5.1 INTRODUCTION TO HYDRAULIC FEED SYSTEM

Oil hydraulics - the science of transmitting and controlling energy through the medium of pressurised oil has several advantages over other methods of energy control. Oil hydraulic systems:

- pack high power in small light components,
- have flat load (torque) - speed characteristics
- can operate continuously under all conditions safely,
- have longer life due to the lubricating properties of the working medium and can withstand heavy duty cycles without undue heating of the medium.

More important oil hydraulic systems can be easily built, using readily available standard elements together with electrical or / and pneumatic interface to perform any complicated sequence of operations. These merits, far outweighing the disadvantages of contaminant sensitivity and noisy operation, have influenced the machine tool designer to employ oil hydraulics for not only the basic tool and work piece movements but also for auxiliary functions such as tool indexing and clamping, loading / unloading of workpieces, etc.

A hydraulic fluid is used to transmit energy / power from one point to another by suitably compressing it by a hydraulic pump. This incompressible
fluid is then made to work on a piston or a hydraulic motor to convert its pressure energy into kinetic energy.

Traditionally hydraulic fluids are made from solvent extracted paraffinic type mineral base stocks of minimum 90 VI. It is now recognised that non-polar hydrocracked base oils duly hydrogenated and dewaxed are preferable base stocks for manufacturing quality hydraulic oils in the lower viscosity ranges viz ISO 32, 46 & 68. These have been found to have superior solvency and water separation characteristics besides better oxidation, colour filterability and viscometrics. For higher grades (ISO 100 and above) solvent extracted base stocks or non-polar bright stocks of adequate viscosity are used. These base oils are fortified with adequate dosage of correct additives to obtain oils meeting the various quality requirements detailed hereinafter. These may be oxidation inhibitors, rust inhibitors, antiwear additives, etc.

It should be noted that the hydraulic oils should not only meet the physical specifications as detailed later, should not only meet rig test requirements, but also meet performance requirements of the system where they are in use.

5.2 PROPERTIES OF HYDRAULIC FLUIDS

5.2.1 Viscosity

Viscosity is the most important property of the oil. Viscosity of the oil selected should be such as to yield optimum pump performance, due consideration being given to the type of pump, its operating speed and
pressure. Use of an oil with too low a viscosity increases leakage and may affect the life of the components because of inadequate lubrication, while that with too high a viscosity can cause inefficient operation due to large pressure drops and viscous drag with subsequent overheating. Pressure losses may be so high as to result in cavitation in pumps.

Viscosity changes with temperature and pressure. The effect of pressure on viscosity can be neglected in machine tool hydraulics. The effect of temperature on the oil is indicated by its viscosity index. An oil with higher viscosity index exhibits less change in viscosity for unit temperature change and is hence preferred. The viscosity index of mineral oils is improved by certain additives.

Viscosity of hydraulic fluid and other industrial lubricants is expressed by dynamic viscosity \([m^2/s]\) which is calculated by dividing absolute viscosity by density. 1 \(cm^2/s\) is called stokes, and centistokes (1/100 stokes) are generally used. Previously Saybolt viscometer (USA), Red wood viscometer (UK), and Engler Viscometer (Germany) for relatively large diameters have been used for industrial purposes. Viscosity of hydraulic fluid is a vital fact in hydraulic equipment.

If a hydraulic equipment is operated by using hydraulic fluid of inappropriate viscosity, poor suction, internal leak, poor lubrication, malfunction of valve, and heat generation in circuit are likely to occur, resulting in shorter equipment life or serious accident.
5.2.2 Bulk Modulus / Compressibility

Bulk modulus which is the reciprocal of compressibility is defined as change in pressure required to cause unit volumetric strain. It is an important parameter in the system design, figuring in calculations of pump output, decompression volumes, pressure surges due to sudden valve operations and drive stiffness resonance.

Two values of bulk moduli - the isothermal and the isentropic - are admissible. Isothermal bulk modulus refers to the value at constant temperature. The isentropic bulk modulus is applicable when pressure changes are rapid allowing no time for entropy change and is also referred to as the dynamic bulk modulus.

These values are further defined as tangent and secant bulk moduli. The isentropic tangent bulk modulus ($B_{st}$) at pressure $p$ and temperature $T$ is given by the relation:

$$B_{st} = - V \left( \frac{dp}{dV} \right)$$

The isentropic secant bulk modulus ($B_{se}$) at pressure $p$ and temperature $T$ is given by

$$B_{se} = \left( - \frac{dp}{dV/V_o} \right)$$
where

\[ \frac{dV}{dp} \]

change in volume,

\[ dV \]

change in pressure,

\[ V_0 \]

initial volume at atmospheric pressure,

and subscripts refers to constant entropy.

At low pressures, the difference between the two values can be ignored. The isentropic bulk modulus of oil under ideal conditions may be as high as 20,000 kgf/cm². However, presence of even small quantities of free air and flexing of hoses, tubes and containers, etc., will considerably reduce the value. For general engineering calculations, a value of 7000 kg/cm² is reckoned practical.

### 5.2.3 Resistance to Foaming

Hydraulic oil contains about 8% of dissolved air by volume. The dissolved air by itself has no harmful effect on the system. However, presence of air in free state does considerably reduce the bulk modulus. The effect is less pronounced at high working pressures, since the free air tends to dissolve in oil at high pressures. On reduction of pressure, the dissolved air is released, promoting the formation of bubbles which may result in loss of drive control as well possible breakdown of pump due to cavitation. Antifoam agents are added to oils for increasing the rate of collapse of the bubbles.
5.2.4 Resistance to Oxidation

Hydraulic fluids being composed of hydrocarbons tend to oxidise. The rate of oxidation increase with high operating temperature, ingress of water and metallic particles which as as catalysts. The products of oxidation which are acidic in nature can be either soluble or insoluble in oil. Soluble oxidation products tend to thicken the oil while the insoluble ones, generally known as sludge, may clog lines, orifices and filters. The oil rapidly degrades with oxidation leading to total breakdown. The extent of oxidation in a fluid sample is assessed by measuring its neutralization number which is the number of milligrams of potassium hydroxide needed to neutralise one gram of oil sample. The rate of increase of neutralisation number is a good measure of the progress of oxidation. A neutralisation number of 1 (a value of 1 mg KOH/gm) is considered as the point for changing / reconditioning of oil. Certain inhibitors added to the oil improve the oil resistance to oxidation.

