HYDRAULIC CIRCUIT DESIGN AND ANALYSIS

A Hydraulic circuit is a group of components such as pumps, actuators, and control valves so arranged that they will perform a useful task. When analyzing or designing a hydraulic circuit, the following three important considerations must be taken into account:

1. Safety of operation
2. Performance of desired function
3. Efficiency of operation

It is very important for the fluid power (Hydraulics and Pneumatics) designer to have a working knowledge of components and how they operate in a circuit. Hydraulic circuits are developed through the use of graphical symbols for all components. The symbols have to conform to the ANSI specification.

5.1 Control of a Single- Acting Hydraulic Cylinder :

A single-acting cylinder can exert a force in only the extending direction as fluid from the pump enters the blank end of the cylinder (usually left side of the piston). Single- acting cylinder do not retract hydraulically. Retraction is accomplished by using gravity or by the inclusion of a compression spring in the rod end.

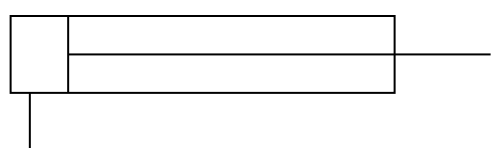


Fig 5.1 Symbol

Force during extension stroke is , $F_{ext} = p * A_P$

Velocity during extension stroke is , $\mathbf{v_{P\,ext} = Q_P / A_P}$

The force and velocity during retraction stroke depends upon spring rate as single – acting cylinder do not retract hydraulically

Figure 5.2 shows a two- postion, three way, manually operated, spring offset directional control valve (DCV) used to control the operation of a single – acting cylinder. In the spring offset mode, full pump flow goes the tank via the pressure relief valve. The spring in the rod end of the cylinder retracts the piston as oil from the blank end ‘A’ drains back to the rank. When the valve is manually actuated the pump flow goes to the cylinder blank end ‘A’ via DCV 1 position. This extends the cylinder. At full extension, pump flow goes through the relief valve. Deactivation of the DCV allows the cylinder to retract as the DCV shift into its spring – offset mode.

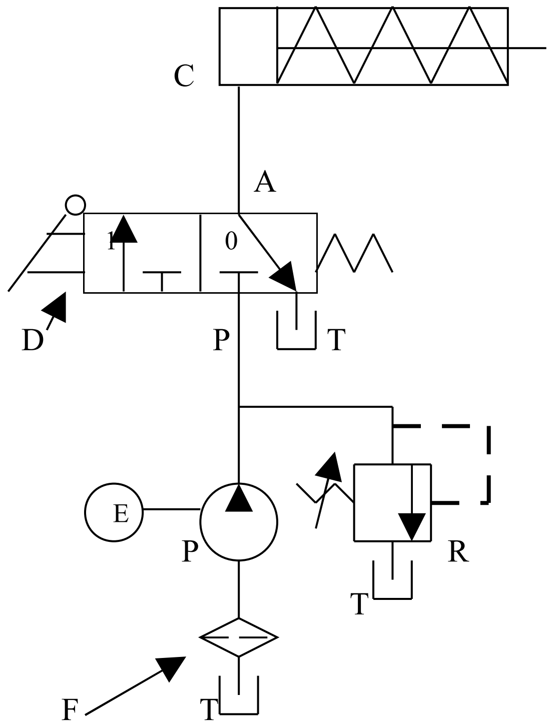


Fig 5.2. **Control of Single -Acting Hydraulic Cylinder**.

C = Single acting cylinder

P = Pump

E = Electric Motor

T = Tank

F = Filter

R = Relief Valve

D =2-position, 3 way DCV Manually operated and spring return

5.2 Control of Double -Acting Hydraulic Cylinder :

Double –Acting cylinders can be extended and retracted hydraulically. Thus, an output force can be applied in two directions.

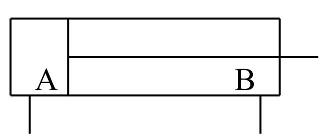


Fig 5.3 **Double acting cylinder**

The output force (F) and piston velocity of double acting cylinder are not the same for extension and retraction strokes. During the extension stroke, fluid enters the blank end (A) of the cylinder through the entire circular area of the piston (A_P). However during the retraction stroke, fluid enters the rod end through the smaller annular area between the rod and cylinder bore ($A_P – A_R$), where A_P = piston area , and A_R = rod area. Since A_P = is grreater than ($A_P – A_R$), the retraction velocity is greater than the extension velocity since the pump flow rate is constant.

Similarly during the extension stroke, fluid pressure bears on the entire area of the piston(A_P). However during the retraction stroke, fluid pressure bears on the smaller annular area ($A_P – A_R$). The difference in area accounts for the difference in output force, with the output force is greater during extension.

<u>Extending stroke</u> :

Force, $F_{ext} = p * A_P$ -------- 1

Velocity, $v_{ext} = Q_p / A_P$ -----2

<u>Retraction Stroke</u> :

Force, $F_{ret} = p * (A_P – A_r)$ --- 3

Velocity, $v_{ret} = Q_p / (A_P – A_r)$ --- 4

It can be seen from the above 4 equations that force during extension stroke and velocity of piston during retraction stroke is greater for the same operating pressure and flow rate.

The power developed by a hydraulic cylinder for either the extension or retraction stroke, can be found out by (velocity multiplied by force) or from (flow rate multiplied by operating pressure)

$$\text{Power (kW)} = v_p \text{(m / s)} * \text{F (kN)} = \text{Q (m}^3\text{ / s)} * \text{p (kPa)}$$

Figure 5.4 shows a circuit used to control a double – acting hydraulic cylinder. When the four way valve is in centered configuration , the cylinder is hydraulically locked as the ports A and B is blocked. The pump flow is unloaded back to the tank at essentially atmospheric pressure.

When the four way valve is actuated into the 1st position , the cylinder is extended against its load force F_{load} as oil flows to the blank end of the cylinder from port P through port A . Also, oil in the rod end of the cylinder is free to flow back to the tank via the four way valve from port B through port T. Note that the cylinder would not extend if this oil were not allowed to leave the rod end of the cylinder.

When the four way valve is actuated into the 2st position , the cylinder is retracts against as oil flows to the rod end of the cylinder from port P through port B. Oil in the blank end of the cylinder is returned to the tank from port A to port T.

At the end of the stroke, there is no system demand for oil. Thus, the pump flow goes through the relief valve at its pressure- level setting unless the four- way valve is deactivated. In any event the system is protected from any cylinder overloads.

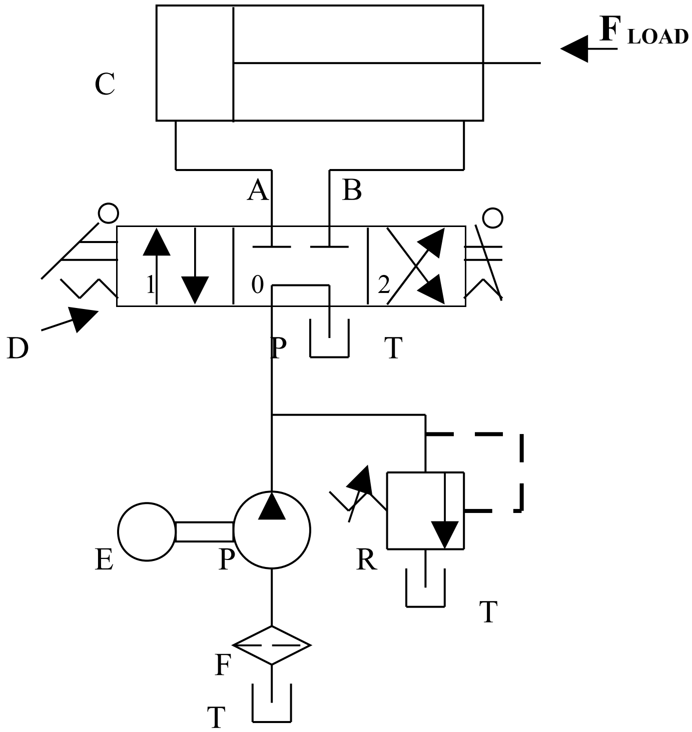


Fig 5.4. **Control of Double -acting hydraulic cylinder.**

C = Double acting cylinder

P = Pump

E = Electric Motor

T = Tank

F = Filter

R = Relief Valve

D =3-position, 4 way ,Tandem center, Manually operated and Spring Centered DCV

Problem 1. A double acting cylinder is hooked up to reciprocate. The relief valve setting is 70 bars. The piston area is 0.016 m^2 and the rod area is 0.0045 m^2. If the pump flow is 0.0013m^3 / s, find the cylinder speed and load- carrying capacity for the

a. Extending stroke

b. Retracting stroke.

Solution:

Relief valve pressure setting, $p = 70$ bars $= 70 * 10^5$ N /m^2

Piston area, $A_p = 0.016$ m^2

Rod area, $A_r = 0.0045$ m^2

Pump flow, $Q_p = 0.0013\ m^3/s$

a) Extending Stroke:

Cylinder speed, $v_{p\ ext} = Q_p / A_P$

$= 0.0013 / 0.016$

$= \mathbf{\underline{0.0813\ m / s}}$

Load carrying capacity, $F_{load} = p * A_P$

$= 70 * 10^5 * 0.016$

$= 112000\ N =$ **<u>112kN</u>**

b) Retracting Stroke:

Cylinder Speed, $v_{p\ Ret} = Q_p / (A_P - A_r)$

$= 0.0013 / (0.016 - 0.0045)$

$= \mathbf{\underline{0.113\ m / s}}$

Load carrying capacity, $F_{load} = p * (A_P - A_r)$

$= 70 * 10^5 * (0.016 - 0.0045$

$= 80500\ N =$ **<u>80.5kN</u>**

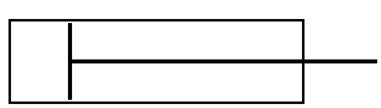
5.3 Regenerative circuit:

Operation

Figure 5.5 shows a regenerative circuit that is used to speed up the extending speed of a double-acting hydraulic cylinder. Here the pipelines to both ends of the hydraulic cylinder are connected to pump, one end (A) through the 2 / 3 way DCV and the other end (B) directly. The operation of the cylinder during the retraction stroke is the same as that of a regular double-acting cylinder. Fluid flows through the DCV zero position from the actuator A side during retraction. In this position, fluid from the pump directly enters the rod end of the cylinder (direct connection). Fluid in the blank end drains back to the tank through the DCV as the cylinder retracts.

When the DCV is shifted to 1 position due to manual actuation, the cylinder extends. The speed of extension is greater than that for a regular double-acting cylinder because flow from the rod end (Q_R) regenerates with the pump flow (Q_P) to provide a total flow rate (Q_T), which is greater than the pump flow rate to the A side of the cylinder. (Area of blank end is more than rod end, thereby blank end provide least resistance)

C



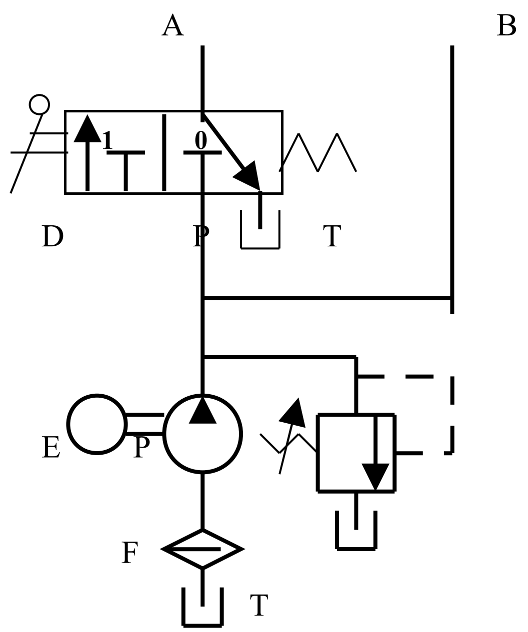


Fig 5.5. Regenerative circuit.

C = Double acting cylinder

P = Pump

E = Electric Motor

T = Tank

F = Filter

R = Relief Valve

D =2-position, 3 way , Manually operated and Spring return DCV

Cylinder Extending Speed

The total flow rate entering the blank end (A) of the cylinder equals the pump flow rate plus the regenerative flow rate coming from the rod end of the cylinder:

$$Q_T = Q_P + Q_R$$

Or $$Q_P = Q_T - Q_R \quad \text{---(1)}$$

We know that the total flow rate equals the piston area multiplied by the extending speed of the piston (Vp_{ext}). Similarly, the regenerative flow rate equals the difference of the piston and rod areas (A_p - A_r) multiplied by the extending speed of the piston. Substituting these two relationships into the eq (1) yields

$$Q_P = A_p Vp_{ext} - (Ap - Ar)Vp_{ext}$$

Therefore, $$Q_P = A_r Vp_{ext}$$

Hence the extending speed of the piston, $V_{pext} = \frac{Q_p}{A_r}$ --- (2)

Thus the extending speed equals the pump flow divided by the rod area. Thus, a small rod area (which produces a large regenerative flow) provides a large extending speed. In fact the extending speed can be greater than the retracting speed if the rod area is made small enough.

