secured with a screw 7, and compensating washers 3 are fitted in the bore of wheel 1. Wheel 19 is free to rotate in bronze bush 18. This bush is fixed to the shaft 5 by means of setscrew 17. Compensating rings 14 and a lock ring 15 hold wheel 19 and prevent it from shifting along the shaft when engaging the cone disk 20.

Both wheels are provided with internal taper bores to receive cone disks 20 and 23. The right cone disk 20 has a lengthened hub. Left cone 23 is fitted on the end of this hub. The cones are fitted on a distance-bush 2 so that they are able to move axially although key 22 prevents their rotation. As the hoop 9 moves to the right its bevelled edge will engage the rounded part of the detent 13 pressing it down and turning it about axle 16. As the detent turns to the right, it moves cone to the right through axle 16, secured in the hub of cone 20 until it is pressed against tapered surface of wheel 19. If the hoop 9 moves to the left it will turn the detent the opposite way. Cone 20 will disengage and cone 23 will be put into motion and wheel 1 will transmit the torque to the shaft 5.

Fig.10.5 Cone friction clutch assembly
The torque that can be transmitted by a cone clutch with a mean radius $R_m$ of the friction surfaces and width is

$$T = \frac{2}{f} \mu \pi R_m^2 b p$$

Usually $b/R_m = \psi = 0.3$ to 0.5

The force required engaging the clutch is

$$F \equiv \frac{f.T . \sin \alpha}{\mu R_m}$$

Where $\alpha$ is the angle between an element of the cone surface and the shaft axis (cone angle).

The taper disks are repaired by working their taper surfaces with emery cloth and machine inner faces to provide a greater clearance for the axial movement of the disk. Sometimes taper position is bored out, a compensating ring is press fitted into it and secured with a pin to prevent its rotation. The conical surface may be machined off and a compensating ring fitted on it. The surfaces are then tapered as required.

It is advisable to mount the rings on epoxide glue instead of press fitting. The rings should be first mounted on the disk with a running fit to provide a uniform thin layer of glue in the joint.

Sometimes it is expedient to manufacture new halves for a small cone friction clutch. The taper by surfaces of the clutch is carefully fitted together by lapping with fine emery powder, after which the lapped surfaces are thoroughly washed in kerosene.

**RADIAL CLUTCHES**

These are radially expanding or contracting clutches / brake of the band, ring or shoe types. Radial clutches can be classified into four categories.

- With outer shoe.
- With inner shoes.
- With band.
- With expanding ring.

For radial clutches with a constant unit pressure $p$ over the entire surface, torque $T$ can be calculated by

$$T = \frac{2 \pi \mu p b . r_0^2}{F}$$

Where $F$ is factor frictional arrangement =1.3 to 1.5

$b$ is the width of achieve surface of drum.

$r_0$ is the radius of drum.

If only small portion of the drum is active at a time, the torque will reduce correspondingly in the same ratio.
(a) With outer shoe: An outer shoe brake consists essentially of a drum mounted on the shaft to be stopped, two or more shoes that presses radially inward on the external surface of the drum, and a suitable lever system to apply and release the shoe. The brake shown in figure 10.6 is hydraulically applied, and spring released. The shoes are pivoted on the brake arms in a “floating” shoe construction. The brake could also be spring applied and released by the electric solenoid. This brake does not have the floating shoes, but instead has shoe integral with the operating levers. Floating shoe construction does not require careful alignment between drum and shoe assembly. The brake drums should be made of “cannonite” cart-iron. An adjustment device is provided to compensate for wear of the shoes. The shoes are commonly lined with a molded asbestos compound brake block, which is fastened to the shoe with brass bolts in counter-shunk holes. Stops are provided to the limit the travel of the shoes so that both shoes move away from the drum when brake is released. Pins provided in the linkage should be hardened and ground. In outer shoe clutch, provision is made for rotating the shoe and for controlling the load on it. The driving plate carries the two pivoted with shoes.

(b) With inner shoes: Placing the shoes inside the drum, as shown figure 10.7 is self-energizing or will help to apply itself. These arrangements can produce a large braking force by the application of a relatively small force at the shoes, and has the added advantage of enclosing the shoes to keep out dust, water etc. but it is more difficult to dissipate the heat generated during the braking period. The shoes should be made somewhat flexible to allow them to conform to the shape of the drum.

Fig.5 Clutch with inner shoe
(c) **With band:** Band brake is more effective with one direction of rotation, than with the other direction of rotation. The band brake can be power actuated by adding a solenoid, air cylinder, or hydraulic cylinder to the, of canvas impregnated with rubber, or of a steel band faced with asbestos.

**10.3. POSITIVE CLUTCHES**

Jaw or toothed clutches are employed to transmit considerable torque under conditions of small overall size, in frequent engagements and when smooth engagement is not compulsory. These clutches provide positive engagement and that is why called positive clutches.

**JAW CLUTCH**

These clutches consist of two halves (clutch member) having interlocking teeth or projecting lugs on their end faces. The members are engaged or disengaged by axially shifting one into or out of mesh with the other along multiple splines or guide keys on the shaft. The clutch is operated by hand, tractive electromagnet or pneumatic or hydraulic means acting through levers, fork and sprags on the moving half. To engage or disengage a jaw clutch under load, high velocity over a short distance is required. Preloaded spring is used sometimes for this purpose. As a rule to reduce wear of the engaging mechanisms, the sliding clutch member is mounted on the driven shaft, so that sliding of the shifting fork in the slot of the sliding clutch member occurs only when the clutch is engaged.
Jaw clutches require accurate alignment of the shafts being joined. This may be accomplished with the aid of a centring ring. In most of the cases teeth or projecting lugs of one clutch member are usually cut directly on the end face of the hub of the gear, which is already properly aligned. Disadvantages of these clutches are that they can be engaged only at very slow speeds, with a peripheral velocity below 1m/sec. The mechanism is subject to shocks during engagement in motion. Jaw clutch tend towards self-disengagement even when angle of front edge is zero (α=0). Self-disengagement is completely eliminated in jaws where α<0. The sliding half of clutch should be of length l ≥1.5d, sliding on splines or double keys.

The working surface of the jaws and also of the hub of the moving half should have high hardness. For this reason, clutches engaged under load are made of low carbon chromium steel case hardened to Rc 56-62 and those engaged at small velocities of medium carbon chromium steel induction hardened to Rc 48-54 or med. carbon steel induction hardened to Rc 35-54.

**TOOTHED JAW CLUTCH**

The following types of clutch teeth or lugs are used,

1. Vee-shaped teeth have sides, which make an angle of 30° to 45° with the axis of the clutch. There are 15 to 60 teeth, and such clutches are used for low torques and speeds because for high torque considerable axial force is required to hold the members in engagement, at high speeds the tops of the teeth may be rapidly crushed and worn off. The chief advantages are easy and rapid engagement due the great number of teeth.

2. Trapezoidal teeth have slides with an angle of α = 3° to 10° to the clutch axis. These clutches are used for-transmitting high torque at high speeds.
3. Straight teeth are close in features to the trapezoidal teeth but cannot be engaged unless some clearance is provided and, consequently, operate poorly with reversing loads. These clutches are more difficult to engage than the proceeding types, but do not require the application of a constant axial force to hold them engaged, and have proper contact between the surfaces of the meshing teeth when not fully engaged. They are used in heavy machinery and for manual engagement.

4. Non-symmetrical Vee (saw shaped) and trapezoidal teeth are used in clutches, which drive in one direction only. Their main advantages are easy engagement.

5. Self-locking trapezoidal teeth are used for positive transmission of high torque. Here $\alpha$ is less than zero. It can not be disengaged on its own. To facilitate the engagement of clutches with trapezoidal or straight teeth they are made with larger peripheral clearance between the teeth or jaws by removing every other jaw (for non-reversing and constant loads), with every other jaw shortened or with 120° chamfers at the tops.
Greater the number of teeth, shorter will be the required engaging times. Odd numbers of teeth are generally selected in the clutch to enable two working surface of opposite jaws to be machined in a single pass.

![Diagram of a typical electromagnet slip ring toothed clutch](image)

**Fig. 10.9 Slip ring type gear coupling (Siemen’s design)**

a) Slipring, b) Toothed rim, c) External meshing toothed rim, d) Fastening movable spacer, e) Electro-magnet body, f) Bush  

A typical electromagnet slip ring toothed clutch is shown in figure 10.9. The clutch is consisted of magnetic body (e) with coil fitted along with a toothed rim (b), slip ring (a) and fastening movable spacer (d) over its external meshing toothed rim (c). With the power supplied through brush to the slip ring, magnetic field is created in the electromagnet body as shown by dotted lines. Both the toothed faces mesh together because of the magnetic force thus created and clutch is engaged transmitting torque to the other shaft coupled with it. Both the toothed rims are made of nitrided steel to make it wear resistant. Toothed clutch of this principle could be meshed only in standstill condition or in approximately synchronized condition. Allowable difference in r.p.m. and so the allowable actuating load-torque depends upon the turning flexibility and moment of inertia of the system. The clutch can be disengaged at all speeds and under load also.
Jaw-damage may consist in wear in operation (due to shaft misalignment) and in engagement under load, and in breakage and damage to the edges of teeth from engagement when the shafts are in motion. Heavy wear of the jaws may lead to self-disengagement of the clutch. Calculations are made on the basis of compressive stress and bending stress.

\[ P_{\text{comp}} = \frac{2T}{Dzh} \quad P_{\text{bend}} = \frac{2KTh}{DzI} \]

\( P_{\text{comp}} = 800 \text{ to } 1200 \text{ kg/cm}^2 \) for clutch engaged at rest.

\( P_{\text{bend}} = 300 \text{ to } 400 \text{ kg/cm}^2 \) for engaged at motion.

\[ p_{\text{bend}} = \frac{\text{bendistrength}}{SI} \]

\( S = \text{safety factor } 1.5 \)

Where \( D = \text{mean diameter of jaw.} \)
\( z = \text{no. of jaws} \)
\( b = \text{length of each jaw along radius} \)
\( h = \text{height of the jaws, measured along the clutch axis.} \)
\( I = \text{section modules of bending of jaws} \)
\( K = \text{non uniformity factor 2 to 5} \)
\( 2 \text{ for more accurate} \)

If the clutch is to be self-locking, the required claw angle is given by

\[ \tan \alpha \leq \frac{\mu_2 (1 + \frac{\mu_1 D}{\mu_2 d})}{1 - \mu_1 \mu_2 \frac{D}{d}} \]

and for claw clutch used as safety clutch

\[ \tan \alpha \geq \frac{\mu_2 (1 + \frac{\mu_1 D}{\mu_2 d})}{1 - \mu_1 \mu_2 \frac{D}{d}} \]

Where \( \mu_1 = \text{coefficient of friction between sliding half of clutch and the shaft} \)
\( \mu_2 = \text{coefficient of friction between the claws.} \)
\( d = \text{Shaft diameter at the sliding half of the clutch} \)
\( D = \text{mean diameter of the clutch.} \)

Torque coming on the clutch is given by

\[ T = 973000 \frac{Kw}{n} \cdot S \]

Where \( Kw = \text{power is kilowatt} \)
\( n = \text{r.p.m} \)
\( S = \text{safety factor } = 1.25 \text{ for machines which cut continuously} \)
\( = 1.5 \text{ for machines which cut intermittently} \)

Clutch engagement force \( F_{\text{eng}} = 2T \left[ \frac{\mu_1}{d} + \frac{\tan(\alpha + \lambda_2)}{D} \right] \)
Where \( \lambda_2 = \tan^{-1} \mu_2 \)

Generally \( \mu_1 & \mu_2 = 0.08 \) when lubricated
\( = 0.2 \) when not lubricated

Clutch disengagement force \( f_{\text{dis}} = 2T \left[ \frac{\mu_1}{d} - \frac{\tan(\alpha - \lambda_2)}{D} \right] \)

**10.4. CYLINDRICAL RADially & AXIALLY CONTACTING AIR CLUTCHES**

Where \( c = \frac{mv^2}{r} \) = centrifugal force acting on each shoe. In these clutches friction is developed between the shoes of a rubber type, secured to one clutch member, and the cylindrical rim of the second member (drum). To engage the clutch, air under pressure is admitted into the inner tube of the tyre, which expands so that the shoes are pressed uniformly against the drum. The tyre consists of (a) an elastic rubber inner tube which holds the air (b) multiple ply load carrying lining of tough rubberised fabric and (c) outer tread of rubber. The shoes are secured to the tyre by plain pins, which pass through holes in the pins. The tyre is heat insulated from shoes by a paronite lining which also protects it against the product of wear. The shoes are coated by a friction lining held by glue. This lining is usually made of an asbestos fabric band impregnated with phenolic resin.

Axial type air clutches are of disk type in which the inner tube plays the part of a controllable spring for axially compressing the disks.

The required force \( F \), which must be developed by the air tyre, is determined for contracting clutches by the equation:

\[ T = \frac{\mu}{S} (F - F_c) R \]

Where \( S \) = margin of frictional engagement.
\( F_c = \) centrifugal force of tyre together with shoes.
\( F = (q - \Delta q) A \)

Where \( q \) = air pressure in the tube (6 to 8Kg/cm\(^2\))
\( \Delta q \) = pressure required to deform the tyre (0.5 Kg/cm\(^2\))
\( A = \Pi Db = \) active area of surface inside the inner tube
\( b = \) width of cylindrical surface inside the inner tube
\( D = \) diameter of the surface inside inner tube.
10.5. CENTRIFUGAL CLUTCHES

Centrifugal clutch serve for automatic engagement or disengagement of coaxial shafts when the driving shaft reaches a given speed. These are clutches self-controlled on the basis of speed of rotation. These clutches are usually applied to actuate machines which have considerable fly wheel torque because the motor possess a comparatively low starting torque while considerable starting torque is necessary to bring machines with large masses into motion. For drives, which are difficult to start, centrifugal clutches allow the use of engine with a smaller power. Centrifugal clutches can also be used to prevent over speeding by using a normally engaged clutch.

Centrifugal clutches are friction clutches in which the ordinary control mechanism has been replaced by special weights, subject to the action of centrifugal forces and springs. When the driving shaft reaches a definite speed, the centrifugal forces acting on the weight overcome the resistance of the springs, force the rubbing surface together and the clutch is engaged. To reduce mass of the weight, they may be used in the form of levers tuning about axes.