5.2.5 Fluid flow

Fluid flow is basically governed by a set of equations called the Navier stokes equations and Equation of continuity. These are non linear partial differential equations having complex boundary conditions and hence have no general solutions. But for practical applications certain approximations have been made to reduce the complexities and make the solutions accurate enough for most of the purposes. Calculations in hydraulics are generally based on (1) Reynolds equations, (2) Hagen - Poiseuille equation for flow through capillaries and (3) Bernolli's equation for steady state flow.
The flow of fluid in hydraulic systems may be either laminar, being physically characterised by orderly, smooth, parallel line motion, or turbulent irregular, erratic and eddy-like motion. The internal fluid friction (viscous) forces dominate in laminar flow, whereas the inertia forces are predominant in turbulent flow. The nature of flow is dependant on the velocity of flow \( (v) \), density of fluid \( (\rho) \) viscosity \( (\mu) \) and the characteristic dimension of the particular flow passage \( (D_n) \), and is expressed as a non-dimensional number called Reynolds number \( (Re) \) being equal to \( \rho v D_n / \mu \).

Laminar or turbulent flow can be either steady or unsteady depending on whether the velocities of fluid particles at a section is independent or dependent on time. Generally the flow is assumed to be steady, incompressible and one dimensional. Also the cubical expansion coefficients for liquids are small and hence the effect of temperature on fluid density and flow is negligible. Based on these assumptions, equations for flow and pressure losses have been derived for different flow passages.

Laminar flow, though desirable in systems to minimise the pressure losses, renders the system bulky. It prevails in leakage paths as well in capillaries used for stabilization of valves, hydrostatic bearings and drive systems. A capillary is characterised by a large length to diameter ratio \( (l/d \geq 400) \) of the flow passage. The capillaries are temperature sensitive and hence are unsuitable for control of flow rates in hydraulic systems. Orifices associated with turbulent flow are commonly used for this purpose. An orifice is defined as an opening of short length causing sudden restriction in a flow passage.
The quality of any product is either defined by the user of the product, or its manufacturer, or by an independent Standards Statutory Authority. This authority could be appointed by a country’s government or its trade or business body. In India The Bureau of Indian Standards (BIS) defines the quality levels of hydraulic oils.

5.3 HYDRAULIC STEPLESS DRIVE EXPLANATION

5.3.1 Drives with fluid delivery control

In these hydraulic drives the speed of travelling unit is regulated by controlling pump delivery. In these drives a variable delivery pump (vane or piston pump) is employed. If the hydraulic drive is for translatory motion and consists of a variable delivery pump and cylinder, the piston speed is given by the relationship.

\[ v = \frac{Q}{A} \text{ m/min} \]

\[ Q = \text{pump delivery m}^3 / \text{min} \]

\[ A = \text{working area of piston m}^2 \]

The piston speed is regulated by controlling the pump delivery which can be varied by changing the eccentricity of the vane or piston pump. In machine tool hydraulic system asymmetrical cylinder (Fig.5.1(a)) are generally used. The piston has different velocities of travel during forward and reverse strokes.
\[ v_1 = \frac{4Q}{\pi D^2} \]

when oil is delivered into left hand chamber.

\[ v_2 = \frac{4Q}{\pi (D^2 - d^2)} \]

when oil is delivered into the right hand chamber, where \( D \) and \( d \) are the piston and rod diameter respectively.

If it is necessary that transversing member should have identical velocity in both direction then symmetrical cylinder is employed (Fig.5.1(b)). The forward and reverse velocities can be made equal even with an asymmetrical cylinder. In this case the piston rod area is made equal to half the piston area

\[ \frac{\pi}{4} d^2 = \frac{1}{2} \quad \frac{\pi}{4} D^2 \]

now when the piston travels to the right both chambers of the cylinder are connected to the pump where as when it travels to the left oil is fed only into the right hand chamber.

If the hydraulic drive is for rotary motion and consists of a pump and hydraulic motor the output rpm of the motor is given by the relationship.
(a) ASYMMETRICAL

(b) SYMMETRICAL

CYLINDERS

Fig. 5.1
\[ n_2 = \frac{C_1 e_1 n_1 - Q_1}{C_2 e_2} \text{ rev/min} \]

\( Q_1 \) - leakage losses
\( e_1 \) - eccentricity of the pump
\( e_2 \) - eccentricity of hydraulic motor
\( n_1 \) - rpm of the pump

C1 and C2 = coefficients which depend upon the design parameters of the pump and motor respectively.

If it is assumed that leakage losses depend only upon oil pressure then as is evident from above equation the output rpm depends entirely upon eccentricities \( e_1 \) and \( e_2 \) being directly proportional to \( e_1 \) and inversely proportional to \( e_2 \). In these system the speed range is fairly wide. In the first range the speed is regulated by varying the pump eccentricity; this variation is characterized by constant torque and increase of power with speed. In the next range speed regulation is affected by varying motor eccentricity; in this range power remains constant and torque falls with increase of speed.

If the motor eccentricity becomes zero the rpm theoretically becomes infinite. In fact at extremely small values of motor eccentricity \( e_2 \) the torque becomes too small to overcome even the frictional resistance of the system and the motor gets stalled. Under these circumstances the motor fails to absorb the oil delivered by the pump and there is danger of damage to the system. The minimum eccentricity of the motor should therefore be limited. In general
speed regulation through variation of pump eccentricity should be preferred as it is safer and also provides a flat torque - rpm characteristic.

5.3.2 Drives with flow-control valves

In these drives the output of a fixed delivery pump is controlled by a flow control valve or throttle. It may be recalled that the throttle is a device for changing the orifice of the piping and its hydraulic resistance.