Ratio of Extending and Retracting Speeds

Let's determine under what condition the extending and retracting speeds are equal. We know that the retracting speed (Vp_{ret}) equals the pump flow divided by the difference of the piston and rod areas:

$$V_{pret} = \frac{Q_p}{A_p - A_r} \qquad \text{---(3)}$$

Dividing eq(1) with (4) we have $\frac{Vp_{ext}}{Vp_{ret}} = \frac{Qp / A_r}{Qp /(Ap - A_{r)}} = \frac{A_p - A_r}{A_r}$

Simplifying we obtain the ratio of extension speed and retracting speed

$$\frac{Vp_{ext}}{Vp_{ret}} = \frac{Ap}{A_r} - 1 \qquad \text{--(4)}$$

We see that when the piston area equals two times the rod area, the extension and retraction speeds are equal. In general, the greater the ratio of piston area to rod area, the greater the ratio of extending speed to retracting speed.

Load-Carrying Capacity during Extension :

It should be noted that the load-carrying capacity of a regenerative cylinder during extension is less than that obtained from a regular double-acting cylinder. The load-carrying capacity (F_{load}) for a regenerative cylinder equals the pressure multiplied by piston rod area rather than the pressure multiplied by piston area. This is due to the same system pressure acting on both sides of the piston during the extending stroke of the regenerative cylinder..

Thus $F_{load} = PA_r$

Thus, the power obtained from the regenerative cylinder is less because the extending speed is increased at the expense of load-carrying capacity.

5.4 **Regenerative center in Drilling Machine:**

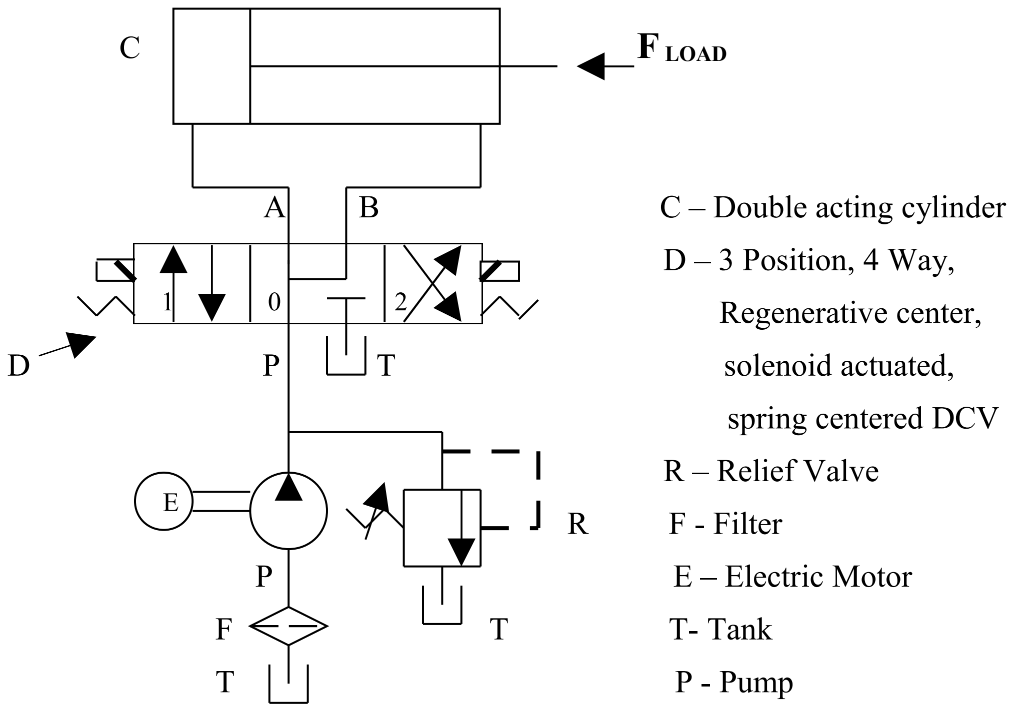


Fig 5.6 **Application in Drilling Machine**

Figure 5.6 shows an application of regenerative circuit in a drilling machine. Here a 3-position, 4-way , regenerative center directional control valve is used.

When the DCV is in the spring-centered position, port P is connected to A and B and tank port T is blocked. In this position pump flow goes to A and flow from rod end of the cylinder also joins the pump flow to gives rapid spindle advance (no work is done during this period)

Why does the spring-centered position give rapid extension of the cylinder (drill spindle)? The reason is simple. Oil from the rod end regenerates with the pump flow going to the blank end. This effectively increases pump flow to the blank end of the cylinder during the spring-centered mode of operation. Once again we have a regenerative cylinder. It should be noted that the cylinder used in a regenerative circuit is actually a regular double-acting cylinder. What makes it a regenerative cylinder is the way it is hooked up in the circuit. The blank and rod ends are connected in parallel during the extending stroke of a regenerative center.

When the DCV shifts to 1^{st} position, P is connected to A and B to T gives slow feed (extension) when the drill starts to cut into the work piece. Similarly when the DCV shifts to 2^{nd} position, P is connected to B and A is connected to T, since the ring area is less the cylinder will have fast return motion.

Problem 2. A double acting cylinder is hooked up in the regenerative circuit. The relief valve setting is 70 bars. The piston area is 0.016 m^2 and the rod area is 0.0045 m^2. If the pump flow is 0.0013m^3 / s, find the cylinder speed and load- carrying capacity for the

a. Extending stroke

b. Retracting stroke.

Solution:

Relief valve pressure setting, $p = 70$ bars $= 70 * 10^5$ N /m^2

Piston area, $A_p = 0.016$ m^2

Rod area, $A_r = 0.0045$ m^2

Pump flow, $Q_p = 0.0013$ m^3/s

c) Extending Stroke:

Cylinder speed, $v_{p\ ext} = Q_p / A_r$

(Regenerative Speed) $= 0.0013 / 0.0045$

$=$ **0.29 m / s**

Load carrying capacity, $F_{load} = p * A_r$

$= 70 * 10^5 * 0.0045$

$= 31500$ N $=$ **31.5kN**

d) Retracting Stroke:

Cylinder Speed, $v_{p\ Ret} = Q_p / (A_P - A_r)$

$= 0.0013 / (0.016 - 0.0045)$

$=$ **0.113 m / s**

—

Load carrying capacity, $F_{load} = p * (A_P - A_r)$

$= 70 * 10^5 * (0.016 - 0.0045$

$= 80500$ N $=$ **80.5kN**

3. A double acting cylinder is hooked up in the regenerative circuit. The relief valve setting is 100 bars and the pump flow is 0.0016m^3 / s. If the regenerative and retracting speed are equal to 0.25m/ s, find the piston and rod area and also load- carrying capacity for the

a. Extending stroke
b. Retracting stroke

Solution:

Relief valve pressure setting, $p = 105$ bars $= 105 * 10^5$ N /m^2

Pump flow, $Q_p = 0.0016$ m^3/s, Speed = 0.25 m/ s

We have regenerative cylinder speed, $v_{p\ ext} = Q_p / A_r$

Therefore Rod area, $Ar = Q_p / v_{p\ ext}$

$= 0.0016 / 0.25$

$=$ **0.0064m^2**

- Piston Area, $A_P = 2\ A_r = 2 * 0.0064 =$ **0.0128 m^2**

a) Extending Stroke:

Load carrying capacity, $F_{load} = p * A_r$

$= 105 * 10^5 * 0.0064$

$= 67200$ N = **67.2kN**

b) Retracting Stroke:

Load carrying capacity, $F_{load} = p * (A_P - A_r)$

$= 105 * 10^5 * (0.0128 - 0.0065)$

= **67200 N = 67.2kN**

5.5 **Pump – unloading circuit** :

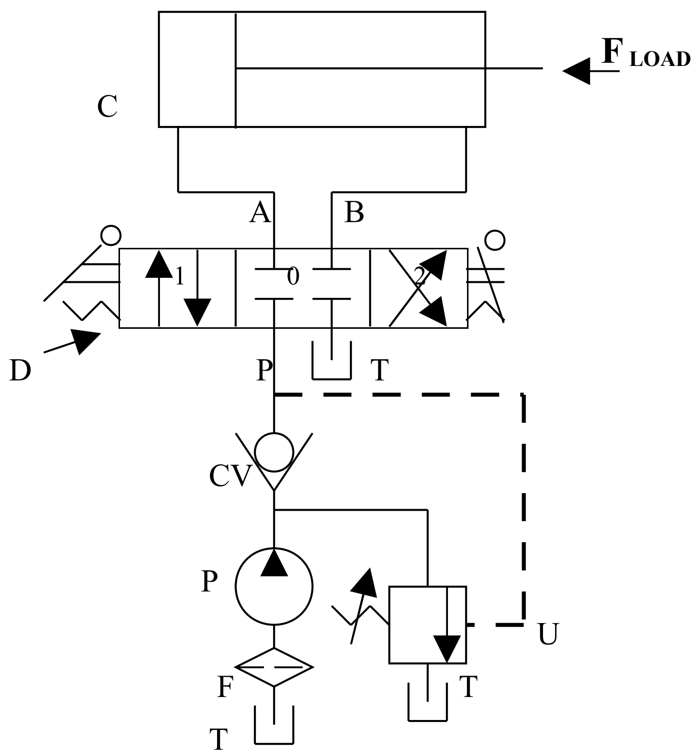


Fig 5.7. **Pump unloading circuit.**

C = Double acting cylinder

P = Pump

T = Tank

F = Filter

U = unloadingValve

D =3-position, 4 way ,closed center, Manually operated and
Spring Centered DCV

In Fig. 5.7 we see a circuit using an unloading valve to unload a pump. The unloading valve opens when the cylinder reaches the end of its extension stroke because the check valve keeps high-pressure oil in the pilot line of the unloading valve. When the DCV is shifted to retract the cylinder, the motion of the piston reduces the pressure in the pilot line of the unloading valve. This resets the unloading valve until the cylinder is fully retracted, at which point the unloading valve unloads the pump. Thus, the unloading valve unloads the pump at the ends of the extending and retraction strokes as well as in the spring-centered position of the DCV.

5.6 Double Pump Hydraulic system (Hi – Lo circuit)

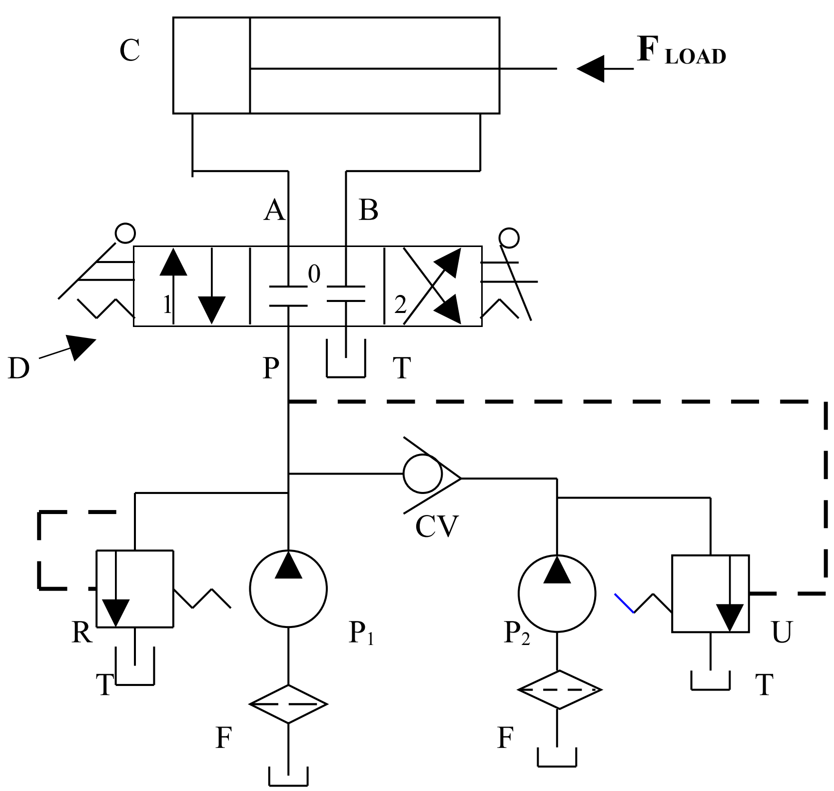


Fig 5.8 Double pump Hydraulic system

P_1 -- Low discharge, High pressure pump
P_2 -- High discharge, Low pressure pump
R -- Relief valve
U – Unloading valve
T - Tank

CV – Check valve
D – 3 Position, 4 Way, closed center, manual actuated DCV
C- Double acting cylinder
F - Filter

Figure 5-8 shows a circuit that uses two pumps, one high-pressure, low-flow pump and the other low-pressure, high-flow pump. One can find application in a punch press in which the hydraulic ram must extend rapidly over a large distance with very low pressure but high flow requirements. However, during the short motion portion when the punching operation occurs, the pressure requirements are high due to the punching load. Since the cylinder travel is small during the punching operation, the flow-rate requirements are also low.