Mostly three types of centrifugal clutches are used (a) with radial springs, pressing the shoes (weights) against the inner, driving clutch member (b) with out springs (c) semi centrifugal clutch. The clutches of first type begin to transmit torque at a shaft speed equal to 75% of the nominal speed; clutches of the (b) type begin to operate at low speeds. Semi centrifugal clutches have hand or foot controls, and the centrifugal force only increase the pressure on the rubbing surfaces.

For the clutch with z shoes, the force required to press each shoe against the rim is

\[ Q = \frac{T}{zRf} \]

The mass of each shoe \( m \) and the force \( S \) that should be exerted by each spring or elastic are found from the following equations of equilibrium of each shoe.

1. At speed \( n \)

\[ Q - c + S = 0 \]

While \( v \) = peripheral velocity of shoe's center of gravity, and \( r \) = distance from axis of rotation to shoe’s C.G.

2. At speed \( n_0 \) when engagement is to begin, \( Q = 0 \)

\[ -C_0 + S = 0 \]

where \( C_0 = \frac{mv^2}{r} \), i.e. centrifugal force at \( n_0 \) speed.

In some clutches steel balls or shot lubricated by graphite serve as the working members. Friction force between these members transmits the torque. When the prime mover is started, centrifugal force throws this charge to the perimeter of housing, which is keyed to the input shaft and pack it against the rotor, which transmit power to the load. At predetermined full speed the shot charge is packed solid and the housing and rotor lock together without slip unless overload exceeds a set torque limit.

Generally in a ball type centrifugal clutch, the driving element is a vane wheel and the driven element is a cylinder enclosed on end faces. The chambers formed by vanes and the cylinders are filled uniformly with shot 5-10mm in diameter. Some 10-15 grams of oil are added to 1 kg of shot. An increase in the vane wheel speed intensifies the centrifugal force, which presses the shot against the driven part of the clutch. The parts of this clutch are usually manufactured from cat-iron or from steel if the clutch is to be engaged frequently. The lubricant should be replaced frequently.
10.6. FLUID CLUTCHES

It consists of two impellers: the pump impeller linked to the driving shaft and the turbine impeller linked to the driven shaft. These impellers have radial vanes in annular semicircular chambers and are mounted so that the chambers face each other. The working space in the clutch is fitted to 80 or 90% of its volume with light mineral oil with kinematic-viscosity of 16 to 32cst. Torque is developed by slip since the pressure differential between impeller and runner rotor, which causes the fluid to circulate and transfer energy, is a function of the difference in speed between the members. Normally 2 to 6% slip is needed to develop full torque.

Hydrodynamic (fluid) clutches are employed in machines which often operate under non-steady state conditions (starting, stopping and varying speeds), having larger masses that must be accelerated to running speed.

10.7. POWDER MAGNETIC CLUTCHES

The principle of this clutch is based on the fact that the iron powder in the clutch, when magnetized by the magnetic flux, resists shear, and its shear strength increases with the intensity of magnetization. The greatest relative displacement of the particles of powder occurs at the middle of the layer. The layers of powder adjacent to the magnetized surfaces do not move with respect to these surfaces and are not subject to wear. Advantages are (a) exceptionally rapid action (b) possibility of fine control of the torque (c) high wear resistance. These clutches operate with their volume rather than their surfaces and can transmit higher torque than the induction clutches. But mixture is required to be changed periodically. Proper sealing is to be ensured. Carbonyl iron powder (containing 0,7 to 0,8% carbon) in spherical size 4 to 8 microns in diameter is mostly used. To reduce oxidation and the resistance to shear when the clutch is disengaged, oil is added to the powder Molybdenum di-sulphide or graphite is added dry powder clutches. The mass ratio of the oil and iron powder is about 1:5.

The main components of the clutch are (a) the core, which is one of the clutch member (b) Coil (c) the second clutch member- a ring. Between the core and ring is an annular gap filled with iron powder in oil slurry. The core is of split design to facilitate clutch assembly and for changing the oil. Gear etc. should be magnetically insulated by a disk of non-magnetic material to reduce dispersion of the magnetic flux.

Magnetic powder clutches are designed with an annular working gap, a plane-working gap, or for high torques with several working gaps. The gaps are from 0,5 to 2 mm wide. If especially rapid action is required, the driven clutch member can be of lightweight design for example, in the form of a thin disk located in a gap between two halves of the core.

Torque (transmitted) = f.q.S.r Kg.cm

Where 
\[ S = \text{working surface of cylinder or disk in cm}^2 \]
\[ r = \text{radius of cylinder or disk in cm.} \]
\[ f = \text{cohesion factor(0,1-0,3)} \]
\[ q = \text{unit pressure} = \left( \frac{B}{5000} \right)^2 \text{Kgf/cm}^2 \]
\[ B = \text{magnetic flux intensity.} \]
10.8. OVER RUNNING CLUTCHES

In machine tools and equipment it becomes often necessary to impart to one shaft two diverse motions (a slow working motion and rapid idle motion) transmitted by two separate kinematic chains. Over running clutches are employed to engage the rapid motion without disengaging the working motion chain. Over running clutches also called free wheeling clutches, automatically connect and disconnect shafts depending on the ratio of their angular velocities. When the velocity of the driving shaft exceeds that of the driven shaft, the overrunning clutch connects the shaft. When the ratio of the velocities is reversed, the clutch is released without hampering the driven shaft to overtake the driving shaft if the former is driven at a higher speed by another kinematics train. Depending on the mode of engagement over running clutches is subdivided into (a) ratchet (b) friction clutches.

RATCHET TYPE OVERRUNNING CLUTCHES

Ratchet type over running clutches are used for slow speed shafts. Ratchet wheels have nonsymmetrical teeth and a pawl, which drops into the tooth space. The pawl transmits torque in one direction and when the direction of the torque is reversed the pawl is disengaged. Merit of these clutches is reliability in operation and the small normal forces developed. Drawbacks are the impossibility of disengagement at any relative position and the application of radial loads to the shafts.

ROLLER TYPE OVERRUNNING CLUTCH

This is the most common type of overrunning friction clutch used in modern machine tools. It consists of a spider, annular shell and roller. The spider and the shell form recess which narrow toward one end and hold the rollers, one in each recess. Usually the shell has plane cylindrical surface and the spider has flat notches. Clutches commonly have three rollers or five rollers and more. Each roller is forced by springs into the narrow end of the recess.
If the shell is the driving member, then the clutch shown in the figure 10.10 can transmit rotation in the clockwise direction, if the spider is the driving member, in the counter clockwise direction. In either case, the rollers are rolled into the narrow end where they jam between the, spider and shell. Upon reverse rotation, the rollers roll toward the wide end of the recesses, disengaging the clutch.

The tangents to the roller circumference at the points of contact with the spider and shell form the small angle $\alpha$. The magnitude of this angle, called the angle of wedge, is usually $\alpha=3-6^\circ$. Lesser value of this angle should be avoided since in this case the release of the clutch is more difficult. Larger angles are not used because the clutch may tend to slip, especially after some wear and deformation of the contact surface.

Another design of overrunning clutch has working surfaces on the spider, which are not flat, but of cylindrical shape with eccentrically located centres (with respect to the axis of the spider). In these spiders, the pressure angle between the working surface and rollers varies comparatively little with changes in roller diameter. Therefore more wear can be allowed and the service life of the worn clutches can be prolonged by replacing the set of rollers with rollers of another diameter.

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**Fig.10.10 Scheme of simple overrunning clutch**

**Fig.10.12 Typical designs of overrunning clutch with fork member**

(a) Type I with shell rotating in one direction. (b) Sectional view of both types (c) Type II with rotation in both the directions
One known design of overrunning clutch exists, as shown in the figure 10.12(a) that can transmit slow rotation in one direction and rapid rotation in both directions. Rapid rotation is transmitted through a fork member 8 that engages the rollers like a cage as shown in fig.10.12 (b). The driven element is the spider 7. From shell 5 connected with worm wheel 4, to spider 7 rotations is transmitted in counter clockwise direction only. Fork 8 connected with independent source of motion (rapid motor) through gear 9 may transmit rotation to spider in both the directions irrespective of its connection with shell.

Overrunning clutch shown in figure 10.12 (c) can transmit both slow and rapid motion in the both directions. It may be considered as combination of two couplings with single fork. Driving shell can transmit rotation to spider in both the directions. Fork can also drive spider in both the directions.

Approximate design proportions for determining the diameter \( d \) and length \( l \) of the rollers: \( d \approx \frac{D}{8} \) and length \( l = (1.5 \text{ to } 2) \, d \), where \( D \) is the diameter of the working surface of the shell. Recommended material, for clutch components are ball bearing steel, or carbonising steel with a large case depth. In practice, an allowable contact stress of \( \frac{15000 \text{ Kg}}{\text{cm}^2} \) is taken when the surface hardness of the compressed members is more than 60Re. With the contact stress and \( \alpha = 7^\circ \), the torque transmitted will be equal to \( T = 8.5 \, zdD \) where \( z \) is no of teeth on spider/no of rollers.

The load capacity of these clutches can be raised by brazing cemented carbide inserts into the spider under the rollers.

**10.9. RIGID COUPLINGS**

Rigid couplings transmit not only torque but also bending moments and axial forces arising in the assembly. The couplings are often arranged near the shaft bearings to relieve these from additional forces. They are small in size, cheap to manufacture and have good resistance to wear. Worm couplings are not repaired but simply replaced by new ones. Different types of couplings are described below.
SLEEVE COUPLINGS

They consist of a sleeve mounted on the ends of shafts and secured by pins, usually taper pins, and straight or woodruff keys; multiple spline joints or setscrews. Assembly of the couplings requires considerable axial motion of shafts. It becomes difficult to use interference fits. For this reason, sleeve couplings do not ensure high rigidity of the joint in bending. Sleeve couplings are commonly employed for shafts upto 100mm. Couplings are made of structural steel.

Outside diameter \( D = (1.5 \text{ to } 1.8) \ d \)

\[ L = (2.5 \text{ to } 4.0) \ d \]

where \( d \) is the diameter of shafts

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**Fig.10.13 Sleeve coupling**

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CLAMP COUPLINGS

The couplings are made of two halves spilt along a plane passing the axis of the shafts being joined. These couplings are fitted on the ends of shafts, which have already been aligned and clamped by bolts. Small and medium size couplings transmit torque only through friction between the shafts and the coupling. In large size couplings a key incorporated between the shafts and couplings transmits the torque.

During assembly, after aligning the shafts seat keys in the keyways and turn one shaft to align the end faces of both the keys. Now place coupling half on both the shafts. The other coupling half should then be mounted and the two untied first with two bolts (arranged on the diagonals), and then by tightening the other bolts. Check that the end faces of both halves are aligned with each other and the keys do not come out.

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FLANGE COUPLINGS

A flange coupling is fitted on to the shaft ends in a hot state or under pressure or tights fit secured with feather keys, before the shafts are mounted in to the bearings. When large torque and forces are to be transmitted, the flanges are made integral with shafts. Flange couplings transmit torque either through friction between the ends of the halves clamped by bolts or through bolts in shear fitted tightly into the holes of the coupling halves.
Total length of a coupling \( l = (5-2.5) d_{\text{shaft}} \)

Outside diameter \( d_{\text{out}} = (4.5-2) d_{\text{shaft}} \)

The halves are cantered with respect to each other by a shoulder on the left half and a recess in the right half. In such flange coupling having shoulder, one shaft is placed in the bearings and coupling half serves as the base for aligning the other half. Attach an indicator to one coupling half and use it to check the rim of the other coupling half for run out. Then slide the shaft towards former coupling half so that recess of coupling half can partially engage the cantering shoulder of other half, leaving clearance between the end faces of the halves. This clearance is checked by a thickness gauge for uniformity.

**COMPRESSION COUPLING**

Cone couplings and fluid pressurized couplings are the most commonly used compression couplings to couple a shaft and a hub of gear or timing pulley etc. The cone coupling utilizes two split cones, which are drawn together by the bolts in order to produce action, which tightens the parts of the coupling and the shafts. Some types of compression couplings do not have keys, but depend entirely on friction produced by the compression pressures to prevent slipping of the parts in transmitting torque.

They are built in the form of taper rings with self-releasing tapers. Such couplings are capable of transmitting a torque from the shaft to the hub or vice versa without any backlash. For assembling the bushes, male and female taper bushes are forced into each other by tightening the screws axially.

This expands the bushes and generates an enormous radial force due to small taper angle, which locks the hub and the shaft, and the torque is transmitted through friction.

*Fig.10.14 Cone type compression coupling*
The fluid pressurized coupling consists of a double walled hardened steel sleeve filled with a pressurized liquid medium, a sealing ring, a piston, a pressure flange, and clamping screws. When tightening the screws, the bush expands uniformly against the shaft and the hub and creates a rigid joint. On loosening the screws, the bush returns to its original position and can easily be dismantled.

*Fig.10.15 (a) Fluid pressurized coupling*

*Fig.10.15 (b) Inner details of fluid pressurized coupling*
10.10. FLEXIBLE COUPLING

Sometimes accurate and constant alignment of shafts cannot be ensured. The inaccurate relative position of the connected shafts, inevitable at the outset due to error of manufacture, is on the course of time aggravated by deformations caused by the working load, temperature fluctuations, the uneven sinking of the foundation. In such cases flexible couplings are used. Small misalignment of the shafts is compensated for by the relative mobility of the coupling elements.

This is achieved by one of the following methods.

1. A larger clearance between the conjugate parts of the coupling, for couplings working at low velocity for small load.
2. The sliding of some parts relative to other; such couplings should be lubricated.
3. The elasticity of the parts, these couplings requires no lubrication but cause an additional load on the shafts and bearings.

Couplings having first two methods of compensation can be called rigid slip couplings while couplings using third method of compensation are called flexible slip or elastic couplings. Rigid slip couplings are subdivided into gear, Oldham and universal joint couplings.

GEAR COUPLINGS

Gear coupling consists of two sleeves with external gear teeth and two casing with internal teeth. The sleeves are fitted onto the ends of the connected shafts. The two casings are held together by screws. The teeth on the sleeves and in the shells have ordinary involute profiles with a pressure angle of 20°. The sleeves have cylindrical concentric shoulders for checking coaxiality of the shafts (with a dial indicator). Oil is essential for operation of a gear coupling, it reduces friction between the teeth and decreases resistance to the relative displacement of the coupling halves. The top of teeth on the sleeves is rounded by a radius equal to the addendum radius of the sleeve teeth i.e. it is spherical. The angle Ψ of the allowed angular misalignment between the shaft axes is found from equation

\[ \sin \frac{\Psi}{2} = \frac{\Delta}{b} \]  

(generally \( \Psi/2 < 0°, 30' \))

Where \( \Delta = \) backlash between the teeth  
\( b = \) active width of the teeth

Allowable offset of centers \( \delta = \frac{l}{b} \Delta \)

Where \( l = \) distance between the tooth centers on both sleeves.
Gear Couplings have high load carrying capacity and can run at high allowable speed (upto 25m/sec). Sleeve teeth should be hardened to 35-45 RC and mating teeth of casing to 45-55 RC. The toothed rims of the casing may be made of a plastic (caprolon) to have more uniform distribution of the load among the teeth and better wear resistance quality.