The use of throttles in the hydraulic system enables us to change the delivery and hence the speed at constant pressure. The throttle permits only a small percentage of oil suctioned by the pump to be delivered further into the circuit. Therefore drives with flow control valves must as a rule be provided with a release valve to ensure back flow of oil into the reservoir.

In a hydraulic system for translatory motion the piston speed is given by the relationship.

\[
v = \frac{C_{th}A_{th} (g/\lambda)^{1/2} \sqrt{2(P_1 - P_2)}}{A} = \frac{k_o \sqrt{2(P_1 - P_2)}}{A} \quad \text{m/sec. ...5.1}
\]

where

- \(C_{th}\) = discharge coefficient of throttle
- \(A_{th}\) = area of throttle \(m^2\)
- \(g\) = gravitational acceleration \(m/sec^2\)
- \(\lambda\) = specific weight of oil \(kgf/m^3\)
- \(P_1, P_2\) = oil pressure before and after respectively \(kgf/m^2\)
- \(A\) = working area of the piston \(m^2\)
The value of $k_o$ depends upon the shape of the throttling device, sharpness of its edges etc. The hydraulic resistance of the throttle should be greater than that of release valve and piping.

In hydraulic systems for rotary motion the variation of the output rpm may be considered proportional to the oil delivery through the throttle which is given by the relationship.

$$Q = k_o \sqrt{2 \left( P_1 - P_2 \right)} \quad \ldots \ldots 5.2$$

the notation being same as in equation as in 5.1

In hydraulic systems for translatory as well as rotary motions the flow control valve is installed in series on a straight line segment of the piping in two ways as shown in Fig.5.2.

If the throttle is installed in the forward pressure line it is known as metering in circuit (Fig.5.2(a)) while a circuit in which the throttle is installed in the back pressure line is known as metering out circuit (Fig. 5.2(b)).

In the metering in circuit counter pressure $P_3$ in the right hand chamber of the cylinder is equal to zero. Therefore if load $P$ suddenly drops the piston jerks forward. This shows the metering in circuit cannot be employed in machine tools in which the machining process is intermittent in nature, e.g., milling machine. In such machine tools the metering out circuit Fig.(b) must be employed because in this arrangement the counter pressure $P_3$ is not equal to zero. As load decreases counter pressure $P_3$ increases and therefore the piston speed is again not independent of the load.
METERING-IN CIRCUIT
1. TANK
2. PUMP
3. FLOW CONTROL VALVE

METERING-OUT CIRCUIT
4, 5, 6. REDUCING VALVE
7. CYLINDER
8. PISTON

Fig. 5.2
Hydraulic circuits with parallel installation of the throttle are extremely sensitive to leakage losses and load variation. They are therefore rarely used in machine tools and the application is limited to servomechanism.

As explained above, non-uniformity of motion is inherent in both metering in and metering out hydraulic system with flow control valve due to leakage losses and variation of pressure drop due to variation of load. The motion of traversing unit can be stabilised by means of reducing valves which are capable of maintaining approximately constant pressure drop (constant \( \Delta p \)-type valve) when properly connected to the flow control valve.

**Pressure Reducing Valves**

These valves are used to maintain a secondary pressure in any part of the hydraulic system. The direct acting type valve uses a spring loaded spool to control the secondary line pressure. When secondary pressure fall below the set value the spool moves down due to spring force to allow more flow through and to restore the secondary pressure to the set level. In this way a constant pressure drop is maintained. The constant \( \Delta p \)-type valve is shown in Fig.5.3. The balance equation of the valve may be written as follows:

\[ P_1 (A_p - A_r) + P_2 \cdot A_3 = P_1 (A_3 - A_r) + P_2 \cdot A_p + F_{sp} \]

\[ P_1 (A_p - A_r) = (A_p - A_3) P_2 + P_1 (A_3 - A_r) + F_{sp} \]
CONSTANT ΔP TYPE REDUCING VALVE

Fig. 5.3
It is assumed that $A_r = A_3$

then $A_p - A_r = A_p - A_3 = A_e$ is the effective piston area and the balance equation can be written as

$$\left(P_1 - P_2\right) A_e = F_{sp}$$

$$P_1 - P_2 = \Delta p = \frac{F_{sp}}{A_e} = \text{Constant}$$

In metering in circuit the reducing valve is connected parallel with the throttle while in metering out circuit the flow control valve and reducing valve are connected in series.

The efficiency of hydraulic speed regulation system with oil delivery control is higher than that of system with flow control valve. Therefore constant delivery pumps used in the second type should have a higher mechanical efficiency than vane or piston pumps used in the first type of systems to some what compensate for the difference.

Hydraulic circuits with flow control valves have a wider speed variation range ($R_v = 70 - 100$) and faster response. They are commonly used in systems with low power consumption upto 3.5 kw.

Hydraulic circuit employing variable delivery pumps with flow control valves are more expensive than the ones using constant delivery pumps but have a higher efficiency. They are used in fine boring machine feed unit.
5.4 CYLINDER

The construction and operation of a hydraulic cylinder is explained below.

Cylinders

Cylinders provide a linear drive and are the most commonly used of hydraulic drives. A cylinder essentially consists of a piston located in a tubular housing and a piston rod passing through one of the end covers. The ports provided in the end covers permit entry and return of hydraulic oil.

The detailed constructional features of the commonly used piston type cylinder is shown in Fig.5.4. Standard cylinders are generally made of cold drawn seamless steel tubes. The tubes are bored/ground and finish honed to the required size. A surface finish between 0.2 to 0.4μm is generally desired, especially when using rubber seals for the piston. The pistons are of high grade cast iron, meehanite or bronze. Pistons for low pressure applications may depend upon the fineness of clearance between the piston and the bore for sealing. However, generally, sealing elements such as O rings (dynamic), piston rings, cupseals, etc., are used for the pistons. The allowable tolerances on the dimensions of the finished bore and the piston using seals can be H7 to H11 and f7 to e8 respectively depending upon the pressure rating and the type of seal used.