The circuit shown eliminates the necessity of having a very expensive high-pressure, high-flow pump. When the punching operation begins, the increased pressure opens the unloading valve to unload the low-pressure pump. The purpose of the relief valve is to protect the high-pressure pump from overpressure at the end of the cylinder stroke. The check valve protects the low-pressure pump from high pressure, which occurs during the punching operation, at the ends of the cylinder stroke, and when the DCV is in its spring-centered mode.

5.7 Counter Balance Valve Application :

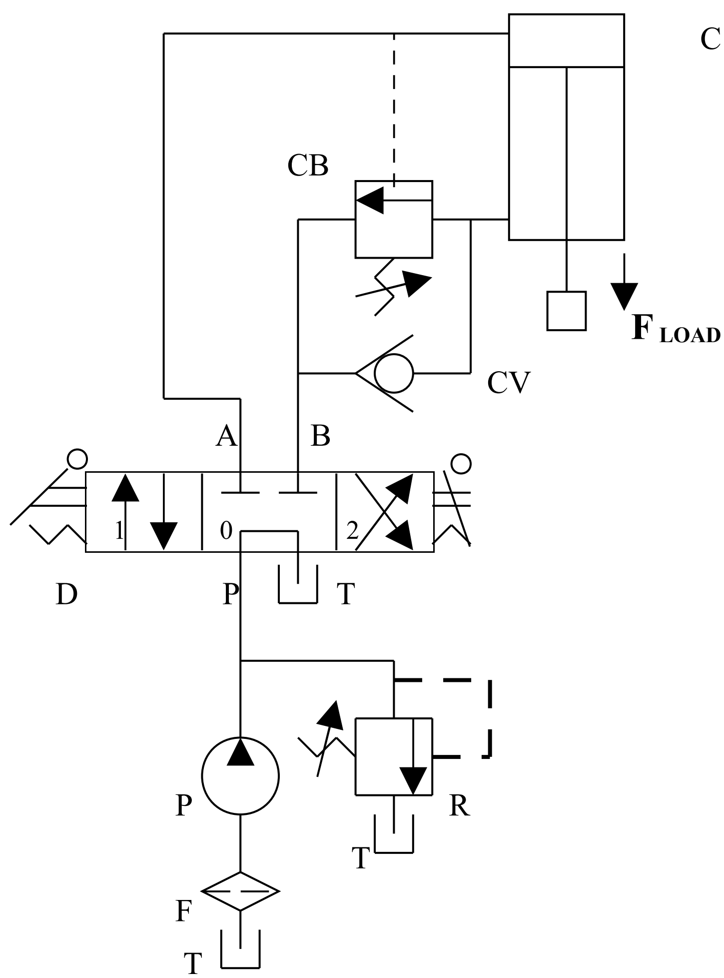


Fig 5.9. **Counter balance application**

C = Double acting cylinder mounted vertically

P = Pump

CB = Counter Balance Valve

CV = Check Valve

T = Tank

F = Filter

R = Relief Valve

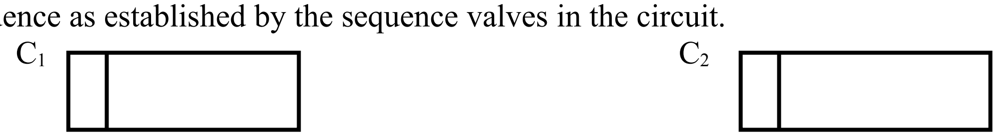
D =3-position, 4 way , Tandem center, Manually operated and Spring return DCV

Figure 5.9 illustrates the use of a counterbalance or back-pressure valve to keep a vertically mounted cylinder in the upward position while the pump is idling i.e when the DCV is in its center position. During the downward movement of the cylinder the counterbalance valve is set to open at slightly above the pressure required to hold the piston up (check valve does not permit flow in this direction). The control signal for the counterbalance valve can be obtained from the blank end or rod end of the cylinder. If derived from the rod end, the pressure setting of the counter balance valve equals $F_L/(A_P-A_r)$. If derived from blank end the pressure setting equals F_L / A_P. This pressure is less and hence usually it has to be derived from blank end. This permits the cylinder to be forced downward when pressure is applied on the top. The check valve is used to lift the cylinder up as the counterbalance valve is closed in this direction. The Tandem -center directional control valve unloads the pump. The DCV is a manually -actuated, spring-centered valve with tandem-center flow path configuration.

5.8 <u>Hydraulic Cylinder Sequencing circuit:</u>

A sequence valve causes operations in a hydraulic circuit sequentially. Figure 5.10 is an example where two sequence valves are used to control the sequence of operations of two double-acting cylinders C_1 and C_2 . When the DCV is shifted into its 1st position, the left cylinder extends completely, and only when the left cylinder pressure reaches the pressure setting of sequence valve, the valve opens and then the right cylinder extends. If the DCV is then shifted into its 2nd position, the right cylinder retracts fully, and then the left cylinder retracts. Hence this sequence of cylinder operation is controlled by the sequence valves. The spring centered position of the DCV locks both cylinders in place.

One can find the application of this circuit in press circuit. For example, the left cylinder the clamping cylinder C_1 could extend and clamp a workpiece. Then the right cylinder C_2, the punching cylinder extends to punch a hole in the workpiece. The right cylinder then retracts the punch , and then the left cylinder retracts to declamp the workpiece for removal. Obviously these machining operations must occur in the proper sequence as established by the sequence valves in the circuit.



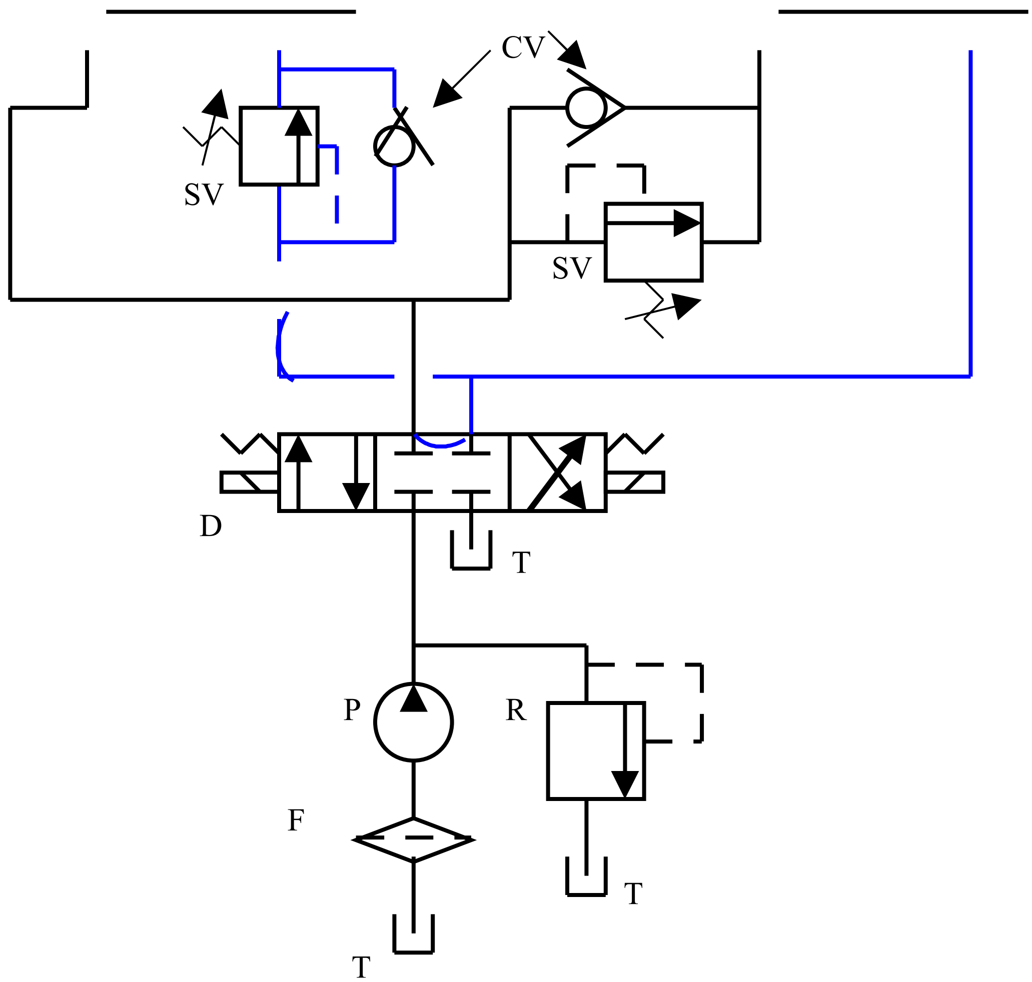


Fig 5.10. **Cylinder Sequencing circuit**

Operation sequence is : $C_1 + C_2 + C_2 - C_1 -$

+ is Extension of piston

- is Retraction of piston

C_1 & C_2 = Double acting cylinder

P = Pump ; SV = Sequence Valve ;

R = Relief Valve

CV = Check Valve ;

T = Tank ; F = Filter

D =3-position, 4 way , Tandem center, Solenoid operated and Spring return DCV

Figure 5.11 Automatic Cylinder Reciprocating System :

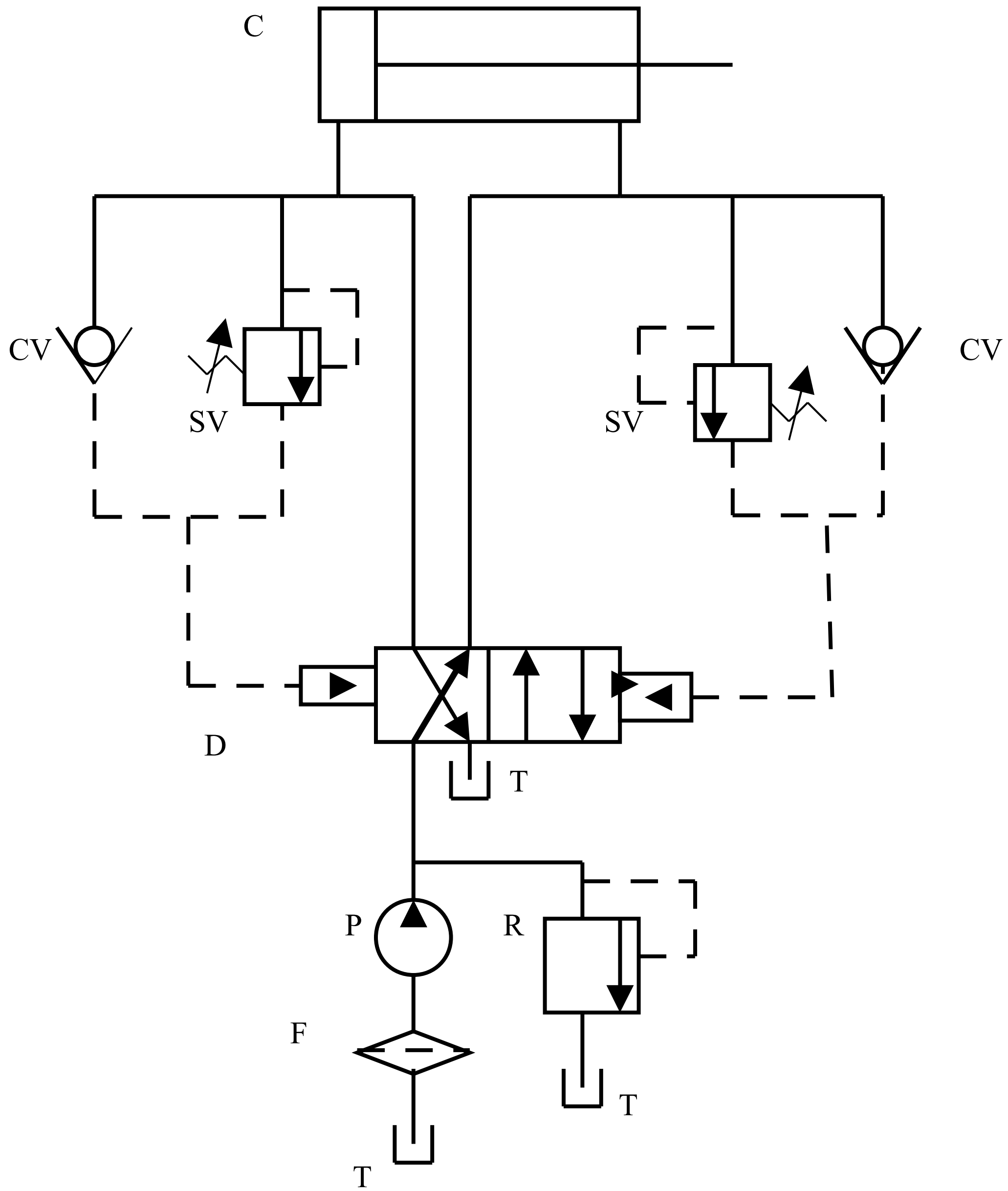


Fig 5.11. Automatic Cylinder Reciprocating System using Sequence valves

C = Double acting cylinder

P = Pump

SV = Sequence Valve

CV = Check Valve ; R = Relief Valve

T = Tank ; F = Filter

D =2-position, 4 way , pilot operated DCV

Figure 5.11 shows a circuit that produces continuous automatic reciprocation of a hydraulic cylinder. This is accomplished by using two sequence valves, each of which senses a stroke completion by the corresponding buildup of pressure. Each check valve and corresponding pilot line prevents shifting of the four-way valve until the particular stroke of the cylinder has been completed. The check valves are needed to allow pilot oil to leave either end of the DCV while pilot pressure is applied to the opposite end. This permits the spool of the DCV to shift as required.