The couplings are lubricated with high viscosity oil, which is poured into the housing formed by the shell. During operation, the contacting teeth slide axially along each other by the distance ($\alpha D$) in each revolution, while $\alpha$ is the angle of misalignment between the sleeve and casing and $D$ is pitch diameter of the teeth. This sliding is responsible for wear of the teeth. Couplings are selected according to the torque to be transmitted.

Torque ($T$) = $c.b.D^2$

Where $c$ is proportionality factor varies from 1 to 1.25

$b$ is width of the teeth

$D$ is pitch diameter of the teeth.

If external and internal teeth are worn out, it is not recommended to repair the teeth wear and the half coupling should be replaced in pair. However if holes of the bolts are worn out, repair is carried out by boring over size holes.

**CHAIN TYPE COUPLINGS**

It consists of two chain sprockets (mounted on the ends of the shafts being joined and having the same number of teeth), a piece of chain encircling the sprocket teeth and housing. Chain type couplings permit angular misalignment of shafts upto $1^\circ$ and radial misalignment upto 1.2mm. If the angular misalignment of the shafts is very large (upto $3^\circ$ or even $5^\circ$), double strand roller chain with barrel shaped roller is used.

Owing to the backlash in chain couplings, they cannot be recommended for reversing drives or drives with high dynamic (shock) loads.
OLD HAM COUPLINGS

It consists of two end pieces with rectilinear grooves and a centrepiece with tongues positional; at right angles to each other. It can take angular misalignment of that of $\psi=1^\circ$ and an offset of the shaft axes of not more than $\delta \approx 0.05 \, d_{sh}$.

If the centrepiece is modified to the form of a square block, the coupling will have a larger active surface and retains oil better and can permit $\delta \approx 0.1 \, d_{sh}$ and $\psi \leq 3^\circ$. An involute tooth torque and a wedge groove allows a bigger misalignment of the order of $\psi \leq 4^\circ$. The intermediate-floating member (ring or slide block) describes with its centre, a circle of diameter equal to the misalignment $\epsilon$ of the axes. So speed of rotation of couplings is limited because of large centrifugal faces. The permissible speed for couplings up to 300 mm diameter is 250 rpm. The slide block is commonly made of laminate fabric base, which makes the coupling electrically insulating, and reduces its mass.

![Fig.10.17 Fitting of Oldham coupling](image)

These couplings are subject to considerable wear. It is therefore advisable to use anti seize oils for their lubrication. Wear affects the grooves in end piece and torques and faces of the centre block. New grooves are cut oversize and centre block is replaced. So that a gap of about 0.1 mm is obtained.

UNIVERSAL COUPLINGS

Such couplings, also called Cardan or Hooks joints, connect shafts whose axes are at an angle during operation. It can transmit torque between shafts whose axes are not collinear by an angle up to $40^\circ$-$45^\circ$ because the coupling has two joints with axes perpendicular to each other. By applying telescopic intermediate shaft, the misalignment (distance between shafts) can be varied during operation.

In small size universal couplings, the centrepiece has the shape of a parallelepiped. The pivots are formed by inserted pins, one being long and passing through the joint, and the other consisting of two short bushing held together by a through rivet. The rubbing components are made of chromium steel and are heat-treated to a hardness of 58-64Re on the working surface. Alloy of chromium and chromium nickel steels are commonly used and for coupling of smaller size, bearing steel as well. In medium type coupling, the centre cross and the yokes are made of steel EN-18 hardened to Re (48-53), Universal -joint couplings should be lubricated and protected from dust and dirt.
The driven shafts of a single universal coupling, when it is not coaxial with the driving shafts, rotate with non-uniform velocity at a constant velocity of the driving shaft. Angular acceleration of the driven shaft caused by this phenomenon develop inertial forces, which increase the load on the coupling components. Synchronous rotation of the driving and driven shafts can be achieved by installing two universal couplings so that the second coupling compensates for the non-uniform rotation produced by the first. For this following conditions must be complied with:

(a) The axes of the driving and driven shafts must make the same angles with intermediate shafts.

(b) The forks at both ends of the intermediate shafts should be arranged in a single plane.

10.11. ELASTIC COUPLINGS

A flexible elastic coupling consists of two halves and the elastic elements, which may be of metal (steel springs) or non-metallic (usually of rubber). A properly selected elastic coupling is an effective means against vibrations. The unit energy of elastic deformation of rubber amounts to about 4500 Kgf-m/Kg whereas for spring steel it is below 3Kgf-m/Kg. But the service life of the rubber elements is shorter. Rubber is having considerable damping capacity. For transmitting low and medium torque it is most advisable to employ natural rubber. Following types of couplings of this category is widely in use.
FLEXIBLE PIN TYPE COUPLINGS

This type of coupling is also some times called box pin coupling. It consists of two flanged halves 1 and 2 (fig.10.19). In one half steel pins 3 with taper shanks are secured and rubber sleeve composed of trapezoidal rings 4 are placed on them. The disk of the other half is provided with round holes for the pins. In place of rubber sleeves, collars made of leather or rubberised fabrics are also used. The collars engaged in holes in the other half and their elasticity permits a certain offset or misalignment of the shafts during operation. The pins are secured with nuts 5. The overall dimensions are D= (3.5 to 4)d and L= (3.5 to 4) d. The couplings are used for transmitting torque upto the maximum torsional stress 200 to 250Kgf/cm$^2$. The pins are made for medium carbon steel (EN-18, steel 45) while flanged hubs are made of cast-iron. The speed on the outside diameter of these couplings is limited to 30 m/sec, because at a higher frequency of load repetition the rubber sleeves are heated and disintegrated. The rings are made of rubber with a tensile strength of at least 80Kgf/cm$^2$ and 80$^o$ IRh hardness. The couplings are intended only to compensate for axial shift of the shafts. Radial and angular misalignment shortens the service life of the elastic element. Radial misalignment of 0.2 to 0.6 mm can be allowed while angular misalignment should not exceed 1$^o$. Wear affects the holes in coupling half accommodating collars, as well as collars themselves which begin to turn on pins. More often than not, nuts also slack off. Then pins begin to rotate too, which wears down their seats and the pin themselves in the other half.

To repair, enlarge holes for pins and collars and change for new pins and collars. The outer diameter of the new collars should precisely correspond to the diameter of the bore out holes in the coupling half. In boring out, ensure the alignment of the hole centers in the two halves. If the fit of the couplings on the shaft deteriorates, it is restored by pressing a bush into its flange.

As per IS: 2693 the coupling has been designated as bush type flexible coupling and various dimensions of standard couplings are given.

COUPLINGS WITH A RUBBER SPROCKET

Such couplings consist of two halves with two (for D=25-40mm) or with three (for D =50-160mm) triangular or trapezoidal jaws. The jaws drop into the respective gashes in the sprocket made of rubber. The teeth of the sprocket operate for compression.

In rubber-metallic couplings, the rubber is tightly bonded to metal parts. The load is uniformly distributed over the entire contact surface between the metal and rubber, thus making it possible to utilize the elastic properties of rubber to the best advantage. In small couplings, for torque upto 650Kgf-cm, the rubber is bonded directly to the coupling halves. In large couplings (upto 1600Kgf-cm) the rubber is bonded to intermediary flanges fastened to the coupling halves by bolts.
METAL BELLOWS COUPLINGS

Metal bellow couplings are torsionally stiff couplings, which are nondisengageable, elastic, as well form and friction fitting. A metal bellow coupling behaves like a rigid element in the direction of rotation. In axial and angular directions however it has elastic properties. Misalignment caused by fitting errors or other influences can be compensated for by these elastic properties. The greater the number of convolutions in the metal bellows, the greater is the ability to compensate for shaft misalignment. However the number of convolutions is limited mainly by the accuracy of transmission, which decreases as the number of convolutions increases. Their application is mainly to couple servomotor and ball screw. Also their use is as transducer coupling for coupling measuring systems like encoder etc.
Following three kinds of errors can be compensated by using metallic bellow coupling.

(a) Radial misalignment ($\gamma$)
(b) Angular misalignment ($\alpha$)
(c) Axial shift

**Fig 10.21(b) Radial and angular misalignment in metallic bellow coupling**

**COUPLINGS WITH METALLIC (STEEL) FLEXIBLE ELEMENTS**

Their chief field of application is in the transmission of high torques. There are several types of couplings in this category. Flak coupling is composed of two slotted members connected by a continuous steel spring, which lies in the slots. The elements of the spring provide the flexibility of the coupling. It may be noted that the effective length of the elements of the spring changes from a maximum $g$ at no load to minimum $g'$ at full load, so that as the torque on the coupling increases the coupling becomes stiffer. This is very desirable characteristic for installations in which torsional vibration of the shaft is an important consideration. The coupling parts may be readily disconnected by removing the steel spring.

Covers are provided to retain the coupling lubricant and prevent dust, grit or other foreign materials from coming in contact with or between the sliding parts. The cover may be split either horizontally or vertically grease hole permit lubrication without disturbing the gaskets or seals.
Proper lubrication is essential. Seals must be in good condition and properly seated. If the lubricant is abnormal the cover should be opened and all parts thoroughly flushed before the new lubricate are added. If the spring elements are significantly wormed, they should be replaced. Misalignment of the connected shafts should be kept within the manufacturer's recommendation. Excessive amounts may cause rapid wear of the spring elements and the hub slots as well as early failure of the cover seals. A spacer bars or straight can usually be used to check angular and parallel misalignment as well as shaft gap.

In assembling a resilient spring coupling first seal the two halves and keys on the ends of the shafts. After that aligns the shafts with respect to each other, place the springs in the slots in the halves and assemble the casing. The only link between the shafts is spring fitted in to the slots. Therefore torque is transmitted from one shaft to the other solely by the springs.

10.12. ALIGNMENT OF THE COUPLINGS

For smooth running the couplings should be aligned with the greatest possible accuracy. Better the initial alignment, more the capacity of the coupling to take care of operational misalignment. Changes from the initial condition occur because of wear, settling of foundations, and base distortion due to torque, thermal changes, and vibration in the connection. To increase the life of the coupling, the alignment of the connected machines should be checked at regular intervals.

There are three conditions of misalignment that a flexible coupling must accommodate:
(i) Angular misalignment.
(ii) Parallel misalignment.
(iii) End float.

Some methods for alignment of coupling are given below.

**A.** (i) Check the angular misalignment using tapered gauges between coupling faces at each 90° around the faces.
(ii) Use straight edges and feeler gauges to check the axial alignment at each 90° of rotation.

When checking the angular and axial alignment, the coupling flange-faces must be parallel with the edges in a straight line, respectively, except for the amount of vertical difference suggested in instruction book.

**B.** Put level on driven shaft or a flat-machined surface parallel to the axis of the driven equipment. Adjust the unit with shims until it is levelled.

In order to find out shim thickness, following example of motor pump set can be taken as guide. A shim placed under B will raise the top of the coupling at D 1.5 times the shim thickness. The motor coupling face at D will open up a distance about equal to the shim thickness. At E the coupling face will open only about half the shim thickness.
A shim placed under A will have the reverse effect on all parts.

For gear type coupling, after pushing back coupling covers, measurements are taken by taper gauges and straight edges as mentioned in A. Measurements are taken on coupling hubs.

D. For floating type coupling, jack shaft between the driven element and driver must be removed and a bracket must be made. It must be long enough to reach from one coupling to another when fastened to one. Fasten the bracket to one coupling and a dial type indicator to the arm. So the indicator contacts the rim of the other coupling half. Revolve the left coupling slowly. See the indicator about the right hand one. Alignment should be within 0, 08 mm. Check for shaft end play with an inside micrometer. When one coupling is aligned reverse the bracket and perform the same step on the other coupling. When alignment is complete insert the jackshaft and bolts up the coupling halves.
Chapter 11: SAFETY DEVICES

Devices for protecting the machine tool and cutting tool against breakage or damage, and the work piece from being spoiled, can be classified into three main groups.

(a) Interlocking devices.
(b) Travel limiting devices.
(c) Overload protection devices.

11.1. INTERLOCKING DEVICES

Interlocking devices serve to prevent simultaneous engagement of two or more pairs of gears in a single transmission group and other mechanisms with conflicting actions. Interlocking devices are also used to ensure that certain control operating not be performed except in a definite sequence.

There are a great variety of mechanical, electrical, hydraulic and Electro-hydro mechanical interlocking device arrangements applied in machine tools. For example where four speeds are transmitted between two shafts, the block mechanism provides guide and lock such that when one handle engages transmission, the other has to be in a neutral position. Rods perpendicular to each other can be interlocked with the slotted disks.

In a single lever control system simultaneous engagement of two speeds is impossible. While in multiple lever system, the parts of the mechanism are linked to a single hand control member to prevent simultaneous engagement of conflicting motions.
11.2. TRAVEL LIMITING DEVICES

Travel limiting devices are of two types: extreme position limiting devices and size maintenance devices. The extreme position limiting devices are so adjusted as to have the travelling machine unit stop 3 or 4 mm short of the dangerous end position. Hence, an accuracy of ± 0.5 to 1mm, and sometimes several millimeters, is sufficient for an extreme position limiting devices. The size maintaining limiting devices should limit the travel with considerably greater accuracy than the extreme position type, because they determine the accuracy of the machined work pieces.

Limit switches can stop travelling units with an accuracy of ± 0.02 or 0.03 mm when higher accuracy is required up to ± 0.001 mm, it will be necessary to resort to mechanical or combined electromechanical or Electro-hydro mechanical devices. The principle of the mechanical systems in precise travel limiting devices is that at definite point on its line of travel the movable unit of the machine tool meets a positive (dead) stop, secured to some stationary machine part. As a result the resistance to further motion increases to a point where the kinematics train of the drive to the travelling part is automatically disengaged.