The piston rods are made of medium carbon steel or case hardening steels depending upon the application and service conditions. The rods are
CONSTRUCTION OF A CYLINDER

Fig. 5.4
hardened and ground and occasionally chrome plated and polished. The rods should be strong enough to prevent buckling. Tubular or hollow rods are preferred to solid ones in case of long strokes. The free end of the rod is usually threaded to facilitate connecting the load. The rod is supported and guided by a bearing in the end cover. The bearing bush is either of bronze or cast iron depending upon the load conditions. O-rings or multiple seals are generally provided for rod sealing. The multiple packagings, when used, are stacked so as not to cause heavy preloads on the rod. It must be noted that the material of the seals for the piston and rod are to be compatible with the fluid medium.

The end covers are made of steel or high grade cast iron. They are generally either screwed to the tube ends or held together by the rods. Supply ports are provided on the end covers, locating them at the top most points with respect to the cylinder mounting to enable automatic scavenging of trapped air. Otherwise separate bleeding points are to be provided.

Cushioning of piston movement at the two ends of its stroke is generally built in to avoid damaging of components due to impact. Cushioning is effected by reducing the velocity of the piston towards the end of the stroke. The flow passage of the oil is blocked by a suitable collar on the piston and the trapped oil is then led through a restructure (adjustable or fixed).

Application of cylinders in machine tools are numerous and is suitable to our application of Table drive for fine boring machine. Simplicity and ease of manufacture, ease of speed control over a wide range and dispensing with
lead screws have been the reasons of the popularity of cylinder drives. They can be conveniently used as feed drives for position control systems with strokes upto a metre. Longer strokes generally suffer from inadequate stiffness due to the larger trapped volume of oil.

5.5 HYDRAULIC CONTROL CONTAINING VALVES
5.5.1 Flow Control Valves

The control of load speed in most hydraulic systems is done by flow control valves. Having sized the pump and actuator for maximum load speed, the lower speeds of load are derived by throttling, bypassing the excess flow through the relief valve. Variable displacement pumps could perform the same task, but are not an economic proposition in many applications. Hence, throttling is commonly resorted to.

The flow through an orifice is given by the equation:

\[ Q_r = K_1 A_0 \sqrt{\Delta P} \]

\[ K_1 = C_d \sqrt{2/\rho} \]

\[ = 955 \text{ cm}^2 (\text{kgf})^{-\frac{1}{2}} \text{ sec}^{-1} \]

\[ C_d = (0.61 \text{ for sharp edged orifices}) \]

\[ \rho = 0.834 \times 10^{-6} \text{ kgf sec}^2 \text{ cm}^{-4} \text{ for petroleum based fluids.} \]

For a given pressure differential across the orifice, the flow rate can be varied by changing the flow cross section A. A few of the basic designs for varying the orifice area are shown in Fig.5.5. Needle valves are the simplest of these valves finding applications for fine metering such as in cushioning of
THROTTLING

Fig. 4.5
cylinders, gauge shut off valves and for very fine feed control. Cross section of a rotary type spool valve is shown in Fig.5.6. The orifice cut along the length of the sleeve is gradually opened or cut out by the rotation of the spool. The helical groove on the spool and the shape of the orifice are selected so as to provide a gradual regulation of flow. Also the length of the orifice is kept as small as possible to minimise the effect of temperature (oil viscosity) on the set flow rates.

A major disadvantage of these simple devices is that any variation in load pressure affects the flow through the valve because of the change of pressure difference across the orifice. Pressure compensated flow control valves (Fig.5.7) provide constant flow for any particular valve setting by maintaining a constant pressure drop across the throttle valve. A common approach of achieving this is by having a pressure reducing valve in series with the throttle valve. The pressures at the inlet and outlet of the orifice are fed on to either sides of the reducing valve spool so as to effect a constant pressure differential, equal to the spring value, across the flow orifice.

Change in temperature and the consequent changes in the viscosity of the oil do affect the flow rates from the orifice type valves. To overcome this problem, temperature compensated valves are used. A flow control valve with both pressure and temperature compensation is shown in Fig.5.8. The length of the temperature compensating element made of an aluminium alloy varies with the oil temperature and the throttle is accordingly adjusted automatically to achieve a constant flow rate. The compensation is brought about by the dissimilar thermal expansion rates of the compensating plunger and the
FLOW CONTROL VALVE

Fig. 5.6
FLOW CONTROL VALVE PRESSURE
COMPENSATED TYPE

Fig. 5.7

FLOW CONTROL VALVE PRESSURE AND TEMPERATURE
COMPENSATED TYPE

Fig. 5.8
matching components. The built-in check valve allows for free flow in the reverse direction.

The flow controls mentioned above are known to offer regulation of flow from about 0.05 l/min to about 100 l/min. Very low flow rate using small orifice is impractical because of plugging of orifice due to contamination of fluids. However, with a fine filter at the inlet to the valve, control of flow of the order of 5 to 20 cm³/min. are claimed to have been achieved.

5.5.2 Direction control valves

These valves are deployed to steer the flow to selected flow paths in any part of a hydraulic circuit. The spool valves both of the linear as well as the rotary movement are devised for the purpose. Principle of operation of these valves is illustrated in Fig.5.9. Rotary type direction control valves are commonly seen as applied to machine tool table reversals such as in cylindrical and surface grinding machines. Owing to the feasibility of application of different modes of control for their operation, the linear spool valves are the most commonly used type. These valves based on their functions and mode of control are classified as in Table 5.1.