An alternative circuit diagram is shown in Figure 5.12 . In the circuit as the cylinder reciprocates, the pilot directional valve is operated by solenoid DCV to supply fluid to either end of the main four way directional valve. This causes the machine to reciprocate automatically, and it will not stop until pump is shut off.

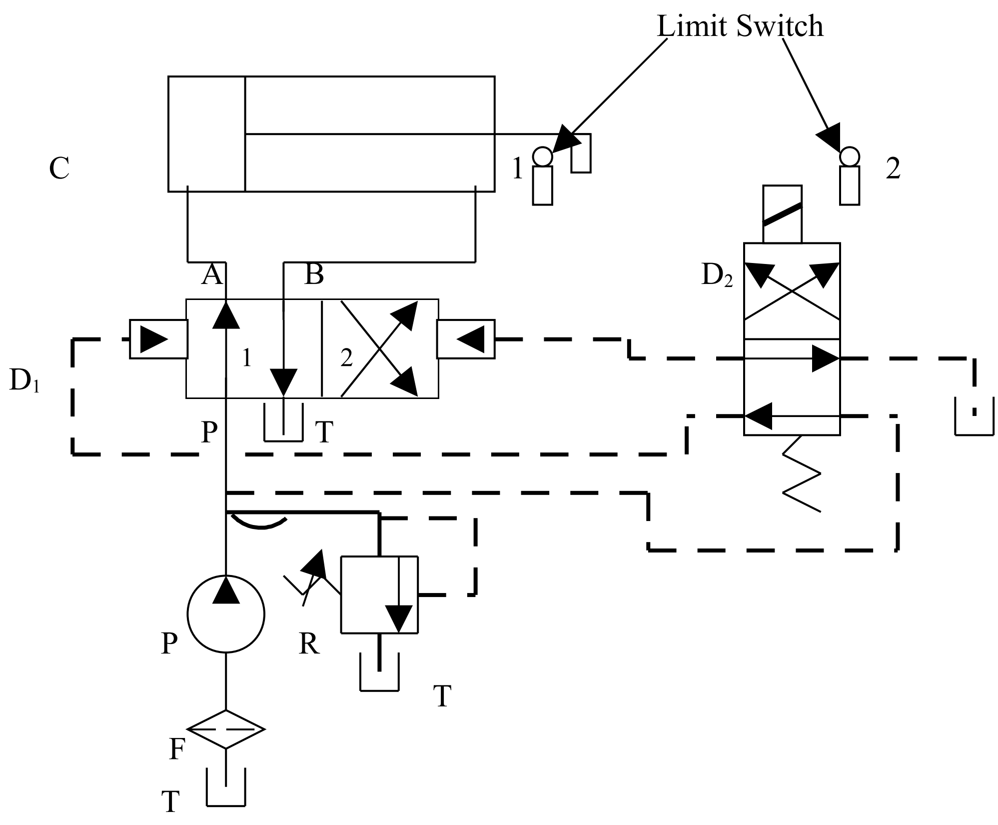


Fig 5.12. **Automatic Cylinder Reciprocating System using DCV's.**

C = Double acting cylinder

P = Pump

R = Relief Valve

T = Tank ; F = Filter

D_1 =2-position, 4 way , pilot operated DCV

D_2 =2-position, 4 way , mechanical operated DCV

The limit switch 1 and 2 will alternatively energize and deenergize solenoid ,thereby changing the directional of fluid flow into the pilot

5.10. Locked cylinder using pilot check valves:

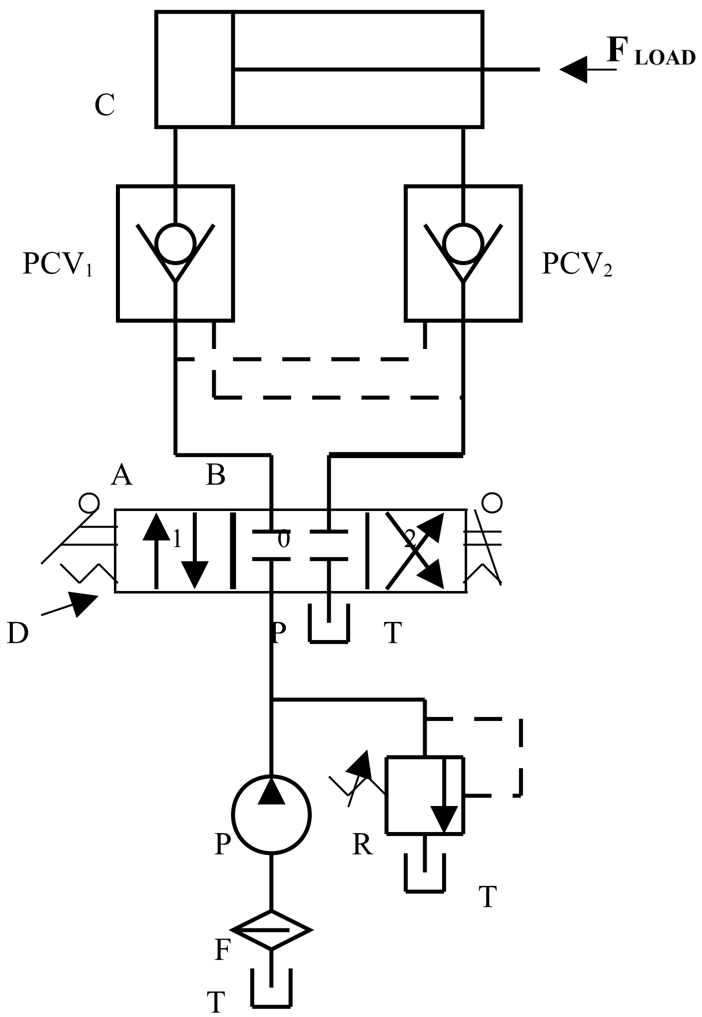


Fig 5.13. **Locked Cylinder using Pilot Check Valve.**

C = Double acting cylinder

P = Pump

PCV_1 & PCV_2 = Pilot Check Valve

R = Relief Valve

T = Tank ; F = Filter

D =3-position, 4 way , closed center, manually operated DCV

In many cylinder applications, it is necessary to lock the cylinder so that its piston cannot be moved due to an external force acting on the piston rod. One method for locking a cylinder in this fashion is by using pilot check valves, as shown in Fig 5.13. The cylinder can be extended and retracted as normally done by the action of the directional control valve. If regular check valves were used, the cylinder could not be extended or retracted by the action of the DCV. An external force, acting on the piston rod, will not move the piston in either direction because reverse flow through either pilot check valve is not permitted under these conditions.

5.12. CYLINDER SYNCHRONIZING CIRCUITS :

a. **Cylinder connected in series** : Figure 5.14 is a very interesting circuit, which seems to show to show how two identical cylinders can be synchronized by piping them in parallel. However, even if the two cylinders are identical, it would be necessary for the loads on the cylinders to be identical in order for them to extend in exact synchronization. If the loads are not exactly identical (as is always the case), the cylinder with the smaller load would extend first because it would move at a lower pressure level. After this cylinder has fully completed its stroke, the system pressure will increase to the higher level required to extend the cylinder with the greater load. It should be pointed out that no two cylinders are really identical. For example, differences in packing friction will vary from cylinder to cylinder. This alone would prevent cylinder synchronization for the circuit of Fig. 5.14.

C_1 & C_2 = Double acting cylinder

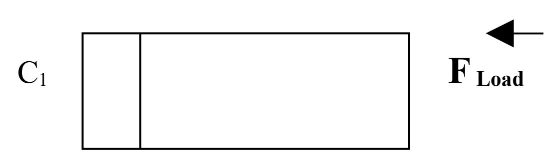
P = Pump

T = Tank

F = Filter

R = Relief Valve

D =3-position, 4 way , Tandem center, Solenoid operated and Spring return DCV



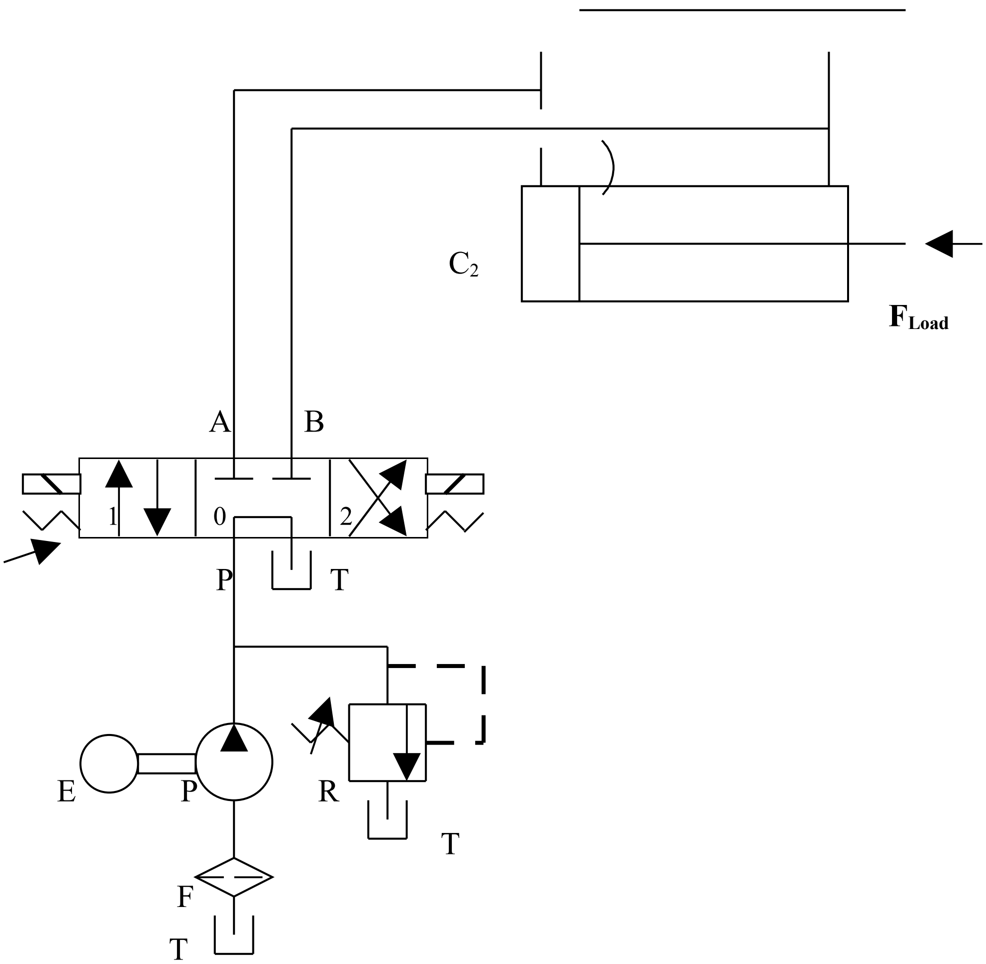


Fig 5.14. **Cylinder hooked in parallel for synchronizing**

(will not operate)

b. **Cylinders connected in Series :**

The circuit of Fig. 5.15 shows a simple way to synchronize two cylinders. Fluid from the pump is delivered to the blank end of cylinder 1, and fluid from the rod end of cylinder 1 is delivered to the blank end of cylinder 2. Fluid returns to the tank from the rod end of cylinder 2 via the DCV. Thus, the cylinders are hooked in series. For the two cylinders to be synchronized, the piston area of cylinder 2 must equal the difference between the areas of the piston and rod for cylinder 1. It should also be noted that the pump must be capable of delivering a pressure equal to that required for the piston of cylinder 1 by itself to overcome the loads acting on both cylinders. It should be noted that the pressure at the blank end of cylinder 1 and the rod end of cylinder 2 are equal as per Pascal's law.