According to figure 11.1 the slide 2 is stopped when it meets positive stop 1 and friction clutch 3 begins to slip. This continues until the slide is withdrawn from the stop

![Diagram](image-url)

*Fig.11.1 Devices with slipping (a) friction clutch (b) ratchet-tooth clutch*
Figure 11.2(a) shows the dropping worm system. The feed movement is transmitted to the moving unit by feed shaft 2 through gears $Z_1$-$Z_2$, shaft 3, a universal joint and shaft 4 on which worms 5 is freely mounted. This worm is connected with shaft 4 by means of safety over load clutch 6. When the unit reaches positive stop 1, worm wheel 9 and worm 5 stop rotating, torque rises and disengages the over load clutch. The movable clutch member shifts to the turning lever system 8, and cradle 7 together with drop worm5 through the pull of gravity, thus bringing the worm out of mesh with the worm wheel.

In a device shown schematically in figure 11.2(b) the rotation is transmitted to worm wheel 2 through gears $Z_1$-$Z_2$ and there by to the slide when the slide meets a positive stop, the worm wheel stops rotating, while worm1 continues to rotate, advancing by screw action in the worm wheel teeth to the right and turning bell crank lever 5 counter clockwise. Here spring 3 disengages clutch 4. The accuracy attainable by travel- limiting devices with drop worms is about 0.02-0.03 mm for idle motion and 0.15-0.2mm under load.

Out of the above described versions, the one incorporating a worm is the best since its operation involves the disengagement of components travelling at low speeds; they have little inertia and consequently only small over travel due to inertia.
The drawbacks of purely mechanical travel limiting device could be eliminated by using combined electromechanical arrangements. Some devices employ electromagnetic or thermal relay, which switch off the drive motor upon a sudden increase in current at the moment that the travelling parts of the machine tool run up against the positive stop. In others a limit switch is operated simultaneously with the disengagement of the clutch, dropping or shifting worm, to switch off the drive motor through a contactor.

The method of fastening the stop depends upon the construction of the part of the machine with which it is mounted. One of the pair of mating stops should be equipped with a micrometric screw or other similar part for setting the length of travel with greater accuracy. In Fig. 11.3, stop 5 is secured on the bed with strap clamp 6, which has teeth that enter the tooth spaces of rack 8. The two screws 7 draw together the clamp and stop, thereby fastening the stop rigidly to the bed. Micrometric screw 1 is supported in bushings 2 and 4. It is set up by turning nut 3 having a scale engraved on surface a. Screw 1 is restrained against rotation by a key which fits a keyway in bushing 2.

11.3. OVERLOAD PROTECTING DEVICES

Electric, hydraulic and mechanical protection devices are extensively employed. Electric protection devices and instantaneous action safety trip clutches are the most advance one.

In mechanical types following devices are generally used. Coefficient of sensitivity of average overloading safety unit should be 0,7 to 0,85. Coefficient is in relation to the load at which sensitive element of safety device start responding at its change to the maximum size of load \( Q_{\text{max}} \). Average shear pin or other element of safety device is sheared at 0,85 \( Q_{\text{max}} \) at steady action of load and 0,7 \( Q_{\text{max}} \) for varying load.

1. Unit with Shear Key
Key of such safety device is made from medium carbon steel, chromium alloy steel hardened upto 52-54. Some time copper and textolite are used for the shear key.

2. Unit with Shear Pin

Similar type of units can be joined by overloading coupling with one shear pin parallel to the axis of the shaft or with one or two shear pin fixed perpendicular or radial to the axis of the shaft.

In coupling with single shear pin (figure 11.4a) one half of coupling is mounted on the shaft with key, while the other half coupling is free on the shaft. On the elongated part of later half coupling driving part (pulley or gear) is mounted with key. Shear pin is mounted through bush made of EN 18 (carbon steel) hardened 50-60 HRC. Bushing is press fitted into holes in the joined component so that edge the hole is not distorted when pin is sheared. Part a keeps the shear pin from falling out.

Coupling with single shear pin is used in feed mechanism of vertical boring m/c. it is situated just outside the housing of feed gearbox so that shear pin can be changed with minimum waste of time.

The magnitude of the force required to shear the pin depends chiefly on the material of the pin, its heat treatment and its minimum diameter. This force can be varied in a sufficiently wide range by using pins with rectangular or Vee shaped necks of various diameters.

\[
\begin{align*}
\text{Diameter of shear pin} & \quad d = \frac{4T_s}{\pi R f_s} \\
\text{Or diameter of neck of pin} & \quad d = \frac{4T_s}{\pi R f_s}
\end{align*}
\]

Where  \( T_s \) = Torsional moment at which shear pin must shear. It should be 0.7 to 0.85 of \( T_w \) (\( T_w \) is the torsional moment of weakest link.)

\( f_s \) = Shear strength of material in Kg/cm²

\( R \) = distance at which shear pin is fitted from axis of the shaft.
3. Safety Clutches
Safety clutches slip by overload so that they disengage the corresponding kinematics, which they automatically restore as soon as the load drops to the normal value. It requires periodic adjustment or replacement of worn parts. There is no difference between safety clutch and any other clutches except that former do not have control component. Basically any clutch can be used as safety clutch if it is capable of self disengagement when the transmitted torque exceed certain maximum value. Jaw clutch, friction clutch and ball type clutches are most common.

*Fig. 11.4 a Typical arrangement for axial shear pin*

*Fig. 11.4 b Typical arrangement for radial shear pin*
**Safety jaw clutch**

Jaw clutches are frequently used as safety device and operate successfully if the angle of inclination of the sides of the jaws and the tension of the spring is properly selected. The clutch shown in the fig.11.5 a, consists of clutch members 2 and 5 with jaw-teeth having inclined sides and keyed on shafts 1 and 7 respectively. In transmitting torque within permissible limit clutch jaws are engaged by the action of coil spring 4. The pressure exerted by the spring along with the value of limiting torque is varied by adjusting sleeve 3 axially. Thrust bearing 6 is needed here because spring 4 bears on one end against clutch member 5 which is rigidly linked to shaft 7 and on other end against sleeve 3 which is linked thru clutch member 2 to shaft 1. Disengagement of the jaws in safety devices of this type requires axial movement of one of the members with jaws.

Safety jaw clutches is installed on the shaft, speed of which does not exceed 150-200 rpm. Experience shows that the friction resistance between the sliding half claw and its guiding key or splines may some times be so large that no axial motion takes place the clutch is not tripped.

In fig. 11.5 b, the upper half shows the normal clutch installed in table drive of heavy machines like plano-milling machines while lower half version shows how excessive friction can be eliminated.
The clutch links hub 3 from driver side, feed screw 1 of the table is to be driven. In the previous type construction shown in upper part hub 3 is freely mounted on a plain part of feed screw 1, clutch member 2 is keyed to the screw. In the revised construction shown below the centre line, intermediate clutch member 4 with jaws having inclined sides has been inserted between hub 3 and clutch member 2, which is keyed on the feed screw. The jaws of clutch member 4 engage on identical jaws on hub 3. Clutch members 2 and 4 are engaged together by five large square jaws with groundsides. In case of an overload, clutch member 4 slides easily to the left since its motion is not restricted by heavy friction on a key.

**Safety friction clutch**

Friction clutches are most commonly used for safety purpose in machine tools. These are similar in construction to ordinary friction clutches with control components. Service life of a safety clutch can be increased and the overloaded unit can be rapidly stopped by linking the clutch to some device for switching off the power. The same materials are used for making the components of safety friction clutches as for those of friction clutches intended for any other purpose. Safety clutch can work with lubricated as well as dry friction surfaces. Coefficient of senility of friction safety clutch is 1. Friction clutches may be either conical or disk type. Both single disk and multy disk type clutches are applied depending upon the load to be transmitted.

Friction disks of the clutches are manufactured from hardened steel pressed asbestos ferrode, cast iron etc. Best result is obtained when steel disk is coupled with that of asbestos. Coupling having disk with sheet from pressed asbestos or ferrode may work without lubrication. Accuracy of disk clutch is better than that of cone clutch. Single disk friction clutch is shown in figure 11.6. Here two shafts are joined thru two friction surfaces of the single disk coupling. A sleeve keyed on left side of the shaft is comprised of a flange, both sides of which have pressed asbestos sheet. Clutch body keyed on the other shaft and a ring freely mounted on the above-mentioned sleeve are joined together with the force of springs. Owing to the limitation of number of friction surfaces, such coupling cannot transmit much heavy torque. For heavy load transmission multi disk safety clutches are employed.
Multi Disk Safety Clutch

Multi disk clutches do not differ much from normal coupling joint. Gear is fixed on toothed sleeve, on which clutch body is mounted. Outer disks rotate along with the clutch body. Internal disks are seated on splines. At one end, thru gear shown in fig.11.7 bolts are provided to shift splined sleeve for regulating spring force of the safety clutch.

Ball Type Safety Clutch

A ball type safety clutch, shown in fig.11.8 operates on the same principle as jaw clutch from which it differs only in that balls have been used in place of jaws. The balls are retained by riveting over the edges of the sockets into which they are inserted. Safety clutches should be provided nearer to load. Position of the clutch should be such that it is easy to regulate, lubricate, and replace sheared or worn parts with spare ones.
Chapter 12: ELEMENTS OF CONTROL SYSTEM

12.1 INTRODUCTION

The operating features of a machine, like output convenience, ease of handling and reliability, depend largely on the efficiency of its control system. The requirements made on control systems are safety, ease and convenience of manipulation, fast operations, and mnemonic controls (coordination of the direction of hand motion with the direction of the controlled unit travel) accuracy of operation. The two main functions of machine control systems are
1. Changing speeds and feeds
2. Providing the working and auxiliary motions in the desired sequence

Machine control systems generally include.
(a) A control member, actuated by the operators hand or foot e.g. handle, lever, hand wheel, push button, limit switch etc.
(b) A transmitting member in the form of mechanical, electrical, electronic, hydraulic and pneumatic devices.
(c) An operative member, like shifting fork, lever, rack etc.

12.2 MECHANICAL CONTROL FOR CHANGING SPEEDS & FEEDS

Controls could be manual or automatic depending upon the frequency of changing operations required. In manual control, multi-handle or single lever control systems are generally applied.

Levers, racks and screws are used for multi lever system while cam link motion and Geneva wheel mechanisms are used for single lever control system. In single lever control system the same rack pinion is made to mesh with several racks, this exclude the need for special interlocking elements.

Multi Handle Control

Figure 12.1 shows a multi handle control system with all the handles mounted on a common axis. These handles 1, 2, 3 operate gear clutches 6 through sectors 4 meshing with racks.
Transmitting with a screw and nut is especially convenient for accurate motion. In combination with high ratio reduction gearing, a screw and nut can be employed for making very small hand operated motions. With the help of screw drive and nut a large force can be produced to shift heavy gears etc.

![Screw type lever control system](image)

**Fig.12.2 Screw type lever control system**

**Single Lever Control**

Single lever control can be divided into two main groups.

- Permanent linkages between the controlling member and controlled parts- widely employed in such trains are cylinders (drum) and plate cams, link motion drives, Geneva wheel mechanism as well as hydraulic, pneumatic and elector-hydraulic devices.

- System in which the same controlling member is linked to several different control-trains. The controlling members in this case a lever or hand wheel, which is shifted, along constant centre of swivel etc.

![Typical single lever control mechanism](image)

**Fig.12.3 Typical single lever control mechanism**
The principle of single lever control system can be understood with the figure 12.3. When handle 3 is pivoted in the horizontal plane in any direction, pinion 10 turning together with shaft 4 will displace rack 11 and there by gear cluster 12 along shaft 9 to one of the cluster’s three corresponding positions. When pivoted in the vertical plane about pin 1, handle 3 moves shaft 4 axially up and down, circular rack 8 turns wheel 7 with shaft 6 and shift fork 14, thus moving gear cluster 13 along shaft 5 to one of its two positions. When handle 3 is outside of the vertical slots in shield, both gear clusters 12 and 13 will be in neutral positions.

Single Lever with Cam-drum arrangement

With the mechanism shown in figure12.4, 8 different speeds can be obtained with the help of single lever. With the help of hand wheel motion is transmitted to 3 cam drums. Each cam drum imparts motion to a follower attached to shifter for shifting a double cluster gear. Cam drum I shifts its cluster gear for 1/8th rotation of cam while drum II shifts its double cluster gear for 1/4th rotation and cam drum III for 1/2 rotation. In same cases two-cam drum are rotated with the help of Geneva wheel mechanism in place of begin mounted on the shaft.

Plate cams are employed for speed changing of lathe M/C. In comparison with cylinder cams plate or disc, cams with a positive return feature have the advantage of occupying less space due to their small thickness.

![Fig.12.4 Single lever system with cam drums](image)

Single Lever with Sliding Key System

This is the compact arrangement in which all the gears are permanently in mesh. A key slides with the help of a lever and transmit motion thru the pair of gears in which the key is fastened during shifting.

![Fig.12.5 Sliding key arrangement](image)
**Selective Speed and Feed Changing System**

Normally speed changing system is so devised that it becomes necessary to pass thru all the intermediate speed in changing over from one speed to any other speed. This consumes large time and causes excessive wear of the gear teeth at their ends or wear of jaws in case of jaw clutches. These drawbacks are eliminated in selective speed-changing system. The system shown in fig.12.6 has a single selector single disk. It has a large number of rack pushers by means of which four double-cluster gears 1, 2, 3 and 4, are shifted to obtain 4x2x2=16 speeds. To engage one of these speeds, selector disk is pulled out of rack-pushers by means of lever having pinion sector meshing with rack integrated with the disk. Then the disk is turned to the position in which an index will point to the required speed and pushed forward as far s it will go. At this the selector disk pushes forward the corresponding pair of pusher racks and the forks of levers to the required positions. Electric arrangement is also provided to inch drive-motor facilitating engagement as the gears slide into mesh.

![Fig.12.6 Single disk selective speed changing mechanism](image)

**Preselection of speeds and feeds**

To reduce the time needed for changeover, the system makes it possible to set up the speed or feed for the next operation while the previous operation is still in progress. After the completion of the operation, the pre-set speed or feed is engaged by a single movement of the control lever or by pressing the push button.

Mechanism of mechanical control for preselection of speeds is shown in figure12.7. While machine is running, revolving selector disc 1 with speed steps marked on it is turned and set into position according to the speed required for the next operation. By revolving selector disc, position of contoured cams 5, mounted on shaft 6, can be changed. Before starting the next operation, handle 2 is shifted, which displaces cams 5 through corresponding gears and circular racks 8. Cams 5 by their end face projections move lever 3 engaging friction clutch 4. One of the levers 3 is shown in the figure. During the speed preselection cams 5, being retracted by spring 7, do not touch lever3. Certain position of cams 5 corresponds to each speed step.
In the system shown in Fig.12.8, first of all lever 1 is turned. At this part 2 shifts fork 3 and 11 along rod 4 and 10 in opposite directions. These forks enter annular grooves of contoured cams 5 and 12 mounted on shafts 6 (on splines) which is turned with a hand wheel. The speed is changed by shifting lever 1 back in the opposite direction. This disengages the main friction clutch, slowing down speed of shafts and then cams 5 and 12, brought together again, actuate with their lobes the pins of lever 7 and 8. This turns levers 7 and 8 about the axis of pin 9 so that the forks at the ends of the levers shift the corresponding cluster gears to their new positions.