The physical construction of a solenoid valve is shown in Fig.5.10. These valves are operated either on AC or DC. The AC operated valves have a drawback in that they tend to burn due to heavy currents drawn, in the event of improper closure of the plunger (as in the case of spool sticking). They are however popular from the point of ease of deriving the control voltage. At high
### Table 5.1
CLASSIFICATION OF DIRECTION CONTROL VALVES

<table>
<thead>
<tr>
<th>Path of Flow</th>
<th>TWO WAY Ports</th>
<th>THREE WAY Ports</th>
<th>FOUR WAY Ports</th>
</tr>
</thead>
<tbody>
<tr>
<td>TWO POSITION</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>THREE POSITION</td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

### Control

<table>
<thead>
<tr>
<th>Control Type</th>
<th>Actuation Type</th>
</tr>
</thead>
<tbody>
<tr>
<td>General</td>
<td>Manual</td>
</tr>
<tr>
<td>Pushbutton Actuated</td>
<td>Leaver Actuated</td>
</tr>
<tr>
<td>Foot Pedal Actuated</td>
<td>Plunger Actuated</td>
</tr>
<tr>
<td>Mechanical</td>
<td>Roler Trip Actuated</td>
</tr>
<tr>
<td>Roler Actuated</td>
<td></td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Actuation Types</th>
</tr>
</thead>
<tbody>
<tr>
<td>Electrical</td>
</tr>
<tr>
<td>Mechanical</td>
</tr>
<tr>
<td>Pilot pressure actuated</td>
</tr>
<tr>
<td>Combined</td>
</tr>
</tbody>
</table>

- **Solenoid Actuated**: SOLENOID SHIFTS FIRST STAGE POOL FOR PILOT PRESSURE ACTUATION OF MAIN SPOOL.
- **Motor Actuated**: MOTOR ACTUATED
- **Pilot Pressure Actuated**: PILOT PRESSURE ACTUATED
- **Two Stage Valve**: TWO STAGE VALVE
DIRECTION CONTROL VALVES

Fig. 5.9
4-WAY, 2 POSITION DIRECTION CONTROL VALVE

Fig. 5.10

4-WAY, 3 POSITION DIRECTION CONTROL VALVE

Fig. 5.11
pressures and flow rates, large operating forces are required making solenoid control untidy. Two stage electrohydraulic valves are therefore devised, whereby low pilot pressure supply from the first stage solenoid valve is used to shift the spool of main valve (Fig. 5.11).

Minimum interport leakage, low pressure drop due to flow through the valve and fast response of operation of solenoid valves are the important requirements of direction control valves. Solenoid valves can function satisfactorily at frequencies as high as 1500-2000 operations an hour.

5.5.3 Check valves

Check valves (non-return valves or one-way valves) are used to control the direction of flow in a circuit to the extent that they permit flow in only one direction. Provision of reversed free flow feature in flow control valves, counterbalance valves and pressure reducing valves and interlock between different pumps of a multipump system are some of the examples of the wide range of application of these valves. These valves consist of a poppet / ball held on to its seat by a spring. A poppet type in-line check valve is shown in Fig. 5.12. Check valves are also available with O-ring sealing. The spring force on the sealing element determines the cracking pressure or the back pressure prior to allowing the flow in the intended direction. Valves without spring are suitable for vertical mounting only.

Two commonly used modifications from the standard design are the restriction check valve and the pilot operated check valve. The former permits
CHECK VALVE INLINE TYPE

Fig. 5.12
free flow in one direction and restricted flow in the opposite direction. The pilot operated check valve operates as a standard valve but can be controlled by a pilot pressure to permit free flow in the reverse direction.

5.5.4 Shut off 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. The direct acting type of relief valves have a ball, poppet or a sliding spool working against a spring. The preload on the spring determines the system pressure and can be adjusted. The pressure at which a relief valve cracks open is termed as the cracking pressure and the pressure when the valve is fully open to by-pass the full rated flow is full flow pressure. The difference between the two—the pressure differential or the pressure override should necessarily be small for close pressure control. The pressure override is due to the extra compression of the spring at higher valid openings to accommodate increased flow rates and is therefore depended on the stiffness of the spring used. Long, weak springs are therefore preferred for the application.

5.6 VARIABLE DISPLACEMENT PUMP VANE PUMP

Sliding vane type pump is the most commonly used types in machine tool hydraulic systems. The working principle of the pump is shown in Fig.5.13. The pump consists of a circular rotor mounted eccentrically inside a circular stator ring, thus providing the suction and delivery chambers. The
VANE PUMP UNBALANCED TYPE

Fig. 5.13
The materials of construction for the different parts of the pump and the desired component accuracies are listed in Table 5.2. The equations for calculation of flow, power and efficiency are given in Table 5.4.

Advantages of vane pump

1. Less pulsation in discharge pressure (practically, the pulsation has no effect).
2. Efficiency decreases to lesser extent even if the vane is worn out.
3. Small dimensions relative to hydraulic power of pump.
4. Ease in maintenance due to less failure.
### Table 5.2 Material of construction (Vane pump)

<table>
<thead>
<tr>
<th>Component</th>
<th>Housing and End Cover</th>
<th>Bearings</th>
<th>Rotor and Shaft</th>
<th>Vanes</th>
<th>Side plate</th>
</tr>
</thead>
<tbody>
<tr>
<td>Material used</td>
<td>Mechanite or high strength cast iron</td>
<td>Sleeve bearing of bronze or sintered bronze impregnated with PTFE liner and roller bearings</td>
<td>Carburizing grade of alloy steel; hardened and ground. If the rotor is split type, the shaft can be of medium carbon alloy steel hardened and ground</td>
<td>High speed tool steel properly hardened and ground</td>
<td>Manganese or silicon bronze</td>
</tr>
</tbody>
</table>

### Table 5.3 Working clearance and geometrical accuracies

<table>
<thead>
<tr>
<th>Type of Pump</th>
<th>Clearance in μm</th>
<th>Geometrical accuracies in μm</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Axial</td>
<td>Radial</td>
</tr>
<tr>
<td>Uncompensated</td>
<td>5 to 15</td>
<td>0.5 to 5</td>
</tr>
<tr>
<td>Axial clearance compensated</td>
<td>0.5 to 5</td>
<td>0.5 to 5</td>
</tr>
</tbody>
</table>