$$P_1Ap_1 - P_2(Ap_1 - A_{r1}) = F_1$$

And

$$P_2A_{p2} - P_3(A_{p2}-A_{r2}) = F_2$$

Adding both equations and noting that $A_{P2} = A_{p1} - A_{R1}$ and that $P_3 = 0$ (due to the drain line to the tank), we obtain the desired result

$$P_1Ap_1 = F_1 + F_2$$

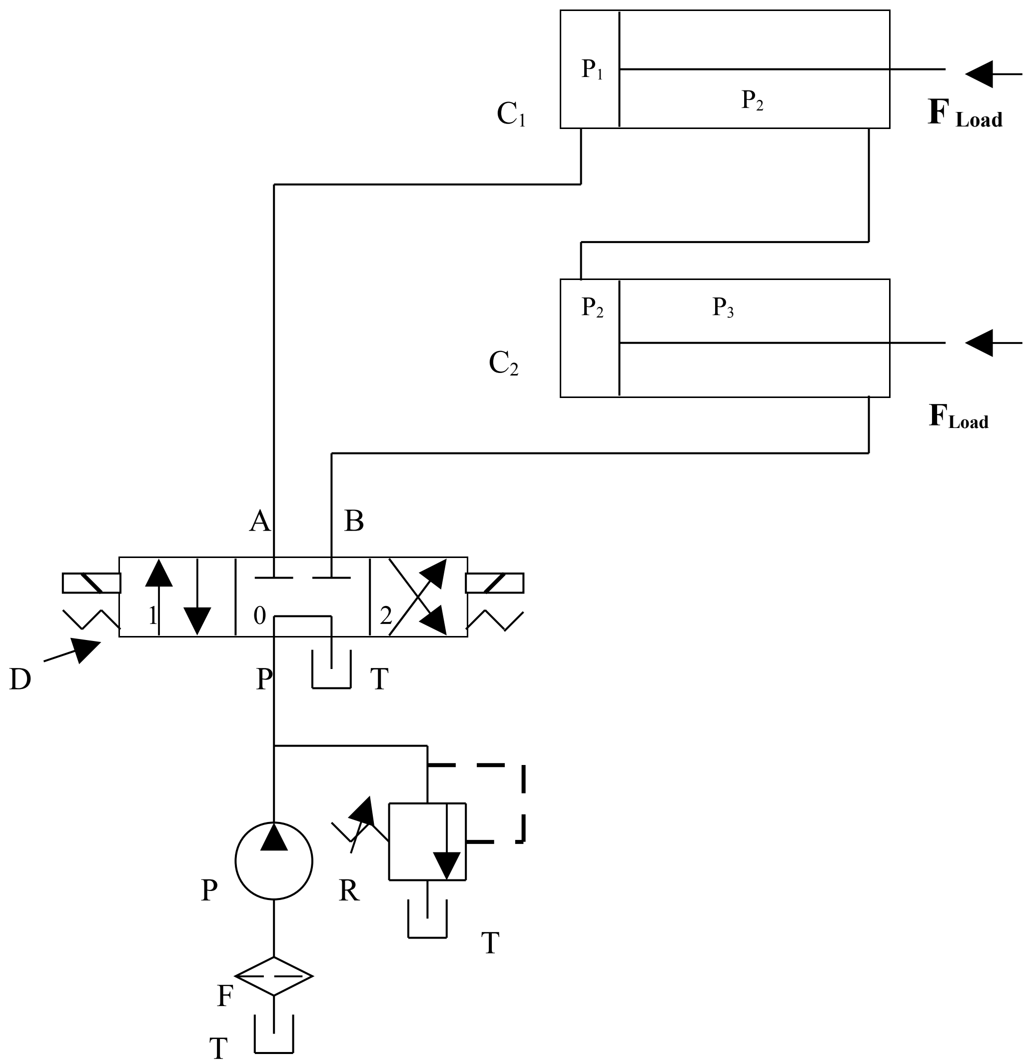


Fig 5.15. **Cylinder hooked in Series for synchronizing**

(Will Operate)

C_1 & C_2 = Double acting cylinder hooked in series

P = Pump

T = Tank

F = Filter

R = Relief Valve

D =3-position, 4 way , Tandem center, Solenoid operated and Spring return DCV

HYDRAULIC CIRCUIT DESIGN AND ANALYSIS -2

5.13 Speed control of Hydraulic Cylinder: Speed control of a hydraulic cylinder is accomplished using a flow control valve. A flow control valve regulate the speed of the cylinder by controlling the flow rate to and of the actuator.

There are 3 types of speed control:

- Meter- in circuit (Primary control)
- Meter-out circuit (Secondary control)
- Bleed - off circuit (By pass control)

1. **Meter – in Circuit** : In this type of speed control, the flow control valve is placed between the pump and the actuator. Thereby, it controls the amount of fluid going into the actuator. Fig 5.16 shows meter-in circuit.

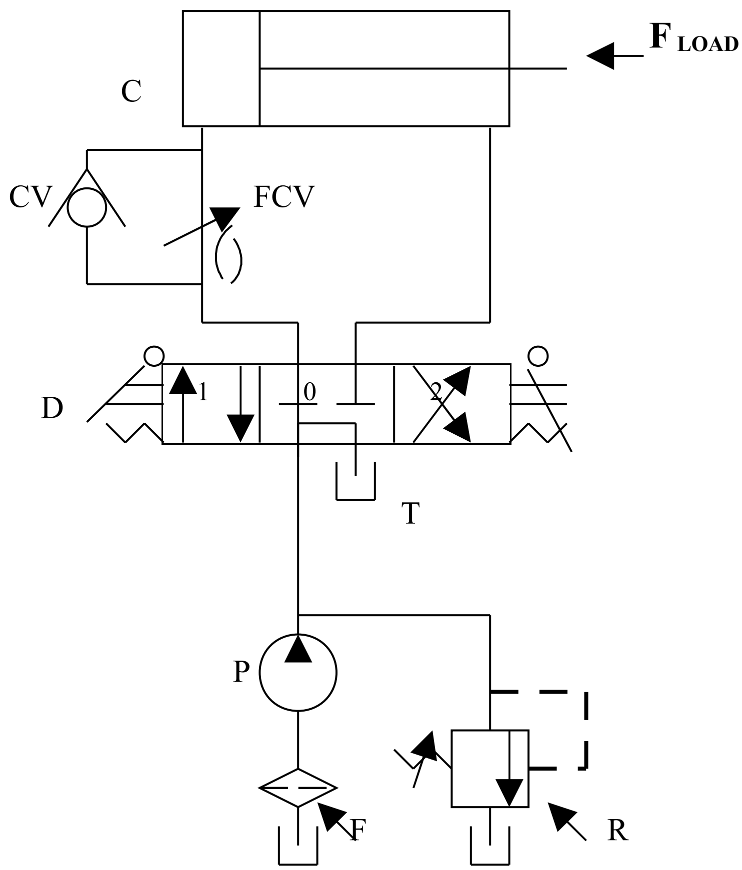


Fig 5.16. **Meter – in circuit.**

C = Double acting cylinder ; P = Pump ; T = Tank ; F = Filter

R = Relief Valve; CV = Check Valve ; FCV = Flow control Valve

D =3-position, 4 way ,Tandem center, Manually operated ,Spring Centered DCV

When the directional control valve is actuated to the 1st position, oil flows through the flow control valve to extend the cylinder. The extending speed of the cylinder depends on the setting (percent of full opening position) of the flow control valve.

When the directional control valve is actuated to the 2nd position, the cylinder retracts as oil flows from the cylinder to the oil tank through the check valve as well as the flow control valve.

Analysis of Extending Speed Control :

During the extension stroke, if the flow control valve is fully open, all the flow from the pump goes to the cylinder to produce maximum cylinder speed. As the flow control valve is partially closed its pressure drop increases. This causes an increase in pressure p_1. Continued closing of the flow control valve ultimately results in pressure p_1 reaching and exceeding the cracking pressure of the pressure relief valve (PRV). The result is a slower cylinder speed since part of the pump flow goes back to the oil tank through the PRV setting and the amount of pump flow that is not desired by the cylinder flows through the PRV. An analysis to determine the extending speed is given as follows:

The flow rate to the cylinder equals pump flow rate minus the flow rate through the PRV.

$Q_{cyl} = Q_{pump} - Q_{PRV}$

The flow rate through the flow control valve (FCV) is governed by

$Q_{FCV} = C_V \sqrt{\Delta P / S_g} = C_V \sqrt{(p_1 - p_2) / S_g}$

Where ΔP = pressure drop across FCV

C_V = capacity coefficient of FCV

S_g = specific gravity of oil

Pressure $p_1 = p_{PRV}$ = Relief valve pressure setting

Also, pressure $p_3 = 0$ (ignoring small frictional pressure drop in drain line from rod end of cylinder to oil tank).

Pressure p_2 can be obtained by summing forces on the hydraulic cylinder.

$p_2 A_{piston} = F_{load}$ or $p_2 = F_{load} / A_{piston}$ ---(a)

Finally the extending speed of the cylinder is found.

$V_{cyl} = Q_{cyl} / A_{piston} = Q_{FCV} / A_{piston}$ ------ (b)

using Eqs. (a) and (b) yields the final result.

$$V_{cyl} = \frac{C_v}{A_{piston}} \sqrt{\frac{P_{PRV} - F_{load} / A_{piston}}{S_g}} \qquad \text{------ (1)}$$

As can be seen by Eq. 1, by varying the setting of the flow control system, and thus the value of C_V, the desired extending speed of the cylinder can be achieved.

2. **Meter – out Circuit** : In this type of speed control, the flow control valve is placed between the actuator and the tank . Thereby, it controls the amount of fluid going out of the actuator. Fig 5.17 shows a meter-out circuit.

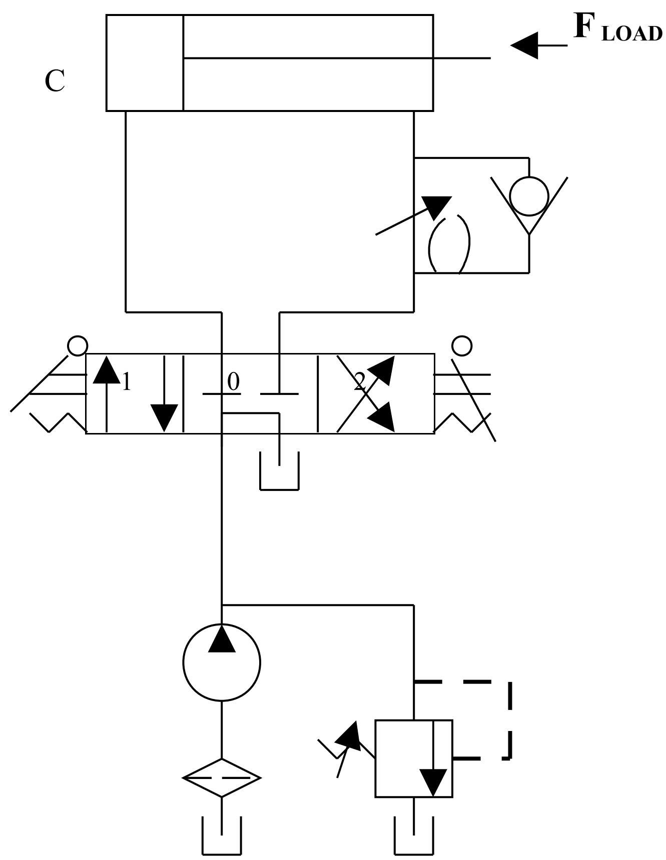


Fig 5.17. **Meter – out Circuit.**

C = Double acting cylinder; P = Pump ; T = Tank; F = Filter

R = Relief Valve; CV = Check Valve ; FCV = Flow control Valve

D =3-position, 4 way ,Tandem center, Manually operated and Spring Centered DCV

Meter-in systems are used primarily when the external load opposes the direction of motion of the hydraulic cylinder. An example of the opposite situation is the case of a weight pulling downward on the piston rod of a vertical cylinder. In this case the weight would suddenly drop by pulling the piston rod down if a meter-in system is used even if the flow control valve is completely closed. Thus, the meter-out system is generally preferred over the meter-in type. One drawback of a meter-out system is the possibility of excessive pressure buildup in the rod end of the cylinder while it is extending. This is due to the magnitude of back pressure that the flow control valve can create depending on its nearness to being fully closed as well as the size of the external load and the piston-to-rod area ratio of the cylinder. In addition an excessive pressure buildup in the rod end of the cylinder results in a large pressure drop across the flow control valve. This produce the undesirable effect of a high heat generation rate with a resulting increase in oil temperature.