Hydraulic preselection system for changing speeds and feeds- system employed in drilling M/C is shown in the figure 12.9. Pump delivers oil to the accumulator, when it is filled with the necessary volume of oil, a port through which oil under a pressure of 10 to 12 Kg/cm$^2$ passes for lubricating bearings and gears.

By turning hand wheels 6, 7 through bevel gears 8 and 9 internal sleeve of preselector valve 5 and 10 are turned. At this the pressure chamber of these valves are connected to the upper and lower ends of the two position cylinder 4 and three position cylinder 11 whose pistons are linked rigidly with the levers of forks that shift the cluster gears. As long as main valve 3 is closed and the ends of the actuating cylinders are not getting under pressure, all the cluster gears remain in their previous positions.
The cluster gears are shifting as soon as valve 3 is connected to the hydraulic control system by shifting lever 12, which also engages the multiple disk clutch of the drill drive. When lever 12 is taken out of the socket, pinion mounted at the rod end will shift the piston of the valve 3 giving oil pressure to the brake ensuring stopping of rotation of the gears. When lever is taken to upward or downward position to engage forward or reverse clutch, valve supply oil pressure to the pre-selector valves 5 and 10. Hydraulic cushioning cylinder is attached to the clutch-engaging rod, which slows down the speed of engagement of clutch. In such situation gears can rotate at less torque and avoid tooth abutting. Thus gears are properly meshed as per selected speed, by the time, clutch is fully engaged.

![Diagram](image)

*Fig.12.9 Hydraulic system for speed pre-selection*

**Requirements of automatic speed changing mechanism**

1. In all automatic speed changing systems, it is impossible to shift the gears if ends of the gear teeth run up against the teeth of the mating gears. So it is necessary to transmit a slight turning motion to the gearbox shafts. For this an arrangement for initial engaging of the clutch at low torque should be provided.
2. Automatic speed changing mechanism should ensure disengagement of the main motion (clutch etc) while speed is bearing changed.
3. Some provision for locking the gears in position should be ensured. Electrical or hydraulic interlocking is provided so that either main does not start at normal load or clutch is not engaged fully if gears are not locked fully in position.
4. Inter locking should be provided so that gears could not be changed while clutch is engaged.
12.3. AUTOMATIC CONTROL SYSTEM

Automatic system could be provided to control following functions and other operations.

(a) Displacement of slide-members.
(b) Angular rotation of circular tables.
(c) Stop/start the main spindle
(d) Change of spindle speed.
(e) Reversing of spindle.
(f) Change of feed rate.
(g) Rotating tool turret.
(h) Changing of tools.
(i) Cutting fluid on/off.
(j) Indexing.
(k) Clamping/unclamping of slide member.
(l) Work piece handling such as pallet changer, job gripper etc.
(m) Continuous process inspection and correction of production error due to tool wear etc.
(n) Diagnostics
(o) Services scarf guards, chip conveyor etc.

These functions are monitored by various mechanical, electrical and electronic devices for their initiation and completion like by trip dogs operating electric, pneumatic; or hydraulic switch adjustable and fitted along side or behind each other on guides or shaft or an auxiliary trip drums. Such drums rotate as a result of position gear drives connected to machining cycle. In most of the complex automatic cycles for example: rapid approach-working feed-rapid withdrawal of table could be achieved by switching over to the different speeds and directions on the command initiated by limit switches struck with trip dogs.

In the fig.12.10a clutch is engaged by electromagnet and disengaged by the spring. The electromagnet is energized and deenergised with the help of the limit switches 3. Further the clutch start and stop the movement of the table. Direction of movement in a hydraulic and pneumatic mechanism could be reversed by reversing the direction of flow of the fluid to the cylinder/motor 5 with the help of the reversing valve 4 shown in the fig.12.10b.

In case switching over clutch-unit require three positions, drive with double acting electromagnet and special piston drive as shown in the fig.12.10c could be applied. The drive consists of two cylinders 6 and 10, piston that provides movement to the fork for switching over of clutches. Diameter of piston 7 is greater than the piston 9 and both are connected with piston rod 8 fitted with switching over fork. The fork comes to the middle position when pressure is delivered to both the cylinders and on either side depending upon whether oil pressure is in cylinder 6 or 10.
For monitoring operational condition and machine loading, strain gauge is used to measure torsional deflection of spindle. Strain gauge is difficult to mount on spindle. For torque measurement force transmission is arranged through an intermediate shaft using helical gears. Then thrust bearing of this shaft will be subjected to an axial force proportional to the torque. Axial force may be measured by stain gauge fixed on drum.

Automatic control mechanism includes three principal members

A) Command signal with corresponding transducer.

The signal may be analogue type or discrete type. In the former case signal is sent continuously and in the later one it is periodic. According to the input signals, transducers can be classified as:

(i) Travel transducer-limit switch, proximity switches hydraulic slide valves and pilot valves etc.
(ii) Sizing transducer-for in process gauging of work e.g. electric control type, photoelectric type.
(iii) Load transducers-designed to generate command impulse as the movement load reaches a certain value, such as electrical relay, pressure relay etc.
(iv) Velocity transducers- inductive, centrifugal or tacho-generator type.

B) Amplifiers: Electrical, Electronics, hydraulic, pneumatic

Amplifiers and mechanical lever, wedge eccentric etc. come under this category.

C) Operative-members

Servomotors are used to transformer input command signal into mechanical motion. Following types of motors can be employed to act as Servomotor.

i) AC electric motor-induction motors with a cage rotor and two stator windings (Control windings and exciter windings).
ii) D.C.electric motor.
iii) Stepper motor
iv) Hydraulic servomotors.
v) Pneumatic motor

Clutches such as electro magnetic particle clutches are also some times used in the operating system.
12.4 LOGICAL SWITCHING CONTROL

AND, OR, NOT, NOR, NAND gate functions are applied in logical switching control. This is also termed as sequence control. Plug board control provide the opportunity to product a hard wire control which can be reprogrammed by the use of plug. Plug board consists of a series of vertical and horizontal connector bars. The vertical bars correspond to the functions of m/c tool which are to be activated if movement of slides, feed rates. The horizontal bars correspond to the steps in the operations and carry the supply voltage in a stepped arrangement. The bars are connected by switches at the inter selections with which the particular function required by the work, program is selected. The diodes are used to disconnect the bars. Using coded plugs upto eight functions may be programmed for each step.

The progression to the next step occurs after a feed back signal has been received from the machine tool, using for example limit switch on the slides, activated by trip dog.

For controlling the movement of traversing elements following methods is adopted:

i) **Programmed guide control:** On the sliding member itself strikers are fitted as per the desired program of the sliding elements. These strikers actuate electrical, hydraulic, pneumatic switch or mechanical elements to give desired signal to the operative members.

ii) **Active control:** Size monitoring arrangement continuously check the dimension of the work piece being manufactured and give signal accordingly to the feed drive proximity switch that could be applied very efficiently for this purpose.

iii) **Functional sequence control:** As per the sequence of the operation, program could be set directly in the machine by means of tumbler switches, plug switch boards, multiple position selector switches etc.

12.5 PROGRAMMABLE LOGIC CONTROLS

Logical processing of signals is achieved with the aid of relays in electrical circuits, whereas in electronic circuits integrated circuits (IC), integrated logic device, micro processor or process computers are employed. The system is normally called as PLC. PLCs perform open loop control tasks such as interlocking sequencing etc. It can also cater to additional requirements with regard to closed loop controls such as positioning and man machine interfaces. These additional tasks are carried out by intelligent modules equipped with their own microprocessors signals are transmitted between input/output modules and CPU .The distributed processing in the main CPU of the PLC and the intelligent modules allow for high processing speeds and short cycle time. In case of simple machine the distribution could be only of the input/output modules. In complex machine, it could also be a decentralization of the functions and intelligence improving; the availability of overall system and shortening the response time. PLC has replaced many of the relay control panels formerly employed in industry.

PLC’s were basically introduced as replacement for hardwired relay control panels. They were developed to be reprogrammed without hardware changes when requirements were altered and thus are reusable. PLC’s are now available with increased functions, more memory and large input/output capabilities. Fig.12.11 gives the generalized PLC block diagram.

In the CPU, all the decisions are made relative to controlling a machine or a process. The CPU receives input data, performs logical decisions based upon stored programs and drives the outputs. Connections to a computer for hierarchical control are done via the CPU.
The I/O structure of the PLC’s is one of their major strengths. The inputs can be push buttons, limit switches, relay contacts, analog sensors, selector switches, proximity switches, float switches, etc. The outputs can be motor starters, solenoid valves, position valves, relay coils, indicator lights, LED displays, etc.

The field devices are typically selected, supplied and installed by the machine tool builder or the end user. The voltage level of the field devices thus normally determines the type of I/O. So, power to actuate these devices must also be supplied external to the PLC. The PLC power supply is designated and rated only to operate the internal portions of the I/O structures.

![Block diagram of PLC](image)

### 12.6 NUMERICAL CONTROL SYSTEM

The system, in which program of motions of machine numbers represents a numbers, is called numerical control system. The input information for controlling the machine is provided thru numerically coded instructions in program input form or other type of forms. The coded instructions are expressed not only thru numerals, but also thru letters etc. symbols. Depending upon architecture of the control numerical control system could be classified into two categories.

**a) Hard wired or conventional NC:** - In the hard wired system the entries data input and data handling sequence including control functions are determined only by the fixed circuit inter connections of decision elements and storage devices. These control systems use combinations of digital and analog circuit elements, each one performing specific functions allocated permanently for it. The logic functions are pre-engineered and fixed. Any modifications required subsequently necessitate wiring changes along with addition or deletions of one or more circuit elements.

**b) Computerized Numerical control (CNC):** - In CNC system a small computer is used to perform all the basic NC functions as per the control programme, stored in the memory of computer, called executive programme. The machine control data comes direct from computer memory and not from the continuously read tape as in case of hard-wired NC system. Thus the control logic for the machine tool is generated by software programme rather than by wired logic circuits. Changing the software of the system can change the machine functions. The system provides for

1. Rate and motion control
2. Circular interpolation
3. Cutter radius, tool off set and tool radius compensation.
4. Part programme storage, source language editing facility
5. Automatic compensation for machine tool inaccuracy
6. Fault finding assistance for logical and sensory circuits.
7. Process option
CNC system could be of two types

i) Open loop: - The open loop system lacks feedback. In this system, the CNC system send out signals for movement but does not check whether actual movement is taking place or not. Stepper motors are used for actual movement and the electronics of these stepper motors is run on digital pulses from the CNC system. Since system controllers have no access to any real time information about the system performance, they cannot counteract disturbances appearing during the operation. They can be utilized in point-to-point system, where loading torque on the axial motor is low and almost constant. It is used to position the axes accurately in the case of large milling machines, boring machine, vertical borer etc.

ii) Closed loop: - A closed-loop system, regardless of the type of feedback device, will constantly try to achieve and maintain a given position by self-correcting. As the slide of the machine tool moves, its movement is fed back to the CNC system for determining the position of the slide to decide how much is yet to be travelled and also to decide whether the movement is as per the commanded rate. If the actual rate is not as per the required rate, the system tries to correct it. In case this is not possible, the system declares fault and initiates action for disabling the drives and if necessary, switches off the machine.

In case no time constraint is put on the system to reach the final programmed position, then the system may not produce the required path or the surface finish accuracy. Hence, velocity feedback must be present along with the position feedback whenever CNC system are used for contouring, in order to produce correct interpolation and also specified acceleration and deceleration velocities. The tacho generator used for velocity feedback is normally connected to the motor and it rotates whenever the motor rotates, thus giving an analog output proportional to the speed of motor. The analog voltage is taken as speed feedback by the servo-controller and swift action is taken by the controller to maintain the speed of the motor within the required limits.

To achieve the special machining features like contouring threading etc. fully closed loop technique is required. L A closed-loop system, regardless of the type of feedback device, will constantly try to achieve and maintain a given position by self-correcting. As the slide of the machine tool moves, its movement is fed back to the CNC system for determining the position of the slide to decide how much is yet to be travelled and also to decide whether the movement is as per the commanded rate. If the actual rate is not as per the required rate, the system tries to correct it. In case this is not possible, the system declares fault and initiates action for disabling the drives and if necessary, switches off the machine.

Lathes, grinders and milling machines are generally retrofitted with this system. This involves replacing of existing lead screw with ball screws, putting individual axis high performance servo-drives, centralized lubrication, feed back transducers for individual axis etc.
Multi-Processor Control (MPC)

Data processing operations with in a CNC unit are subdivided into independent functional blocks to such a degree that there is minimal need for data interchange between them. Each functional block is then allocated an individual microcomputer board. The jointly perform the complete control function by bearing connected through a data transmission channel. Advantage is that the individual existing modules may be replaced with more effective version.
Chapter 13: LUBRICATION

13.1 INTRODUCTION

Whenever there is a relative motion between two surfaces, friction force start acting in a direction opposite to that of the motion. The frictional force is maximum when the object just start moving and this is known as ‘limiting friction’. There is always resistance to movement when two surfaces slide or roll against each other. This resistance is known as Friction. Basically the friction is of two types:
Sliding friction
Rolling friction

The friction force is due to the resistance offered by the interaction of the projections on both the surfaces. The coefficient of rolling friction is much less than sliding friction hence lubrication is very necessary in case of sliding surfaces as compared to rolling surfaces. The friction is both blessing and a curse. Without it, one cannot walk, motorcars cannot run, and screws cannot hold. While on the other side, any machine from sewing to aircraft, from watch to automobile would come to grinding halt, if friction is not overcome.

During the process of sliding, very high local temperature is generated and chances of tearing of metal surface become high. Rapid cooling follows local melting of the projections and the ultimate result is that a very highly polished extremely hard layer is formed. It is also known as ‘running in’ process. This could be facilitated by successful lubrication.

13.2 LUBRICATION DEVICES

Lubricants could be applied between the mating surfaces with the help of the several devices. Most common out of the same is manual lubrication and splash lubrication.