### Table 5.4 Equation for derived capacity

<table>
<thead>
<tr>
<th>Sl.No.</th>
<th>Type of Pump</th>
<th>Derived Capacity (q) (cm³ / revolution)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.</td>
<td>Vane Pump</td>
<td>Unbalanced</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Balanced</td>
</tr>
</tbody>
</table>

- $D_r$ = Diameter of rotor,
- $e$ = Eccentricity between rotor and stator centres,
- $e'$ = 1/4 (major diameter - minor diameter),
- $b_r$ = Width of rotor.
5.7 ACCUMULATORS

Accumulators are employed in hydraulic circuits to store the excess flow (hydraulic power) from a pump to meet any of the following requirements:

- to meet the demand for increased flow rates during a portion of the working cycle, thus allowing a smaller pump to be used for the system,

- to provide an emergency source of energy to render a fail-safe system in the event of failure of the pump,

- to maintain the system pressure within tolerable limits in locked circuits by compensating for leakage flow or increase in pressure due to thermal expansion and

- to absorb pump flow ripples and pressure surges in a system.

Accumulators can be of weight, spring or hydropneumatic type. The hydropneumatic type of accumulators using precharged nitrogen to act as a spring is commonly used. Nitrogen is separated from the oil side by a piston, bag or diaphragm. Hydraulic fluid pressure, when admitted into the oil side of the accumulator, moves the separator compressing the nitrogen till a balance of pressure is reached on the two sides. Nitrogen is used in preference to air since the latter has the tendency to cause dieselling of air-oil vapour. The constructional details of bag type accumulator is shown in Fig.5.14. The charging unit shown alongside enables regulated precharging. Accumulators
BAG TYPE ACCUMULATOR

Fig. 5.14
are specified by the maximum swept volume of oil and the maximum operating pressure. The diaphragm and bag type accumulators are generally supplied with a small precharge of gas to protect the separator element.

Presuming the availability of sufficient flow from the pump, the flow into and from the accumulator is governed by the law $p v^n = \text{constant}$, the index $n$ varying between 1 to 1.3 depending upon the construction and operation. The accumulator is chosen based on the requirements of maximum flow and operating pressure. The precharge pressure of the gas is determined based on the flow and operating pressure range, aided by the p-v diagram supplied by the manufacturer of the accumulator. Operation at higher precharge pressures provides for larger flow for a given operating pressure range and is therefore advantageous particularly in accumulator applications for leakage compensation.

5.8 PRESSURE SWITCHES

These are auxiliary control elements required in a hydraulic system to sense the pressure level in any branch of the system to electrically trigger any other function in the system. It consists essentially of a micro switch whose contacts are opened or closed by a pressure sensing mechanism. Based on the pressure sensing mechanism the switches are classified as-diaphragm, plunger and bellows type. A diaphragm type construction is shown in Fig.5.15 the deflection of the diaphragm under pressure is utilised to operate the microswitch. The initial gap between the diaphragm and the plunger of the switch is adjusted for the desired pressure setting.
PRESSURE SWITCH

Fig. 5.15
Pressure switches can be built for providing signals both at low and high limits of system pressure and also to operate on a pressure differential.

5.9 TABLE DRIVE USING HYDRAULIC FEED SYSTEM

The cylinder attached with piston and table of Fine boring machine is shown in Fig.5.16. When the hydraulic fluid with high pressure is passed through the cylinder the piston is pushed forward. When the piston is pushed forward since the table is connected with the piston the table also moves. This is how the hydraulic energy of the fluid is converted into mechanical energy. The fine boring machine consists of boring units on both sides of the machine. So the cycle of events that takes place is as shown in Fig.5.16.

1. Job which is fixed on table is moved from left to right fast.
2. Boring at right spindle takes place at slow speed.
3. Table’s moved from right to left fast.
4. Boring takes place at left spindle slowly.
5. Table is brought to home position fast.

This is done in hydraulic feed mechanism by direction control valves. The speed is varied by means of flow control valves infinitesmally. The check valve allows the fluid to flow in only in one direction the function of these 3 valves were explained earlier.

The circuit showing the operation of control valves for the table movement of fine boring machine is shown in Fig.5.17. There are 4 solenoid operated valves, four check valves and four flow control valves.
Fig. 5.16

Cylinder Piston Table

Arrangement

Section A-A
CIRCUIT SHOWING OPERATION OF VALVES

Fig. 5.17
The general function of solenoid valves are given in the diagram. The sequence of events that takes place are in the following order.

The solenoid operated valves have the following position.

<table>
<thead>
<tr>
<th>Sol</th>
<th>ON/OFF</th>
<th>Position</th>
</tr>
</thead>
<tbody>
<tr>
<td>3</td>
<td>ON</td>
<td>High speed forward</td>
</tr>
<tr>
<td>3</td>
<td>OFF</td>
<td>Dec</td>
</tr>
<tr>
<td>1</td>
<td>ON</td>
<td>Low speed forward</td>
</tr>
<tr>
<td>1</td>
<td>OFF</td>
<td>Stops</td>
</tr>
<tr>
<td>4</td>
<td>ON</td>
<td>High speed reverse</td>
</tr>
<tr>
<td>4</td>
<td>OFF</td>
<td>Dec</td>
</tr>
<tr>
<td>2</td>
<td>ON</td>
<td>Low speed reverse</td>
</tr>
<tr>
<td>2</td>
<td>OFF</td>
<td>Stops</td>
</tr>
</tbody>
</table>

After switching on the spindle drive, to switch on the table drive using hydraulic system, Sol 3 is in ON position, Sol 3 is meant to move the table at high speed forward.