3. **Bleed – off Circuit** : In this type of speed control, the flow control valve is placed between the pressure line and return line . Thereby, it controls the fluid by bleeding off the excess not needed by the working cylinder. Fig 5.18 shows the bleed-off circuit.

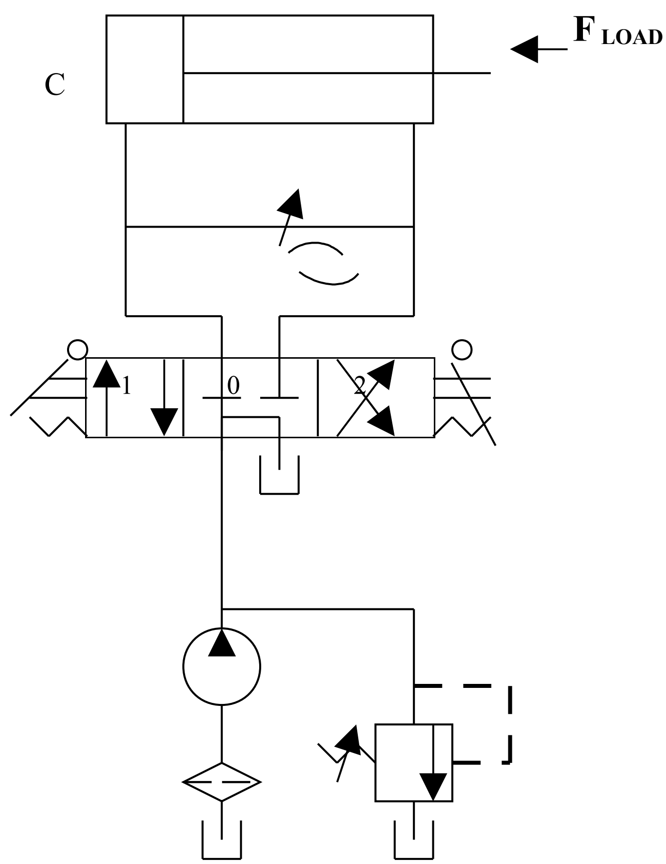


Fig 5.18. **Bleed – 0ff Circuit.**

C = Double acting cylinder; P = Pump ; T = Tank; F = Filter

R = Relief Valve; FCV = Flow control Valve

D =3-position, 4 way ,Tandem center, Manually operated , Spring Centered DCV

This type of flow control is much more efficient than the inlet restricting type for meter-in, because the bypass feature allows fluid to be exhausted to the tank at just slightly higher pressure than that necessary to do the work. With the meter-in type, pump delivery not used would discharge over the main relief valve at maximum pressure.

Problem 1 : An actuator forward speed is controlled by a meter-in circuit. The pressure setting of relief valve is 50 bar and the pump discharge = 30 litres /min. The cylinder has to carry a load of 3600 N during the forward motion. The area of piston is 15 cm^2 and rod area = $8cm^2$. The flow control valve is set to allow only 10 litres/ min. Calculate the power input to motor, forward speed and return speed and efficiency of the circuit.

Solution:

$p_2 = F_{load} / A_P = 3600 / 0.0015$

$= 2400000 N/m^2 = 24$ bar.

$\mathbf{\Delta P} = 50\text{-}24 = 26$ bar

- Power input to the motor $= p_1 * Q_P / \eta_m$

  $= 50 * 10^5 * 0.0005 / 0.85$

  $= 2941$ watts= **2.94 kW** **Ans**

- Forward speed, $v_F = Q_{FCV} / A_P$

  $= 0.00016 / 0.0015$

  $=$ **0.16 m/s** **Ans**

- Return speed , $v_r = Q_P / (A_P - A_r)$

  $= 0.0005 / (0.0015 - 0.0008)$

  $=$ **0.71m/s** **Ans**

- η = output / input $= p_2 Q_{FCV} / p_1 Q_P$

  $$= \frac{24*10^5*0.00016}{50*10^5*0.0005}$$

  $= 0.1536 =$ **15.36%** **Ans**

5.14 Speed control of Hydraulic Motor: Figure 5.19 shows a circuit where speed control of a hydraulic motor (Bi -directional motor) is accomplished using a flow control valve to control the fluid flow to the motor.

In the spring-centered position of the tandem four-way valve, the motor is hydraulically locked. When the four-way valve is actuated into the 1st position, the motor rotates in one direction. Its speed can be varied by adjusting the setting of the throttle of the flow control valve. In this way the speed can be infinitely varied as the excess oil goes to the tank through the pressure relief valve. When the four-way valve is deactivated, the motor stops suddenly and becomes locked. When the 2nd position of the four-way valve is in operation, the motor turns in the opposite direction. The pressure relief valve provides overload protection if, for example, the motor experiences an excessive torque load.

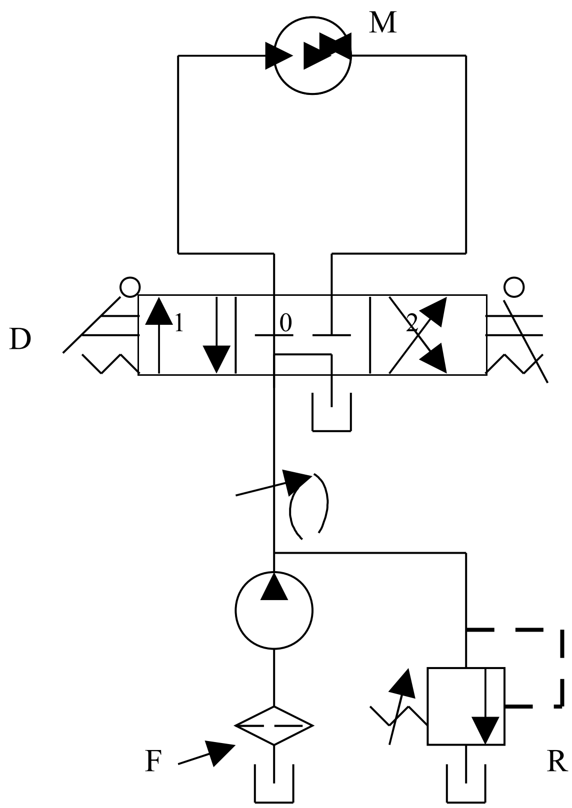


Fig 5.19. **<u>Speed control of Hydraulic motor using Flow control valve.</u>**

M = Bi-directional Hydraulic motor ; P = Pump ; T = Tank; F = Filter

R = Relief Valve; FCV = Flow control Valve

D =3-position, 4 way ,Tandem center, Manually operated and Spring Centered DCV

The speed of hydraulic motor can be controlled either by meter-in control or meter- out control.

Figure 5.20 shows a unidirectional hydraulic motor speed is controlled by a meter- in circuit. Here the flow control valve is placed between the pump and motor.

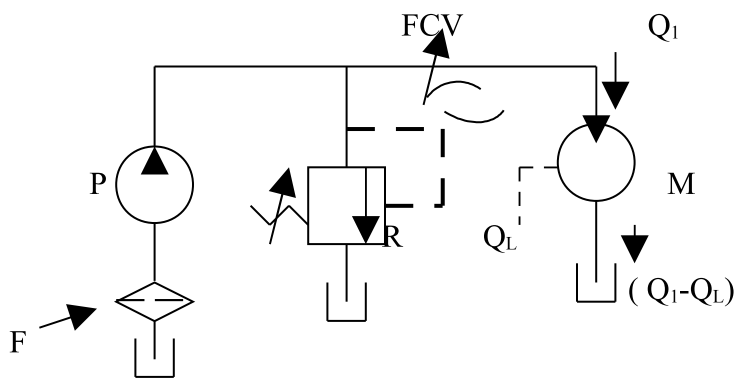


Fig 5.20. **<u>Meter- in Speed control of Hydraulic motor</u>**

M = Uni-directional Hydraulic Motor ; P = Pump ; T = Tank; F = Filter
R = Relief Valve; FCV = Flow control Valve

Figure 5.21 shows a unidirectional hydraulic motor speed is controlled by a meter- out circuit. Here the flow control valve is placed between the motor and tank.

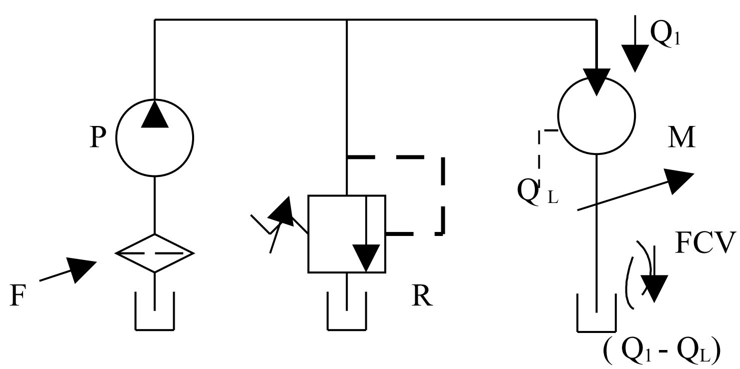


Fig 5.20. **<u>Meter- out Speed control of Hydraulic motor</u>**

M = Uni-directional Hydraulic Motor ; P = Pump ; T = Tank; F = Filter
R = Relief Valve; FCV = Flow control Valve

We know that the volumetric efficiency of the motor is given by

$$\eta_{Vol} = \frac{\text{Theoretical flow rate the motor should consume}}{\text{Actual flow rate consumed by motor}}$$

$$\eta_{Vol} = Q_T / Q_A$$

Due to leakage, a hydraulic motor consumes more flow rate than it should theoretically consume. The theoretical flow rate is the flow rate a hydraulic motor would consume if there were no leakage. If $Q_1(=Q_A)$ is the flow of fluid to the motor, and Q_L the leakage , Then Q_T is equal to $Q_1 - Q_L$

Then $\eta_{Vol} = (Q_1 - Q_L) / Q_1$

In meter- in control Q_1 is maintained constant despite varying load , Q_L varies with load , therefore volumetric efficiency varies with load. Hence meter-in system will not give precise control of speed. Whereas in meter – out control Q_1 varies with load hence precise control of speed regardless of flow.

5.15 **<u>Hydrostatic Transmission</u> (HST)**: These are special cases of energy transmission system. It consists of a drive with hydraulic energy as input. Hydraulic motor convert hydraulic energy to mechanical energy. Hydrostatic transmission is a whole unit in which pumps and motors are designed to match (the speed torque characteristics) to get optimum transmission. The HST can be open or closed circuit. (fig 5.21)

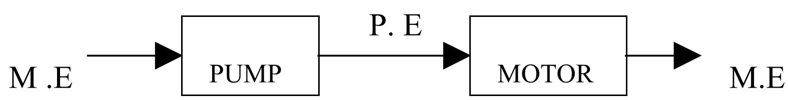


Fig 5.21 Hydrostatic transmission

<u>Open circuit HST</u> : Figure 5.22 shows open type circuit. They are called open circuit drives because the pump draws its fluid from a tank. Its output is then directed to a hydraulic motor and discharged from the motor back to the tank. In the closed circuit drive, exhaust oil from the motor is returned directly to the pump inlet.