MANUAL LUBRICATION

Simplest method of applying lubricant to a bearing point is from an oilcan through a small hole in the top of the bearing. Piston type oil can be used for feeding more quantity of oil by pressing handle knob, which moves the piston throwing out oil through valve to nose of oilcan. Due to spring force, piston returns to its initial position.

For filling pipes with lubricants dose under pressure, oil guns and grease guns are used at lubricating nipples fitted in the machines. Lubricating nipples contain a non-return valve, usually a ball and spring unit to prevent exudation of lubricant or ingress of contaminants. Nipples are of various designs e.g.

Hydraulic 2) Button head 3) Lubricating plug 4) Bayonet 5) Push on.

Normally ‘hydraulic ‘ and 'Button head ‘ types are most commonly used. In the hydraulic nipple the head projects beyond the non-return valve. In the button head type the ball project slightly over the nipple head. The oil gun and grease guns should be fitted with the suitable connectors matching with the nipples. Oil gun can deliver 0.3 - 0.5 cm³ of oil at 30Kg/cm² pressure in one or two strokes of plunger, while grease gun can deliver 1cm³ grease per stroke of piston.
SPLASH LUBRICATION

The simplest form of lubrication is that in which part to be lubricated are completely immersed or partially submerged in oil. Distribution of oil is effected by splash. Basically this system has a built in reservoir where clean oil is filled. The machine gears, chains and other drives partially remain in the oil and splash the oil to the upper part of the machine. This is simplest way of foolproof lubrication particularly for gearboxes of the machines, automobiles and various mills.

Fig. 13.1 Manual lubricating nipples

Gears, chains and thrust collars running at comparatively low speeds may be lubricated simply by dipping into the oil bath. At moderately high speed (upto 45 rpm) amount of oil thrown up as splash or spray become sufficient to be used for the lubrication of upper bearings and gears.
Generally for upper gears and bearings a trough or pans are arranged in which the oil thus thrown is retained or collected through pump and redistributed. Some time the oil is circulated by a pump, which may be driven from a part of the machine, or by a separate electric motor. This pressure circulating oil system is necessary where peripheral speed of gears is too high. Oil supply to spray is usually at 1Kg/cm².

In oil bath lubrication or splash lubrication system, it is most important to maintain the correct oil level. Too high a level is likely to cause excessive churning and agitation, resulting in excessively high operating temperature. In slow running gears, oils should kept at higher level to ensure that enough oil is carried up from the bath to the zone of mesh. In fast running gears a small depth of immersion may suffice for generation of spray at the required rate. The slower medium speed gears may require immersion to two or three times the height of the teeth while in high speed gears only half way up the teeth.

RING & CHAIN OILER

Journal bearings may be ring oiled, that is they have a ring, which resets on the journal and dips into an oil sumps. The ring rotates freely with journal and in so doing carries oil from the sump on to the journal. Maximum depth (t) of oil ring submerged in oil may be calculated by the following formula.

\[
\begin{align*}
\text{t} &= \frac{D}{4} \quad \text{when } D = 25 \text{ to } 40\text{mm}. \\
\text{t} &= \frac{D}{5} \quad \text{when } D = 45 \text{ to } 65\text{mm}. \\
\text{t} &= \frac{D}{6} \quad \text{when } D = 70 \text{ to } 310\text{mm}. \\
\end{align*}
\]

(Where D is internal diameter of ring.)

Quantity of oil, supplied by ring, decreases with the increase of temperature of oil in reservoir. Freely hanging ring assures oil supply from 2 to 10 cm³ oil in a minute. If bearing length is 1.5 times more than its diameter, place two rings. Internal surface of the ring must have groove to give better result.

Oil carrying capacity of chain is higher than that of a ring. But chain should not be used for higher speed as this reduces the arc of contact between the chain & shaft.

BOTTLE OILERS

A bottle oilier consists of an inverted bottle with a metal base containing a closely fitted movable pin. The pin rides slightly on the shaft. As the shaft rotates the pin is vibrated causing a slight pumping action and a supply of oil is carried down to the journal but so long as the shaft remain rotating. The rate of oil feed is determined by the clearance between pin and its sleeve.

DROP FEED LUBRICATOR

The drop feed lubricator generally comprises a glass, plastic or metal cylinder mounted on a metal base containing on orifice fitted with a needle. The rate of oil supply is controlled by the position of the needle. The rate of oil supply is controlled by the position of the needle when the valve is open. The drip feed can be cut-off by moving the small lever at the top of lubricator through 90. The supply of oil is replenished through a hole in the cover. Most of these lubricators are fitted with a sight glass underneath the reservoir.
WICK FEED LUBRICATOR

The wick feed lubrication can be of different shape. In the stationary positions" tail siphon" type is more common. In this a pipe is extend above the oil level. A wick made of one or more strands of wool yarn, is threaded through the eye of a hook by which part of the wick is held in the pipe while bulk of it remain in the oil reservoir. Through a combination of a capillary and siphonic action oil flows along the wick and drips down the central tube. The greater the number of strands yarn used, the greater the rate of oil feed. A wick consisting of 25 cotton threads can ensure oil feed up to $1.5\text{cm}^3/\text{hour}$. For wick feed lubrication oil level in the tank should not exceed 50mm. A shallow form of tank is preferred to minimize the effect of variation of head of oil.

The wick act as filters and should therefore be removed periodically and either replaced with new ones or washed in white spirits. Best material for wick is undyed unprocessed wool knitting yarn.

FELT PAD LUBRICATION

Felt pad lubrication is maintained in contact with shaft. The pad is charged with oil with the help of a dust proof oil cap after a month or so.

In some cases the felt pad or cotton waste packets pick up the oil from the pit or well provided below the shaft and transport it to the journal. This type of lubrication is applied for bush bearing with speed upto 4meter/sec.

ROLLING IN OIL BATH

This system is generally applied to lubricate friction pair e.g. slideway. Small oil packets are provided in the guideways. In the pocket rollers are submerged in oil and suspended with the help of spring strips. The rollers rotate with the movement in the slideways and lubricate it. Thus uninterrupted supply of oil could be maintained to the mating surfaces.

LUBRICATION WITH SMALL PUMPS

For uninterrupted supply of oil under pressure to the larger size of mating pairs small size plunger pumps, blade pumps and gears pump are used. These pumps can be driven with the help of cam in case of plunger pump or by gear of gearbox unit of the machine itself.

In the case of cam driven plunger pump, spring loaded piston is activated. The piston forces lubricants through the oil outlet valve. The spring returns the piston and oil is sucked through a check valve preparing the device for next delivery. In some cases operating lever is provided between cam and spring loaded piston. Stroke of the lever could be adjusted to vary the return stroke of the piston. Thus quantity of oil to be dispensed in each cycle could be controlled as per requirement.

13.3 PRESSURE CIRCULATING SYSTEM

For relativity large oil quantities per lubrications point and continuous oil supply, this system is used. This system is meant for circulating oil from the reservoir to the machine point and after lubrication of the machine bearings/gears the same oil returns back to the tank. The returning dirty oil is first filtered and cooled. Normally theses systems are made from 30 l to 10,000 l reservoir capacity depending upon the application. The applications of oil circulating systems are in paper industries: steel, sugar and cement plants; turbine generators; rubber mixing machines; and high power electric drives.
Oil returns to the sump either by gravity or through a return line. The quantity of oil delivered by the pump is continuously distributed to the individual lubrication points, either directly through distributors or through a tank at the top. In the later case oil is fed due to gravity and pump failures will not cause immediate starved condition at lubrication points.

**13.4 CENTRALISED DISTRIBUTOR SYSTEM**

In centralized system number of lubrication points can be serviced from one place with the help of hand pumps or intermittently operated piston pumps or gear pumps. Generally oil gets lost in this system. So it is also sometimes termed as ‘total loss system’

**Single shot Oil Lubrication System:**

In this system, the oil is fed to the various points of the machine from the central source, i.e., pump. The main advantage of this system is the depending upon the bearing area; the oil can be put cyclically for optimum lubrication. This type of system is used in: metal cutting, metal forming, injection/blow moulding, die-casting, packing, and printing machines.

**Single-line Progressive Lubrication System:**

This is a progressive system meant for both oil and grease and here the pump source feeds the lubricant to a flow divider, i.e., flow the lubricant to ports one after the other and if there is malfunctioning of one pin because of choking, the whole system becomes inoperative. This can be utilized for monitoring the malfunctioning. This system is suitable for: calendaring, earthmoving, die casting, and tyre-making machines, cement plants, sugar machinery, and vehicles.

**Multi-line Lubrication System**

This type of system normally has a multi-plunger lubricator where the reservoir houses the grease /oil. The multi-plunger pumps are put on the periphery of the housing below the reservoir. An eccentric cam operates the plunger pumps and grease /oil is fed to different points on the machine independently. This system normally runs through motor. The adjustment of grease/oil is possible independently. Additional features like grease level sensing, filters, pressure sensing are required as per application. It is suitable for forging hammers; presses; petrochemical plants; sugar, steel, and cements plants; earth-moving equipment; crushers, etc.

**Dual-line Grease Lubrication System**

This system is basically used for heavy-duty machines and plant. It is a very reliable system, has a very high-pressure pump, which feeds grease to various does feeders. Here the adjustment of does is possible from outside for individual points. Steels and cement plants; turbine generators; earthmoving equipment; furnaces; and coal and mineral machinery can make use of this system.
Mist Lubrication System

Mist lubrication system is particularly useful when oil and grease lubrication systems cannot be used. In this system, a pneumatic line is put on the reservoir and discharges drops into the atomizer. In the atomizer, a fog/air spray is formed which is made to fall on the required machine point. This system is available in both versions; intermittent as well as continuous. Its applications are in; machine tool spindles, girth gears, conveyors, chains, Railway wheel flange, moulding dies, etc.

All such systems have some basic layout comprising of following components.

- Pumps
- Distributors
- Delivery tubes
- Lubrication points

In the system with automatic control following additional items are also provided:

- Timer (to decide frequency of lubrication)
- Pressure switch / flow switch.
- Float switch.
- Warning device (Indicating lamp or other signal)

PUMPS

Several types of pumps are applied depending upon the requirement of the system. For manual lubrication hand pumps are used; for automatic system following pumps are more commonly used.

**Multi plunger pump:** These are the multiple delivery pumps. A battery of elementary reciprocating oil pumps is mounted in a line or in a circle. These pumps are actuated by disc cam or other eccentric device. During periods of operation each pumping unit delivers oil continuously at slow rate. Quantity of oil to be delivered can be controlled by regulating the throw of the plungers. Oil distributions with multi plunger pumps are of very simply design as oil delivery lines are not pressurized.

**Hydraulically and pneumatically operated piston pumps:** These pumps are well suited for total loss lubrication system and are generally equipped with residual pressure release valves, essential for the proper functioning of the distributors. In this system, the pressure in the main line between pump and distributors will be automatically released on return of the piston. The system can be controlled by the timer or counter. When the pump is actuated, the dosed oil quantities are delivered through pipes to the lubrication points.

**Gears pumps units:** The unit being discussed here is used in modern machine tools for centralized lubrication system utilizing distributors. These units are generally supplied with residual pressure release valve and relief valve.

The distributors require intermittent operations i.e. when the pump operates; the distributors deliver oil to the lubrication points and when the pump stops the distributors are refilled. This work cycle is achieved by switching on & off the electric motor by means of electrical control equipment at pre-determined intervals.
The Hydraulic circuit of the in built gear pump unit operates as follows. The oil is sucked and delivered through check valve. Pressure relief valve controls the pressure of the circuit and excess oil is drained. Bleed off valve is provided just after check valve to bleed oil to the drain, if oil is mixed with air and sufficient pressure is not generated. In the delivery line residual pressure valves is provided. This valve maintains certain pressure of oil in the delivery line draining extra oil when pump is in stopped condition. This pressure release system is required for the correct operation of the piston distributors.

DISTRIBUTORS

A centralized distributing system provides combined timed and measured lubrication. Commonly applied distributors are being discussed below:-

Spring Return Metering Valves: The pump supplies pressurized lubricant. From the main line pressurized lubricant enters the valves and causes the piston to move forward so that oil in front is pushed to the bearing point (Lubrication point). At the same time valve cavity is filled with lubricant. On releases of main line pressure, the piston of the valve returns to its initial position allowing the oil to flow into the space in front of it. Thus the compression spring brings the piston to its initial position to recycle the valve.

The spring return metering valves are connected in parallel. A pump delivers oil through the main line to the piston distributors. From the distributors the oil is delivered in metered quantities to the lubrication points. If one lubrication point is blocked or broken all other lubrication points are, nevertheless, remained supplied with the predetermined quantity of oil.

Fig.13.2 Lubricating pump unit
Piping and distributors should be arranged in such a way that any air if entered in the lubrication system may escape via lubrication points. For this purpose, the distributors should be fitted at suitable positions and at the end of the system with the outlets to the lubrication points pointing upwards. It should be ensured that main pipeline should remain rising from pump to the distributors.

![Fig. 13.3 Metering valves](image1)

**b) Dual Line Distributors:** These valves are used in two line systems. From the pump oil is supplied to a four-way valve. Two supply lines run from the four way slide valves to the metering valves.

![Fig. 13.4 Dual line distributor](image2)
When pressurized lubricant enters the metering valves from the supply line on the left, the supply line at the right being vented and the pressure is exhausted through non return valve to the vent side at the end of metering piston movement and the metered quantities of lubricants is delivered to the lubrication point through passage 'A'. Similarly when the right side supply line is pressurized the metered quantity of lubricant is delivered through passage 'B' to the lubricating point.

Another type of metering valve used with two lines system is shown in the figure. This valve can be used for both oil & grease, when line 4 is pressurized the lubricant enters on the side 12 of the metering piston 13 and the metered quantity of lubricant is delivered from side 14 of the piston to the lubricant point. Similarly when line 5 is pressurized, metered quantity of lubricant is delivered from side 12 of the piston to the lubricating point. Here flexible valve 7 plays an important role of non-return valve cum slide valves.

![Fig. 13.5 Working of multi-purpose metering valve](image)

c) System Using Flow Resistors: Another method of distribution is to impose flow resistors in each outlet from a branched system. Commonly the number of points observed by a system of this kind is of the order of 50 and more. The flow resistors employed to apportion supply of oil to each point of lubrication may be based on flow of oil an annular channel between a metering pin and its bore, or may be based on flow in capillaries. The pin type is more effective in pressure impulse cyclic system while, the capillary type is more suited to continuous system.

(i) The metering pin floats in an accurately cut bore, the diameter of which in relation to that of pin determines the amount of oil able to pass to the points of applications during each delivery cycle. The check valve is reseated under spring loading when delivery ceases.