While the table moves correspondingly fluid passes via check valve which allows the fluid only in one direction and Table moves fast from left to right. This is how rapid advance (RA) takes place. While table reaches near right spindle sol 3 is in off position. Table decelerates and at this time Sol. 1 is in on position. Sol 1 is meant to move the table slowly towards forward direction. At this the circuit designed such that the fluid goes via a flow control valve. By varying the flow we can get speed infinitesmally. We can get as many feeds as we want. As low as 0.05 mm/min table feed can be obtained.
Next Sol 4 is in on position. Sol 4 is meant to move the table at High speed reverse direction. Table moves from right to left fast. Rapid return take place correspondingly fluid goes via check value. The check valve allows the fluid to flow in only one direction. So table only moves in reverse direction. When table reaches near left side Sol 2 is in on position Sol 2 is meant to move the table slowly in reverse direction. At this time fluid flows via a flow control valve. We can vary the feed infinitesmally by controlling the flow. Next Sol 2 is in OFF position Table comes to home position and stops. This is how the cycle of events takes place in Hydraulic feed mechanism.

5.10 DESIGNING VARIOUS PARAMETERS FOR HYDRAULIC SYSTEM

5.10.1 Designing cycle time for the above operation

During the operation of fine boring, the table is moved from left to right boring done at right side and table moved from right to left, boring done at left side and table brought back to home position. The cycle time for the above operation is calculated as follows.

<table>
<thead>
<tr>
<th>Tool Material</th>
<th>-</th>
<th>Carbide</th>
</tr>
</thead>
<tbody>
<tr>
<td>Work Material</td>
<td>-</td>
<td>Cast Iron</td>
</tr>
<tr>
<td>Cutting speed</td>
<td>-</td>
<td>90 metres/minute</td>
</tr>
<tr>
<td>Minimum feed</td>
<td>-</td>
<td>0.05mm/Rev.</td>
</tr>
<tr>
<td>$\phi$ of bore</td>
<td>-</td>
<td>$\phi$ 50mm</td>
</tr>
<tr>
<td>Cutting speed</td>
<td>$\frac{\pi DN}{1000}$</td>
<td>$= \ 90 \ m/min$</td>
</tr>
</tbody>
</table>
Spindle Speed \( N = \frac{90 \times 1000}{\pi \times 50} \) 600 rpm

Feed/Min \( = \frac{600 \times 0.05}{1} = 30 \text{mm/min} \)

Bore length - 70mm

Machining time \( = \frac{70}{30} \times 60 = 140 \text{sec} \)

Rapid rate \( = 3000 \text{mm/min} \)

Rapid travel \( = 300 \text{mm assumed} \)

Cycle time for rapid travel \( = \frac{300}{3000} \times 60 = 6 \text{sec} \)

Table left to right rapid-
right side boring \( - 140 \text{sec} \)
right to left \( - 12 \text{sec} \)
left side boring \( - 140 \text{sec} \)
left to home position - \( 6 \text{sec} \)

Total \( = 304 \text{sec} \)

Total cycle time - 304 sec.

5.10.2 Table Feed: Table movement rapid rate 3000mm/min, slow feed rate 0.05mm/rev.
5.10.3 **Circuit Pressure**

Circuit pressure for Machine Tool is 30 bar.

5.10.4 **Selecting the actuator : Designing cylinder.**

Total load acting on the cylinder.

<p>| | | |</p>
<table>
<thead>
<tr>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Table</td>
<td></td>
<td>500 kg</td>
</tr>
<tr>
<td>Fixtures</td>
<td></td>
<td>200 kg</td>
</tr>
<tr>
<td>Job</td>
<td></td>
<td>20 kg</td>
</tr>
<tr>
<td>Total Weight F</td>
<td></td>
<td>720 kg</td>
</tr>
<tr>
<td>Cutting Thrust</td>
<td></td>
<td>300 kg</td>
</tr>
</tbody>
</table>

Feed thrust required by cylinder

\[ F \times \mu + 300 = 720 \times 0.2 + 300 \]

\[ = 144 + 300 \text{ kg} = 444 \text{ kg} \]

Assume maximum thrust on cylinder 4 times required thrust = 1776 kg.

\[ \frac{F}{30} = \frac{P \times A}{A} \]

\[ 1776 = 30 \times A \]

\[ A = \frac{1776}{30} = 59.2 \]
\[
\pi/4 \times d^2 = 59.2 \\
d = 8.68 \text{ cm} = 87 \text{ mm} \\
\text{Take } d = 90 \text{ mm}
\]

**Cylinder bore** \[ d = 90 \text{ mm} \]

**Stroke** = 630 mm. (Ref. Chapter 4)

### 5.10.5 Selecting the pump

The pump selected is vane pump. The advantages of vane pump is explained earlier.

**Cylinder bore** = 90 mm

Area = 63.62 cm² maximum speed is = 300 cm/min

**Pump discharge** = \[ \frac{A \times V}{1000} \]

\[
= \frac{63.62 \times 300}{1000} = 19.08 \text{ lit/min}
\]

Take \[ Q = 20 \text{ lit/min} \]

**Power of motor** = \[ \frac{PQ \times 100}{612 \eta} \]
= \frac{30 \times 20 \times 100}{612 \times 80} = 1.2 \text{ kw}

= 1.63 \text{ HP}

Power of motor selected is 2 HP.

5.10.6 Determining the pipe size

Pipe size as per chart (Fig. 5.18) for 20 lit/min and velocity 3 m/sec. is

1 cm = 10 mm

We have taken 12 mm size pipe.

5.10.7 Selecting the Oil

The oil selected is paraffin based mineral oil.

5.10.8 Determining tank capacity

Tank capacity \ > \ 3 \ times \ the \ pump \ delivery
Should be \ > \ 3 \times 20
\ = \ 75 \ \text{lit/min.}
Tank capacity taken 75 lit/min.

5.10.9 Filter capacity

Suction filter 30 lit/min
Return line filter 40 lit/min.
Delivery Flow Rate, Flow Velocity and Pipe Size

Example: What should be the diameter of the delivery pipe for a hydraulic pump of delivery flow rate 60 l/min? As the pipe is on the delivery side, assume the flow velocity as 3 m/sec and draw a linear line connecting 60 and 3. SGP 3/4 is obtained as the pipe diameter.