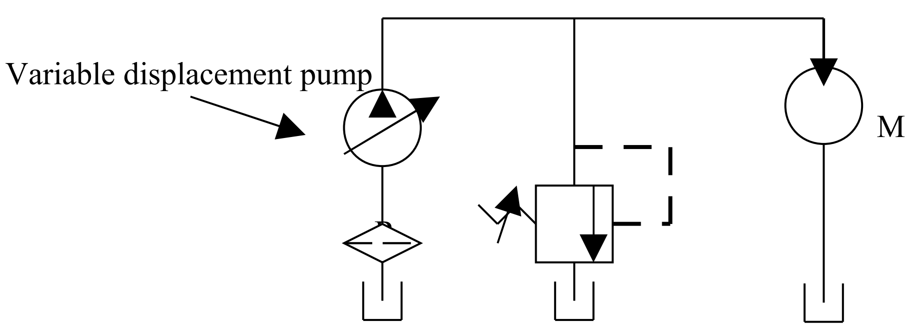


Fig 5.22 <u>Open circuit (open loop) HST</u>

Figure 5.23 shows a closed circuit that allows either direction of motor rotation. The feed pump is provided for replenishing the fluid in the circuit. The check valves prevent the oil flow from the main pump to the feed pump. Here two relief valve R_1 and R_2 are used to protect the main pump in both the direction of rotation.

The motor speed is varied by changing the pump displacement. The torque capacity of the motor can be adjusted by the pressure setting of the relief valve.

Closed circuit drives are available as completely integrated units with all the controls and valving enclosed in a single, compact housing.

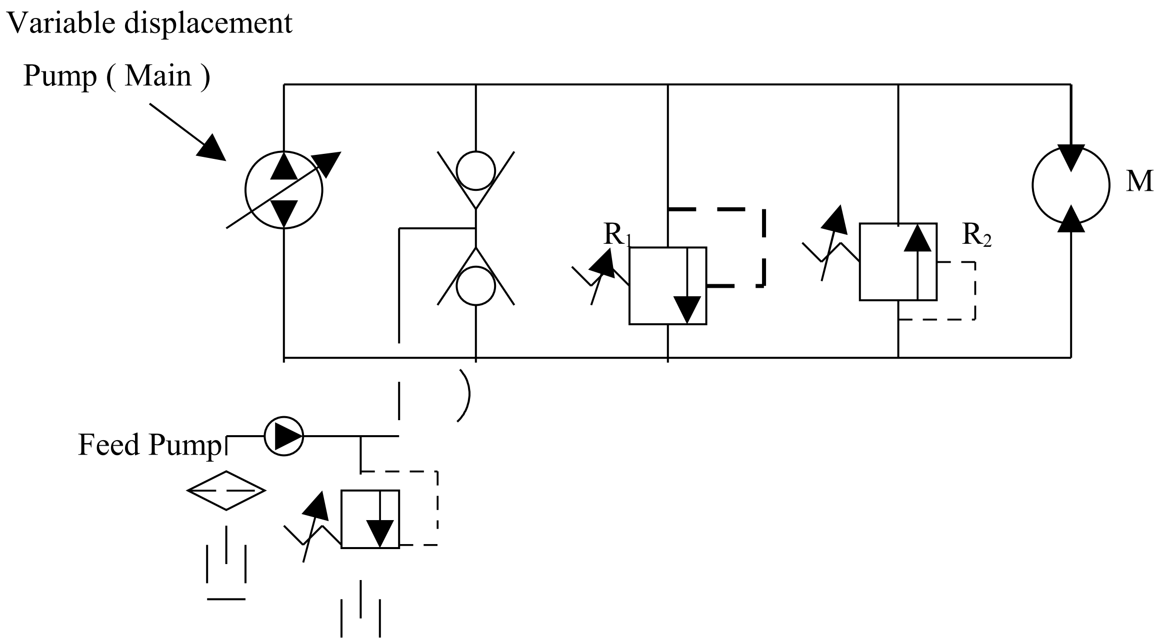


Fig 5.23 **Closed circuit (open loop) HST**

Performance:

1. Hydraulic power input , $P_{hyd} = p\,Q_P$ (Watts)

   where p = pressure setting of relief valve in N / m^2

   Q_P = pump theoretical flow rate = pump displacement (m^3/ rev) $*$ Speed (rps)

2. $Q_1 = Q_p * \eta_{VP}$ where η_{VP} = volumetric efficiency of the pump;

Q_1 = Actual flow rate to motor

3. Motor theoretical flow rate ,$Q_M = Q_1 * \eta_{VM}$

where η_{VM} = volumetric efficiency of the motor

1. Motor capacity , $C_M = Q_M$ / speed of motor (m^3/ rev)

2. Power delivered to motor, P_{hyd} = System pressure $* Q_1$ (Watts)

3. Mechanical power generated, $P_{Mech} = P_{hyd} * \eta_{VM} * \eta_{MM}$

where η_{MM} = mechanical efficiency of hydraulic motor

4. Actual Torque developed by motor, $T_a = P_{Mech} / 2\pi N$

Where N = speed of motor in rps

8. Ideal torque , $T_i = C_M * \Delta P / 2\pi$

9. Actual torque, $T_a = T_i * \eta_{MM}$

Problem 2. A Hydrostatic transmission operating at 70 bar pressure has the following characteristic for the pump and the motor:

Pump : Capacity of pump, $C_P = 82\ cm^3/rev$ (pump displacement)

Volumetric efficiency of pump, $\eta_{VP} = 82\ \%$

Mechanical efficiency of pump, $\eta_{MP} = 88\ \%$

Speed of pump , N = 500 rev / min

Motor: Capacity of motor, C_M = **?**

Volumetric efficiency of motor, $\eta_{VM} = 92\ \%$

Mechanical efficiency of motor, $\eta_{MM} = 90\ \%$

Desired speed of motor , N = 400 rev / min

Actual Torque, T_a = **?**

Solution :

Pump theoretical flow rate, Q_P = pump displacement * speed

$= 82 * 10^{-06} * 500 / 60$

$= 0.00068\ m^3 / sec$

Actual flow rate to the motor , $Q_1 = Q_P * \eta_{VP}$

$= 0.00068 * 0.82$

$= 0.00056 \ m^3 / sec$

Motor theoretical flow rate, $Q_M = Q_1 * \eta_{VM}$

$= 0.00056 * 0.92$

$= 0.000515 \ m^3 / sec$

- Motor capacity, $C_M = Q_M$ / speed of motor

  $= 0.000515 / 400/60$

  $=$ **$\underline{0.0000773 \ m^3 / sec}$** **Ans**

Power delivered to motor, P_{hyd} = system pressure $* Q_1$

$= 70 * 10^5 * 0.00056$

$= 3920$ Watts $= 3.92$ kW

Mechanical power generated, $P_{Mech} = P_{hyd} * \eta_{VM} * \eta_{MM}$

$= 3.92 * 0.92 * 0.90$

$= 3.246$ kW

- Actual Torque developed by motor, $T_a = P_{Mech} / 2\pi N$

  $$= 3.246 * 1000 / \left(2\pi * \frac{400}{60}\right)$$

  $=$ **$\underline{77.49 \ N\text{-} m}$** **Ans**

Ideal torque, $T_i = C_M * \Delta P / 2\pi$

$= 0.0000773 * 70 * 10^5 / 2\pi$

$= 86.1$ N-m

- Actual torque, $T_a = T_i * \eta_{MM}$

  $= 86.1 * 0.90$

  $=$ **$\underline{77.49 \ N\text{–}m}$** **Ans**

ACCUMULATORS AND ACCUMULATOR CIRCUITS:

A hydraulic accumulator is a device that stores the potential energy of an incompressible fluid held under pressure by an external source against some dynamic force. The dynamic force can come three different sources : Gravity, Mechanical Springs, and Compressed gases. The stored potential energy in the accumulator is a quick secondary source of fluid power capable of doing useful work as required by the system.

There are three basic types of accumulator used in hydraulic system. They are:

1. Weight – Loaded, or gravity, type
2. Spring -Loaded type
3. Gas – Loaded type

1. **Weight – Loaded Accumulator :** This type consists of a vertical, heavy- wall steel cylinder, which incorporates a piston with packing to pressure leakage (Fig 5.24) . A dead weight is attached to the top of the piston. The force of gravity of the dead weight provides the potential energy in the accumulator. This type of accumulator creates a constant fluid pressure throughout the full volume output of the unit regardless of the rate and quantity of output. The main disadvantage of this type of accumulator is extremely large size and heavy weight which makes it unsuitable for mobile equipment.

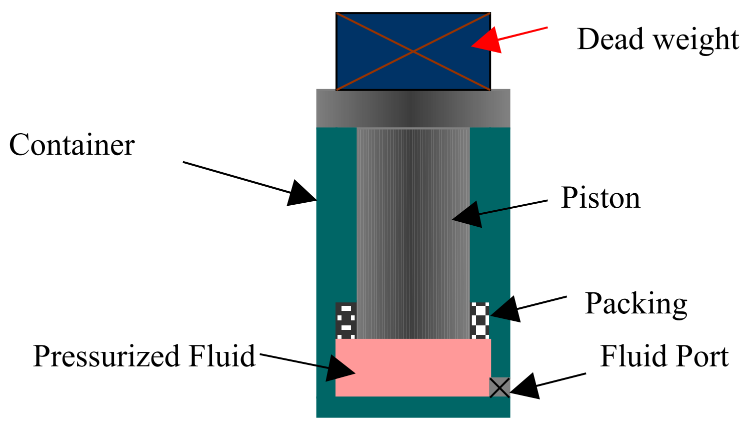
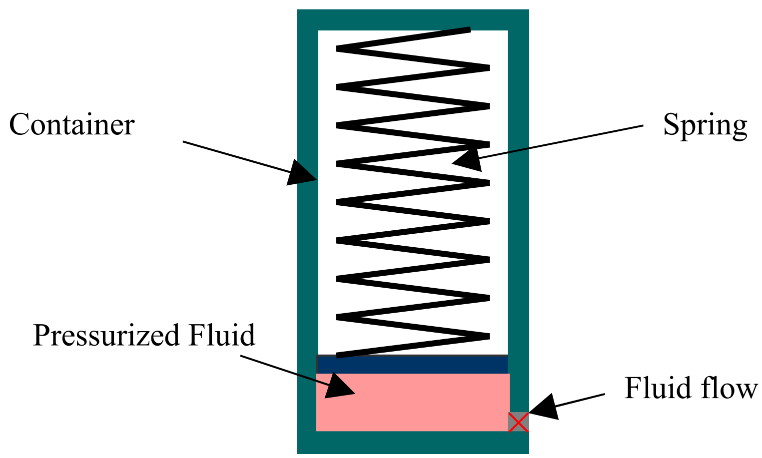


Fig 5.24 Weight – Loaded accumulator

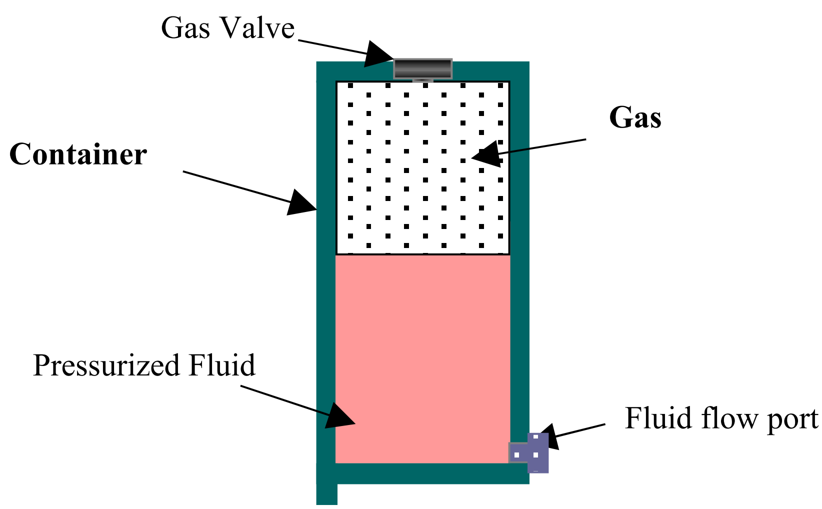
2. **Spring – Loaded Accumulator :** A spring loaded accumulator is similar to the weight – loaded type except that the piston is preloaded with a spring as shown in fig 5.25. The spring is the source of energy that acts against the piston, forcing the fluid into the hydraulic system. The pressure generated by this type of accumulator depends on the size and pre-loading of the spring. In addition, the pressure exerted on the fluid is not a constant. The spring- loaded accumulator typically delivers a relatively small volume of oil at low pressures. Thus, they tend to be heavy and large for high- pressure, large – volume systems. This type of accumulator should not be used for applications requiring

high cycle rates because the spring will fatigue and lose its elasticity. The result is an inoperative accumulator.