(ii) In capillary type, resistance to flow depends upon the length of a capillary. Hence the shorter the capillary, the greater the amount of oil it will pass under given conditions of temperature and pressure.
OPERATING FUNCTIONS OF AUTOMATIC GENERALIZED LUBRICATION SYSTEM

Where complete automatic system is installed, electro-hydraulic system operates in the following manner.

With switching on of the machine, pre-lubrication is initiated. The pump starts running and delivers oil in predetermined quantities through distributors to lubrication points. As soon as the necessary oil pressure is reached, the pressure switch installed at the end point of main line starts the time delay mechanism in the main timer. After the preset time delay has elapsed, the pump is switched off and distributors return to initial position for recycling. Further lubrication cycle can now be repeated by the timer, which has returned to the initial setting according to the working cycle determined by the interval time set at the second timer.

MONITORING: As the piston distributors are very reliable it will generally be sufficient to monitor only the main line by ensuring that the system is pressurized at the desired interval. A float switch is provided to monitor the level of oil in the reservoir. Further control can be provided such as a counter mechanism actuated by a single hydraulic piston during every lubrication cycle to ascertain frequency of lubrication. The same effect can be achieved by a pressure switch and electronic counter. The oil flow between the distributors and the individual lubrication points may be monitored if the output exceeds 0.2 cm$^3$ per stroke.

13.5 LUBRICATION FAILURE

Though lubrication is meant for smooth operation of the machinery and to prevent their failure, lubricant itself may fail sometimes. The causes are: insufficient and excessive lubrication, chemical decomposition, contamination by dust, dirt, water, etc. however if care is taken to follow proper lubrication schedule, the better selection of right kind of oil or grease, and periodic maintenance, these could be prevented to a large extent.

CHECKING CONDITION OF LUBRICANTS

Although, for deciding the change period for small lubrication systems, laboratory tests are not economically justified and generally decided on experience, but testing should be carried out to find when the lubricant is approaching the end of its useful service life. For this, a combination of visual examination and laboratory analysis is suggested. For visual examination, a simple (about 100 ml.) is taken in a clear glass bottle. If dirty or opaque, The sample is kept at 60C for one hour. The radiator of an office air-conditioning machine or the exhaust of a light engine may provide the necessary heat source.
Table 5. Visual examination of oil

<table>
<thead>
<tr>
<th>Appearance</th>
<th>Reason</th>
<th>System without filter</th>
<th>System filter with</th>
</tr>
</thead>
<tbody>
<tr>
<td>Initial</td>
<td>After 1 Hr.</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Opaque</td>
<td>Clear</td>
<td>Foaming</td>
<td>Find cause</td>
</tr>
<tr>
<td>-do-</td>
<td>Clear</td>
<td>Unstable emulsion</td>
<td>Run-off water or sludge</td>
</tr>
<tr>
<td>-do-</td>
<td>No change</td>
<td>Stable emulsion</td>
<td>Send analysis for clear change oil</td>
</tr>
<tr>
<td>Dirty</td>
<td>Solid suspension</td>
<td>Contamination</td>
<td>Check filter</td>
</tr>
<tr>
<td>Black &amp; acrid smell</td>
<td>No change</td>
<td>Oxidised</td>
<td>Send analysis for</td>
</tr>
</tbody>
</table>

Table 6. Testing methods for lubricants

<table>
<thead>
<tr>
<th>Reason</th>
<th>Test method*</th>
<th>Rejection limits</th>
</tr>
</thead>
<tbody>
<tr>
<td>Degradation by oxidation</td>
<td>Acidity IP-1</td>
<td>1-3 mg KOH/g</td>
</tr>
<tr>
<td>Loss of viscosity</td>
<td>Viscosity IP-71</td>
<td>15% deviation</td>
</tr>
<tr>
<td>Solid contamination</td>
<td>Ash IP-4</td>
<td>More than 0.2%</td>
</tr>
<tr>
<td>Water contamination</td>
<td>Water conten IP-74</td>
<td>More than 0.1%</td>
</tr>
<tr>
<td>Additive loss</td>
<td>TBN IP-177</td>
<td>1 mg KOH/gm</td>
</tr>
</tbody>
</table>

* As per Institute of Petroleum standards for petroleum and its products.

USE OF KNOWLEDGE BASE

A knowledge base is much helpful to compare the test results with the initial limiting values and also analysing the statistical trends of degradation of different properties of various lubricants. As the periodic test results are received and incorporated into the system, the knowledge base in it compares each data with the limiting values of that particular property and gives signal if it has gone beyond the limiting value.

If not, the statistical trend of degradation is formulated for that property with the help of earlier test data. The degradation trend successfully predicts the time when this particular property crosses the output values and accordingly gives a message through the output to enable the user to know about the corrective actions to be taken well before hand.
14.1. INTRODUCTION TO HYDRAULIC SYSTEM

Hydraulics is the science of transmitting energy through the medium of pressurized fluid. The logical and sequential arrangement of various elements to obtain the desired function through fluid is called hydraulic system. Oil hydraulic systems can be built using readily available standard elements together with electrical/ pneumatic interface to perform any complicated sequence of operation. The system is more widely used in machine tools as principal and feed movement drives providing rotary as well as translatory motion with stepless regulation of feed and speed rate, speed changing devices, automatic control of machine cycle, etc. The innovation of electro hydraulic servo valve and proportional valves, which could conveniently interface with electrical and electronic measuring and signalling devices, led to the popular use of electro-hydraulic servo drives in CNC machines. The latest is the application of electro-hydraulic stepping motors with hydraulic torque amplifiers for feed drives in an open loop configuration. This extensive use of hydraulic system is due to their capability of providing infinitely variable speed over a wide range, smooth reversal of moving machine members, automatic overload protection, easy lubrication, etc. Among their shortcomings is leakage of hydraulic fluid through seals and gaps, ingress of air into fluid, effect of temperature and time on fluid properties, etc.

Fig. 14.1 Simple hydraulic system
Figure 14.1 shows a simple hydraulic system for driving a load. The pump driven by a motor delivers oil to the cylinder after sucking it from reservoir through strainer or filter. Flow path of oil is from reservoir (A) through pump (B), relief valve (C), directional valve (E) and flow control (F), to cylinder (D). Flow from (D) is through (E) to (A). Depending upon the throw of the spool in the directional valve, the flow of oil from the pump is carried to the left or to the right. The pressure necessary to overcome the load force can be set by adjusting the relief valve suitably. Speed of the cylinder can be regulated by adjusting flow control valve D. Judicious selection of various hydraulic elements and arranging in a circuit as per requirement can perform many more operations, simultaneously or sequentially.

Based upon the functions, hydraulic elements are classified as follows

- Power generating elements.
- Power controlling elements like pressure control valves, direction and flow control valves, etc.
- Power utilizing elements like hydraulic cylinders and motors.
- Accessories such as filters, accumulators, seals, etc.

Positive displacement pumps are used as power generating elements. Base on the geometry of the pumping mechanism these pumps are either fixed or variable displacement type.

### 14.2. POSITIVE DISPLACEMENT PUMPS

Hydrostatic or displacement pumps provide a given amount of fluid for every stroke, revolution or cycle. Their output except for leakage losses is independent of outlet pressure making them well suited for use in the transmission of power. Selection of pumps is made based on the following performance characteristics.

**Pump rating:** Pumps are generally rated by their maximum operating pressure capability and their output in gallon per minute or litre/min. at a given drive speed.

**Pressure rating:** The pressure rating of a pump is determined by the manufacturer based on reasonable service life expectancy under specified operating conditions. Operating at higher pressure may result in reduced pump life or more serious damage.

**Displacement:** The flow capacity of a pump can be expressed as its displacement per revolution or by its output in litre per minute. Displacement is the volume of liquid transferred in one revolution. It is equal to the volume one pumping chamber multiplied by the number of chambers that pass the outlet per revolution.

**Volumetric efficiency:** In theory, a pump delivers an amount of fluid equal to its displacement each cycle or revolution. In reality the actual output is reduced because of internal leakage or slipping. As pressure increases, the leakage from the outlet back to the inlet or to the drain increases and volumetric efficiency decreases.

\[
\text{Efficiency} = \frac{\text{Actual output}}{\text{Theoretical output}}
\]

For example: if a pump theoretically should deliver 100 litre/min but delivers only 90 litre/min at 80 kg/cm² its volumetric efficiency at that pressure is 90%.

**Delivery:** A pump may be nominally rated as 50 liter/min. actually it may pump more than that under no load condition and less then that at its rated operating pressure. Delivery will be proportional to drive shaft speed.
Most pump manufacturers recommend a vacuum of no more than 12.5 cms of mercury. It means at the pump inlet the min pressure should be $8.7 \text{Kg/cm}^2$ absolute. It is because the liquid vaporizes and air gets dissolved in a vacuum. This puts gas bubbles in the oil, collapsing with considerable force when exposed to load pressure at the outlet and causing damage (cavitations) that will reduce pump life. Further if the inlet fittings are not tight, air can enter in the suction line. This air oil mixture also causes trouble and noise.

For hydraulic pump discharge, power and efficiency can be calculated with the help of the following equations.

Theoretical discharge from pump $Q = \frac{qn\text{lit}}{1000\text{mit}}$

Where $q$ is flow rate in $\text{cm}^3/\text{revolution}$

And $n$ is revolution per minute.

Hydraulic power $P_h = \frac{PQ}{612} \text{KW} = \frac{PQ}{450} \text{hp}$

Where $P$ is the delivery pressure in $\text{Kg/cm}^2$.

Overall efficiency $= \frac{P_h}{P_m}$

Where $P_m = \text{input-power or mechanical power}$

$= \frac{2\Pi nT}{6120} \text{Kw}$

$= \frac{nT}{71620} \text{hp}$

Where $T$ is torque in Kg meter.

The most commonly used positive displacement pumps in hydraulic system are rotary pump like gear pumps, vane pumps, axial and radial piston pumps, etc.
GEAR PUMPS

In its simplest form gear pumps consist of two spur gears generally of equal diameter meshing with each other and enclosed in a body with two intersecting bores with suitable bearings in the end covers. These run in a tight casing with the clearances between the walls of the casing and the gears as small as possible. One of the gears is keyed on the driving shaft that drives the other gear. The suction connection is situated at the point where the meshing ends and for the outlet where it begins, as the rotation of the gear creates a partial vacuum in the chambers formed by the gear teeth coming gradually out of mesh. Thus rotation of the gears creates a high pressure in the space behind and a low pressure in the front of the mating teeth.

Care and Maintenance of Pump: - Gear pump should not be operated at its maximum rated pressure for a long time, otherwise internal wear will accelerate very fast and volumetric efficiency will fall in very short time. Higher pressure will try to deflect the gears more, accelerating the rate of wear. Since a gear pump is generally hydraulically unbalanced and the pressure on the outlet side deflects the gears towards the inlet side. It results in increased gap between the gears and the housing on the outlet side. Increased operating pressure increases the load on the pump bearing resulting in their premature failure.
VANE PUMPS

A vane pump consists of a circular rotor mounted eccentrically inside a circular stator ring or controlled in the elliptical cam ring. The rotor has suitable slots for accommodating radially moving vanes. As the rotor turns, centrifugal force drives the vanes outward so that they always press against the stator ring or cam ring in order to provide the radial sealing between the adjacent chambers and hence between the inlet and outlet ports. Pump inlet is located at a point where the chambers are expanding in size towards direction of rotation. Liquid is drawn into the pump by the partial vacuum caused by this expansion. Then the chamber full of liquid goes to the point where it contracts and forces the fluid through the outlet port. The delivery rate of the pump depends upon the degree of eccentricity. In the case of pumps with circular stator ring delivery rate could be calculated by:

\[ q = 4b \cdot r \cdot z \cdot l \cdot \sin \left( \frac{\pi}{2} \right) \]  
(ignoring the thickness of vanes)

or by  
\[ q = 2\pi De b \]

Where  
- \( q \) = discharge per revolution \( \text{cm}^3/\text{rev} \)
- \( b \) = width of vane/rotor in cm.
- \( r \) = radius of circular stator ring in cm
- \( z \) = no. Of vanes
- \( e \) = eccentricity between rotor and stator ring.
- \( D \) = diameter of rotor in cm.

Thus by varying the eccentricity, flow rate of the pump could be varied. This phenomenon has been utilized in manufacturing variable delivery vane pumps. The eccentricity could be controlled manually by pressure compensator or by servo control. In the stator ring if one suction and one delivery port is formed, the pump is called unbalanced pump as high pressure is formed only on one side (outlet side) of the rotor and shaft. The main bearing would be heavily side loaded. The pressure at the outlet can push the vanes back into the rotor. So arrangement is also made to feed pressure to the under side of vanes to assist centrifugal force.
Fig.14.3 shows the sectional view of a typical constant capacity pump used to feed hydraulic systems of machine tools, presses, excavators etc. A hardened steel stator 13, fitted with twelve sliding vanes 5, is mounted between cast-iron housing 1, and a pump plate 12. A rotor 9 splined to the shaft 3 rotates freely in ball bearings. Distributor discs 11 and 14 provided with two suction and two delivery ports and 7 bear against the faces of the stator and rotor. As the rotor turns, centrifugal force and the pressure of oil brought beneath the stator vanes through the stator ports 4 in the housing, push the vanes against the internal surface of the stator.

Fig.14.3 Sectional views of vane pump assembly
Each vane moves radially in slots in the rotor in conformity with the curved shape of the stator. The chamber between two adjacent vanes thus increases in volume when passing the suction ports 6 and filled with oil from them. When passing the delivery ports 7, the chamber decreases in volume as it expels oil through the ports in delivery line. The space between the housing and pump plate is sealed by cork gasket 8. Leakage along shaft 3 is prevented by an oil resistant rubber seal, and a felt ring mounted in housing 1 and flange 2. Oil seeping between the stator and discs is tapped off through a hole in the pump plate and returned to the reservoir along a pipe connected by a union. Thus any slippage of oil that occurs is directed back to the inlet port to ensure that pressure will not build up on the shaft end.

**Care and Maintenance of Vane Pumps**

Wear mainly affects the rotor, vanes, discs and packing rings as well as bearing. Stator ring wear out at the points of transition from one radius to another. Stator ring may be ground or manufactured a new one from alloy steel hardened to RC 60-64.

Wear in slots of rotor, up to 0.05mm, is repaired by lapping with the abrasive powder. Non-parallelism of slot walls should not exceed 0.02mm. Slot walls can be machined with a thin abrasive disc and lapping.