Fig. 5.18
5.10.10 Electric Motor selected

2 HP 1500 rpm AC 50 HZ.

5.10.11 Level of contamination in filter is 25μ to 50μ

5.10.12 Pressure gauge 0-10 bar.

5.10.13 Pressure switch - 0-30 bar.

5.11 DESIGN OF HYDRAULIC POWER PACK (FIG.5.19)

The fundamental requirement of a system is a suitable pressure supply or a power pack as is generally referred to. A powerpack may be a simple pump-motor unit with a reservoir or a package consisting of the entire system. Power pack generally implies a source of supply of pressurized fluid in a condition acceptable to the drive and control circuitry.

The whole unit of power pack pump tank model consists of following:

1. Hydraulic tank 75 litres capacity (800x500x500) comprising of cleaning door, level gauge oil filter/air breather and drain plug.

2. Suction filter 30 lt/min capacity 50μ filtration.

3. Return line filter 40 lit/min capacity 25μ filtration.

4. Variable displacement pump vane pump 20 lit/min at 30 bar pressure.
### HYDRAULIC POWER PACK

**Fig. 5.19**

<table>
<thead>
<tr>
<th>Part</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>11</td>
<td>Hydraulic tank 75 litres capacity</td>
</tr>
<tr>
<td></td>
<td>800x500x5000 comprising of clearing</td>
</tr>
<tr>
<td></td>
<td>Oil level gauge, oil filter, air breather and a drain plug</td>
</tr>
<tr>
<td>12</td>
<td>Suction filter, 30l/min capacity 50μm filtration</td>
</tr>
<tr>
<td>13</td>
<td>Return line filter, 40l/min capacity 25μm filtration</td>
</tr>
<tr>
<td>14</td>
<td>Variable displacement pump, vane type</td>
</tr>
<tr>
<td></td>
<td>80l/min at 30 bar pressure</td>
</tr>
<tr>
<td>15</td>
<td>Electric motor, 3hp 1500 rpm</td>
</tr>
<tr>
<td></td>
<td>440V A/C 50 Hz</td>
</tr>
<tr>
<td>16</td>
<td>Double female flange with coupling</td>
</tr>
<tr>
<td>17</td>
<td>Check valve, line mounting, 30 l/min</td>
</tr>
<tr>
<td>18</td>
<td>Multistage gauge isolator 6 in. R1/4&quot;</td>
</tr>
<tr>
<td>19</td>
<td>Pressure gauge 0-70 bar 100 dia, dsl panel mounting on snubber</td>
</tr>
<tr>
<td>20</td>
<td>Accumulator 1 litre capacity block type</td>
</tr>
<tr>
<td>21</td>
<td>Shut off valve R 1/4&quot;</td>
</tr>
<tr>
<td>22</td>
<td>Pressure switch 0-50 bar R 1/4&quot; Port</td>
</tr>
</tbody>
</table>
5. Electric motor 2HP 1500rpm 440V A/c.

6. Double foot flange with coupling.

7. Check valve line mounting.

8. Multi stage gauge isolater 6 stn.

9. Pressure gauge 0-30 bar 100 dia dial.

10. Accumulator 1 litre capacity.

11. Shut off valve.

12. Pressure switch 0-30 bar.

5.11.1 Hydraulic tank and filter

These are essentially storage tanks for the system fluid, although they may often facilitate mounting of atleast a part of the hydraulic system. Reservoirs are to be generally kept separated from the machine actual for reasons of isolation of thermal conditions and ease of servicing. Points to be reckoned in designing a reservoir are:

i. sealing of fluid chamber from external source of contamination;

ii. sizing of the reservoir
a. To hold adequate volume of fluid reckoning the amount of oil that may drain back from the system either during a portion of the cycle or at the time of servicing.

b. to have sufficient radiating area for dissipating the heat generated in the system so that the fluid temperature in the tank does not exceed 60°C. A separate cooler may have to be included if the tank tends to be too large.

iii. the bottom of the reservoir is to be kept 120 to 200 mm above the ground level to facilitate draining, cleaning, transportation and improved heat dissipation

iv. Drain plug or some other means for draining almost the entire content of oil

v. Convenient access for cleaning the inside of reservoir-cleanout openings are to be provided in case of reservoirs with permanently fitted top covers

vi. provision of a breather hole with a filter,

vii. provision of a filter cup with wiremesh screen

viii. fluid level indicator showing the minimum and maximum permissible levels of the fluid in the reservoirs,
ix. inside of the reservoir to be painted with an oil resistant paint and

x. baffle plates to separate pump suction and return lines.

Reservoirs are generally sized to hold 3 to 5 times the pump discharge per minute. Tanks are generally rectangular in shape. Too shallow a reservoir does not provide enough surface area for heat dissipation and too deep and narrow a tank does not provide enough surface for proper separation of entrained air bubbles or foam. Pumps are sometimes mounted at the base of the reservoir on a suitable extension to provide positive suction head due to the column of oil in the reservoir.

Heaters and coolers to maintain the fluid temperature within the maximum permissible limits are required to be provided based on the heat generation in the system, the ambient temperature range (at the work-site) and dissipation through the tank walls. Thermostat controllers to this effect, if essential, and temperature indicators are to be provided.

Pressure gauges with snubbers and gauge shut off valves are to be provided at a suitable location with respect to the relief valve and operator’s position.

The fluid in the reservoir is required to be cleaned prior to entry into the system to a level of contamination acceptable for satisfactory functioning of the circuit. Location of filters, filtration capacity, flow and pressure ratings and dirt holding capacity of filter elements are to be reckoned while providing
filtration. The general practice in most of the hydraulic circuits for machine tools is to provide a strainer at the pump inlet. A 125 μm (equivalent of 120 mesh) rating for strainers is considered a good practice. The strainers need to be of adequate capacity, not creating more than 0.15 kgf/cm² pressure drop at the prevailing flow rate. Apart from a strainer, either a pressure line filter or return line filter of 30-40 μm absolute rating is adequate for most of the common applications. However, when critical components like servovalves are used, a pressure line filter of suitable micron rating immediately behind the valve is essential. Servo drive manufacturers recommend the use of a filter rated at 25 μm absolute for the purpose.