The rotor journal can be ground and chromium plated. Disc of less bore may be fitted to suit the new journals. Concentricity of journals should be with in 0.02mm. Run out of the end faces of rotor should be with in 0.015mm at a radius of 40mm. Badly worn vanes are replaced by new one manufactured as per new matching dimensions.

Thus repair of vane pumps is costly and labour consuming. It is therefore, necessary to take proper care during operation and maintenance. For that, reservoir should be sealed from atmospheric air. Return line should terminate below oil level. Oil should be clean; 25micron filters are normally preferred. Suction line joints should be airtight. No air bubbles should be there in the reservoir otherwise pump would fail very soon. Inlet vacuum should not exceed 12cm of mercury. Long suction line and higher speed cause excessive vacuum resulting in cavitations. Operating at higher speed and maximum pressure can lead to pump seizure.

**PISTON PUMPS**

Piston pumps offer high volumetric efficiency together with unlimited capacity. As its construction feature is of more complex in nature, they are not competitive with gear or vane pumps unless high system pressure is required. Piston pumps are capable of developing pressure up to $700 \frac{Kg}{cm^2}$ and more which cannot be obtained from either gear or vane pumps. The lower pressure limit is $60 \frac{Kg}{cm^2}$. In contrast to gear and vane pump, piston pumps depend on the suction head and not on the creation of vacuum. So provision for initial suction head should be there. Rotary piston pumps can broadly classified into following categories:

- Radial piston pumps
- Axial piston pumps
Maintenance of piston pumps: The most important care one should take during installation of the piston pump is that the height of the inlet of pump should be as less as possible. The temperature of oil should in no case exceed 60°C, as the high temperature will try to expand moving parts, which may become the cause of wear and seizure.

The pump should be filled with oil during its initial starting in order to ensure lubrication of the moving parts. Relief valve should be set low during the initial starting. Oil should be filtered properly before using in the reservoir as the dirt or foreign particle may score the highly polished surface of the moving parts.

Worn out cylinders should be reamed or bored reducing their taper or out of roundness to 0.01mm. In the case of slight wear just lap the parts using special laps made of cast-iron or bronze. Worn pistons are replaced by new ones, which are lapped and fitted to the whole of cylinders. The pistons should enter the cylinders freely without wobbling so that they sink slowly down under their own weight. The maximum permissible clearance between piston and cylinder is 0.015mm.

14.3. PRESSURE CONTROLLING ELEMENTS

The control devices control the pressure and the displacement, velocity and acceleration of the mechanism. These are links between the generating and converting elements and are called valves. They are basically assemblies of one or more flow restricting elements, which fall into three main classes: -sliding (spool and plate) seating (puppet, ball and flapper) and flow dividing.

One sided pressure acting on valves can produce excessive frictional force. In order to reduce this force 0.2 × 0.2 mm cross section circular grooves are turned on cylindrical valve pistons at intervals of 2 or 3mm. The same objective can be achieved by drilling axial and radial holes connecting symmetrically located spaces serving the same purpose.

Pressure controls are designed for limiting the pressure in any part of the system, for unloading a pump when the present pressure is reached, for building up back pressure in the exhaust line of a reciprocating or rotary hydraulic motor, for bleeding off surplus flow from the pump to maintain a constant pressure in the system, and for reducing the pressure. Pressure control valves are usually named for their primary function, such as relief valve, sequence valve, brake valve etc.
PRESSURE CONTROL AND RELIEF VALVES

Relief valves protect the other elements in the system from excessive pressure by diverting the excess fluid to the tank when the system pressure tends to exceed the set level. In principle valve are of two types (a) simple direct acting relief valve (b) compound relief valve.

The direct acting type of relief valves (fig.14.5) has a ball, poppet or a sliding spool working against a spring. The preload on the spring determines the systems pressure and can be adjusted by screw.

![Fig.14.5 Direct acting type relief valve](image)

The pressure, at which the valve first begins to divert flow, is called the cracking pressure. As flow through the valve increases, the poppet is forced farther off its seat causing increased compression of spring and the pressure. When the valve is fully open to by-pass the full rated flow is full flow pressure. The difference between full flow pressure and cracking pressure is called pressure override.

In some cases pressure override is objectionable. It can result in considerable wasted power due to the fluid lost through the valve before its maximum setting is reached. Ball and poppet valves suffer from high pressure over ride and tendency to chatter. Spool type relief valves of the direct acting type provide smooth and stable operation with superior pressure flow characteristics.

MAINTENANCE OF PRESSURE CONTROL VALVES

Pressure control valve is normally mounted vertically keeping the return port downward. Relief valve should not be set for overload of more than 25% of actual working pressure.

The smooth operation of a pressure control valve depends upon cleanliness of the hydraulic fluid. If, due to foreign particle the small orifice gets chocked, hydraulic balance will be disturbed. In this case body piston will remain lifted all the time and the whole delivery of the pump will be dumped into reservoir if it's a relief valve.
If another hole (drain hole) of the body piston or valve body (as the case may be) is blocked by the foreign particles, the entire working of the relief valve will be inoperative because fluid past the pilot valve is unable to find a drain to the reservoir. Because of the close clearance between the body piston and body bore, foreign particles between them may create jamming of moveable parts, erratic operation and damage to the internal parts. Due to jamming piston will go up at a higher than the set pressure. The relief valve should be periodically flushed at a reduced pressure for thorough cleaning of internal parts. But the adjusting screw should never be completely removed if system is running.

If the spring has become weak, either replace it or in some cases add shims to increase compression on the spring. But avoid adding too many shims; otherwise the spring will be compressed solid.

Check valve seals may get scored. If flat spots are found on the valve seals or on the poppet, replace the affected parts. If scoring is not deep, metallic seat and poppet can be lapped, but be careful; do not remove much valve material.

If the poppet valve has nylon seats, they can take up wear without damaging the mating metallic poppet. Nylon parts are always replaced by new one.

Before dismantling a relief valve ensure that

- Main pump is off.
- All vertical cylinders are lowered.
- Load whose movement may generate pressure, is blocked
- Discharge accumulator is discharged.
- All exposed opening are capped in order to prevent dirt from entering.

### 14.4 FLOW CONTROL VALVES

Speed of a hydraulic actuator depends upon the flow rate of the fluid entering into it. The flow to the actuator is regulated by throttling, thus by passing the excess flow from the pump through the relief valve. The flow through an orifice is given by the equation-

\[
Q = K.A \sqrt{\Delta P}
\]

Where

\[
k = C_d \left( \frac{2}{\rho} \right) \quad C_d = \text{coefficient of discharge 0.61}
\]

\[
\rho = \text{Density } = 0.834 \times 10^{-6} \text{ Kgf}^2 \text{ cm}^{-4} \text{ for petroleum oil}
\]

\[
A = \text{Area of orifice (cm}^2\text{)}
\]

\[
\Delta P = \text{Pressure difference across the orifice Kgf/cm}^2
\]

Valve of 'k' is taken as \[955(Kgf)^{-1/2} cm^7 \text{ sec}^{-1}\] for petroleum oil.

Throttle valves are used in constant delivery systems. As indicated by the equation above, rate of flow depends upon pressure difference between inlet and outlet port so the flow rate of fluid varies according to the resistance faced by the fluid passing through the orifice of the valve. So flow control valves fall into two main categories.

(1) Simple flow control valves  (2) Pressure compensated flow control valves.
14.5 DIRECTIONAL CONTROL VALVE

These valves are used to steer the flow to selected flow in a hydraulic system. The directional valves are mainly classified into two categories.

- Rotary spool type
- Sliding spool type

ROTARY SPOOL TYPE

Rotary type direction control valves are commonly used for table reversal and pre-selection of spindle speed in the speed changing mechanism. In the rotary valves, close contacts are maintained between a rotating port plate and a backup member. High operating force may be required unless the elements are pressure balanced. They are therefore, more usually applicable to lower pressure systems. For speed changing mechanism the valve is turned in the condition where there is no hydraulic pressure through valve. The valve can be rotated by hand, mechanical drive or with the help of servomotor depending upon the design feature of the valve.

The valve used for table reversal consists of a round block or rotor, which is rotated inside a sleeve having four connections for controlling the flow through the four radial channels. During its rotation the passages drilled into the block or rotor, connect the ports of the sleeve. These connections of the ports lead to the different parts of the system. The rotary cock is closed by the lid. It is turned through 45° by the table or slide or manually by means of handle. In its centre position all the ports of the valve are closed. This type of valve is generally used as a pilot who shifts a larger four-way valve for the reversal of table of surface or cylindrical grinder or ram of the slotter.

SLIDING SPOOL TYPE DIRECTION CONTROL VALVE

It consists of a rectangular cost body specially shaped sliding spool and means of positioning the spool. The spool is fitted with precision into the body bore, through the longitudinal axis of the valve body. Cored or machined passages from the port connections in the body are interconnected through annular grooves (under cuts) in the spool or blocked by the spool lands.
The lands of the spool divide this body bore into a series of separate chamber. The ports of the valve body lead into these chambers and the position of the spool determines the nature of the inter connections between the ports. Depending upon the number of ports available for pipe connections the direction control valves can be classified as:

- Two-way directions control valve.
- Three-way directions control valve.
- Four-way directions control valve.

**Fig. 14.7 Sliding spool four way valve**

**MAINTENANCE CARE OF SLIDE VALVES**

Minimum inter-port leakage, low-pressure drop through the valve and fast response of operation of solenoid valves are the important requirements. To obtain low inter port leakage diametral clearance should be maintained with in 5-10µm. The cylindricity and concentricity of bore/spool are to be maintained in the order of 2 to 3 µm. If due to wear, spool becomes unbalanced, the valve may strike or operate erratically. When the spool gets worn, the flow is lost to a certain extent, which causes the slower piston speed.

During repair, o-rings and pickings should be changed. Spring should also be changed if temper is lost. During assembly a little good quality grease can be applied on the surface of the o-ring. Leakage from the worn out valve bore and spool is permissible only up to 0.02% of the rated flow capacity.

If the leakage past the spool increases beyond flow capacity of the drain line, it will create backpressure and keep the spool shifted in one direction making the valve inoperative. Axis of directional valve should generally be kept horizontal.

In selecting solenoids it is necessary to know the number of reversal per minute. In order to avoid over heating and rapid failure of the electromagnets, the valve should have a stroke sufficiently long to enable the armature of the solenoid to be completely drawn inside the coil. The characteristic of spring should be in close agreement with the characteristics of the electromagnet.

Directly operated solenoid, valves are available only upto 45lit/minute-flow rate. For higher flow rate, the directly operated solenoid valves become bulky. So pilot operated valves are generally used.

D.C. solenoids are generally preferred to A.C. since D.C. operation is not subjected to peak initial current, which can cause over heating. AC solenoids are preferred, where fast responses of solenoids are required.
14.6 CHECK VALVE

A check valve can function as either a directional control or a pressure control. It is nothing more than a directional valve. It permits free flow in one direction and blocks flow in the other. The check valve consists of a valve body a poppet or ball. The valve is closed in one direction by the pressure and opened in the other direction after overcoming spring force.

MAINTENANCE OF CHECK VALVES

For a ball check valve to remain leak proof, its ball must continue to be in the form of a sphere. The surface of the seat must conform exactly to the spherical form of the ball otherwise the contact between the ball and seat is reduced to a line and it may start leaking. A check valve with poppet or disc tends to remain leak proof for a longer period. It can be more easily repaired by lapping.

Check valves are also used as foot valve in the suction line. In some designs swing type check valves are used with low-pressure line. The check mechanisms of such valves incorporate a disc swinging on a hinge.

14.7 HYDRAULIC ACTUATORS

Hydraulic actuators perform function opposite to the hydraulic pumps. They convert hydraulic energy back to mechanical energy to perform useful work. Actuators can be classified as

(1) Linear actuator such as cylinder or ram.
(2) Rotary actuator-hydraulic motors.

HYDRAULIC CYLINDERS

Hydraulic cylinders provide a linear motion and are most commonly used of hydraulic drives. Hydraulic cylinders are broadly classified into two categories.

i) Single acting cylinders.
ii) Double acting cylinders.

Single acting cylinder applies force only in one direction. The return motion is accomplished by releasing the pressure when the piston is moved back to its original position by a spring or some external force.

Double acting cylinder is operated by hydraulic fluid in both directions and it is capable of a power stroke either way. In such cylinders, fluid ports are fitted to each end, to function alternately as inlet and outlet ports. The maximum output force available is slightly less than that obtainable from a single acting cylinder due to backpressure generated in the return line. Further in the reverse direction piston rod seals may offer frictional resistance. The following designs of double cylinder exist fulfilling varying requirements.

HYDRAULIC MOTORS

Hydraulic pumps and motors are essentially similar in construction and most of the pumps can act as motors certain valves are not in built with the component. Hydraulic motor can act as a direct alternative to the electric motor with built in infinitely gearbox.
Even clutch is unnecessary with the transmission driven by hydraulic motor. Thus no clutch gear changing is involved and the hydraulic motor can remain stalled indefinitely without harm. Even frequent reversing does not cause damage and it can be used for dynamic braking.

Like hydraulic pump, motors can also be classified as fixed type and variable delivery type. According to the constructional feature hydraulic motors are termed as gear type, vane type, piston type etc.

14.8 POWERPACK AND OTHER ELEMENTS OF HYDRAULIC SYSTEM

A hydraulic system can be built to operate satisfactorily by the judicious selection of the elements and careful system design. Electro-hydraulic servo valve and electro-hydraulic proportional valves are also being used in the system applied in high-tech machines. A power pack generally consists of reservoir for storing and supplying the oil, a strainer at the pump inlet to filter out the large size particles from entering into pump, a pump driven by electric motor, a relief valve, a pressure gauge and a gauge damper.

Depending upon the system requirement, some power packs may consist of a pressure/return line filter, oil level controls, pressure switches and accumulator. Interconnection of various elements in a hydraulic system is obtained through several methods-tubing, hoses and panel mounting. In general the panel mounting needs to be supplemented by tube and/or hose connections to provide flexibility of making connections and also to meet needs of remoteness of disposition of control and drive elements.
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Bhagwati Prasad Gupta has spent more than 35 years in the field of maintenance, and has been exposed to various industries in India, Russia, Germany and Switzerland. As General Manager (Facility Rebuilding Division) in Bharat Heavy Electricals Ltd, he has been responsible for Machine-rebuilding and Modernization of the plants and machinery. He has organized several training workshops, made presentations at seminars and published papers on various topics of Operation and Maintenance of Machine-tools and Equipment in different technical journals. He is Fellow of Institution of Engineers and Member of All India Management Association.

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