<u>Fig 5.25 Spring - Type Accumulator</u>

4. **<u>Gas Loaded Accumulator</u> :** Two main categories:



<u>Fig 5.26 Non Separator - Type Gas Accumulator</u>

a. **<u>Non separator- Type Accumulator</u>**: The non separator type of accumulator (fig 5.26)consists of a fully enclosed shell containing an oil port on the bottom and a gas charging valve on the top. The gas is confined in the top and the oil at the bottom of the shell. There is no physical separator between the gas and oil and thus the gas pushes directly on oil. The main advantage of this type is its ability to handle large volume of oil. The main disadvantage is absorption of gas in the oil due to the lack of a separator. Absorption of gas in the oil also makes the oil compressible, resulting in spongy operation of the hydraulic actuators. This type must be installed vertically to keep the gas confined at the top of the shell.

b. **Separator – Type Accumulator** : The commonly accepted design of gas loaded accumulators is the separator type. In this type there is a physical barrier between the gas and the oil. The three major type of separator accumulator are

i) **Piston type**: The piston type of accumulator consists of a cylinder containing a freely floating ~~piston~~ with proper seals. The piston serves as a barrier between the gas and oil.(fig5.27). The main disadvantage of the piston types of accumulator are that they are expensive to manufacture and have practical size limitation. The principal advantage of the piston accumulator is its ability to handle very high or low temperature system fluids through the utilization to compatible O- ring seals.

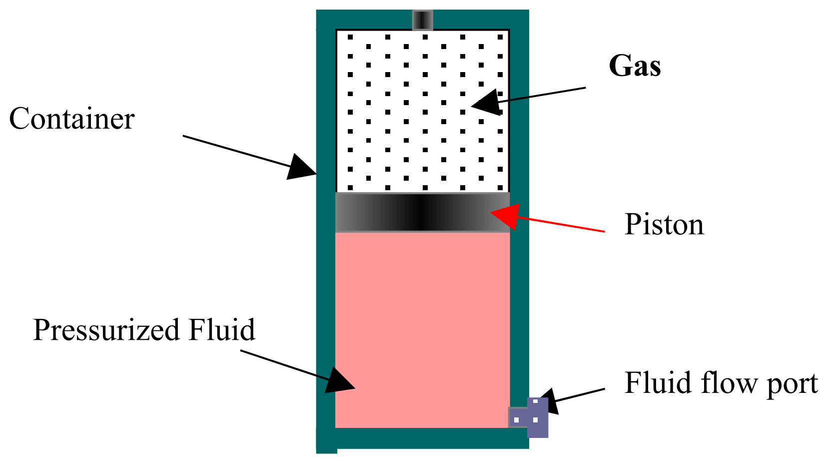


Fig 5.27 Separator - Type Piston Accumulator

ii) **Diaphragm Accumulator**: The diaphragm type accumulator consists of a diaphragm, secured in the shell, which serves as an elastic barrier between the oil and gas(fig5.28). A shutoff button, which is secured at the base of the diaphragm, covers the inlet of the line connection when the diaphragm is fully stretched. The primary advantage of this type of accumulator is its small weight-to – volume ratio, which makes it suitable almost exclusively for mobile applications. The restriction is on the deflection of the diaphragm

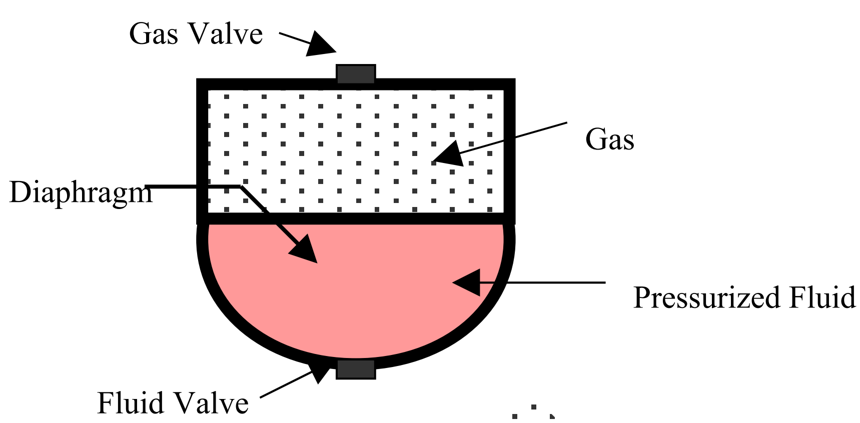
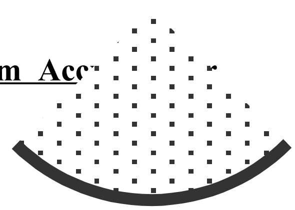


Fig 5.28 Diaphragm Acc



iii) **Bladder type Accumulator**: A bladder type- accumulator contains an elastic barrier(bladder) between the oil and gas(fig5.29). The bladder is fitted in the accumulator by means of a vulcanized gas- valve element and can be installed or removed through the shell opening at the poppet valve. The poppet valve closes the inlet when the accumulator bladder is fully expanded. This prevents the bladder from being pressed into the opening. The greatest advantage of this type of accumulator is the positive sealing between the gas and oil chambers. Most widely used type of accumulator.

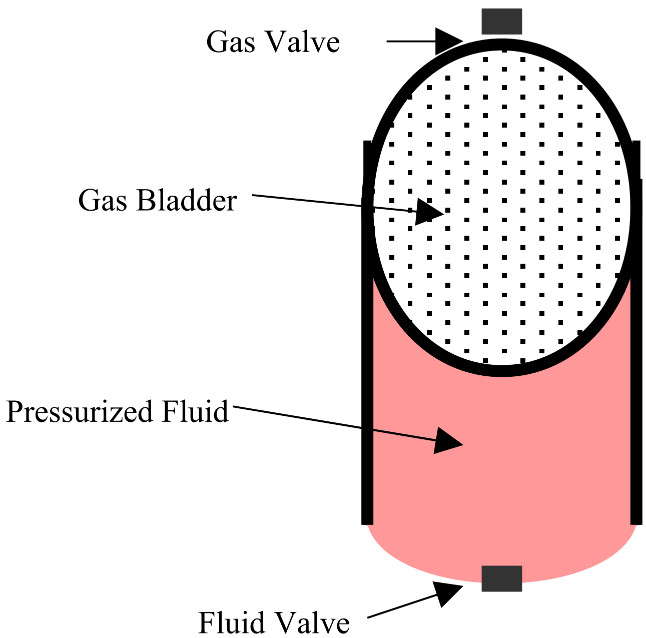


Fig 5.29 Bladder type Accumulator

Accumulator Circuits :

1. **Accumulator as a auxiliary power source** :

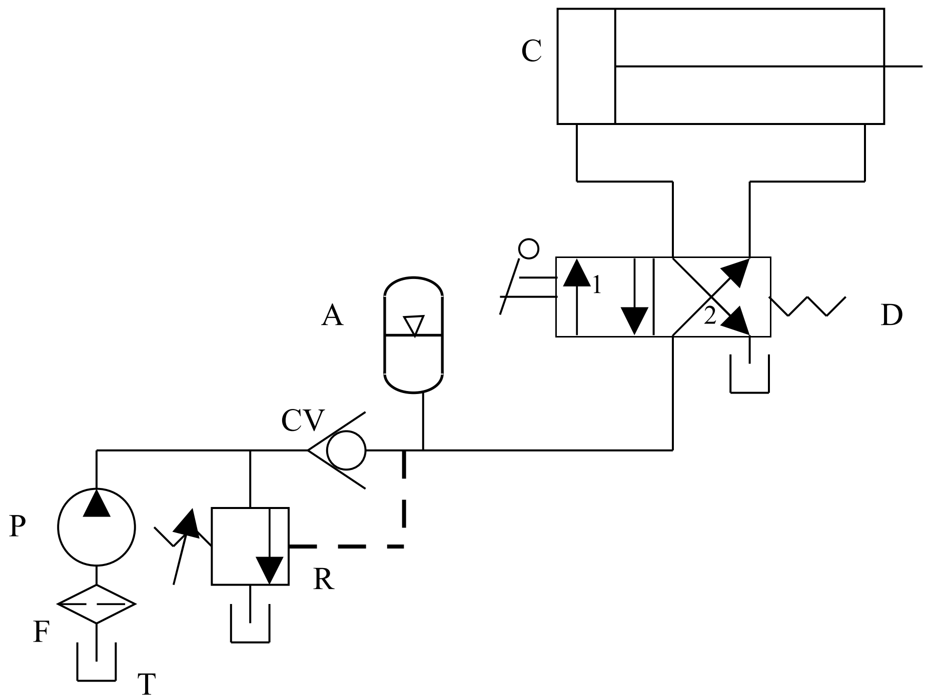


Fig 5.30. **Accumulator as an auxiliary power source.**

C = Double acting cylinder

A = Accumulator

P = Pump; T = Tank ; F = Filter

R = Relief Valve , CV = Check Valve

D =2-position, 4 way , Manually operated DCV

One of the most common application of accumulator is as an auxiliary power source. The purpose of the accumulator in this application is to store oil delivered by the pump during a portion of the work cycle. The accumulator then releases this stored oil on demand to complete the cycle, thereby serving as a secondary power source to assist the pump. In such a system where intermittent operations are performed, the use of an accumulator results in being able to use a smaller size pump.

Figure 5.30 shows such a application in which a four way valve is used in conjunction with an accumulator. When the four way valve is manually actuated, oil flows from the accumulator to the blank end of the cylinder. This extends the piston until it reaches the end of its stroke. While the desired operation is occurring (the cylinder is fully extended position), the accumulator is being charged by the pump. The four way is then deactivated for the retraction of the cylinder. Oil from both the pump and accumulator is used to retract the cylinder rapidly. The check valve prevents the back flow of oil from the accumulator when the pump is not working. The control signal for the relief valve is obtained after the check valve, to control the combined pressure of pump and the accumulator.

2. **Accumulator as an emergency power source**:

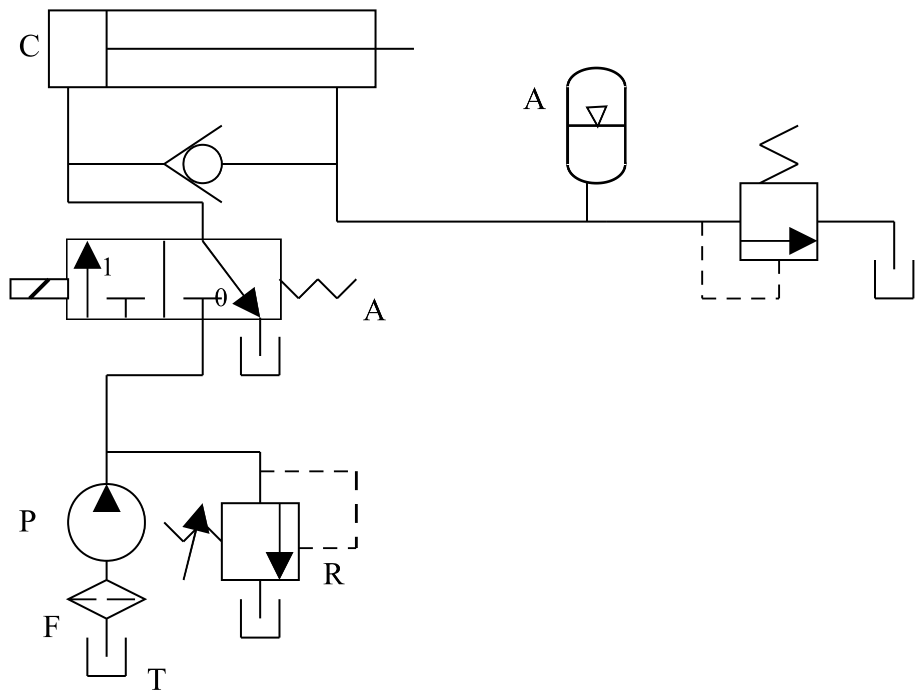


Fig 5.31. **Accumulator as an Emergency power source.**

C = Double acting cylinder

A = Accumulator

P = Pump; T = Tank ; F = Filter

R = Relief Valve , CV = Check Valve

D =2-position, 4 way , Manually operated DCV

In some hydraulic system, safety dictates that a cylinder be retracted even though the normal supply of oil pressure is lost due to a pump or electrical power failure. Such an application requires the use of an accumulator as an emergency power source.

Figure 5.31 shows such a application in which a solenoid actuated three way valve is used in conjunction with an accumulator. When the three way valve is energized, oil flows to the blank end of the cylinder and also through the check valve into the accumulator and rod end of the cylinder. The accumulator charges as the cylinder extends. If the pump fails due to an electrical failure, the solenoid will deenergize, shifting the valve to the spring -offset position. Then the oil stored under pressure is forced from the accumulator to the rod end of the cylinder. This retracts the cylinder to the starting position. In normal working, when the solenoid is deenergized, the valve shifts to the spring offset position. In this position the accumulator will retract the cylinder.

3. **Accumulator as a hydraulic shock absorber** :

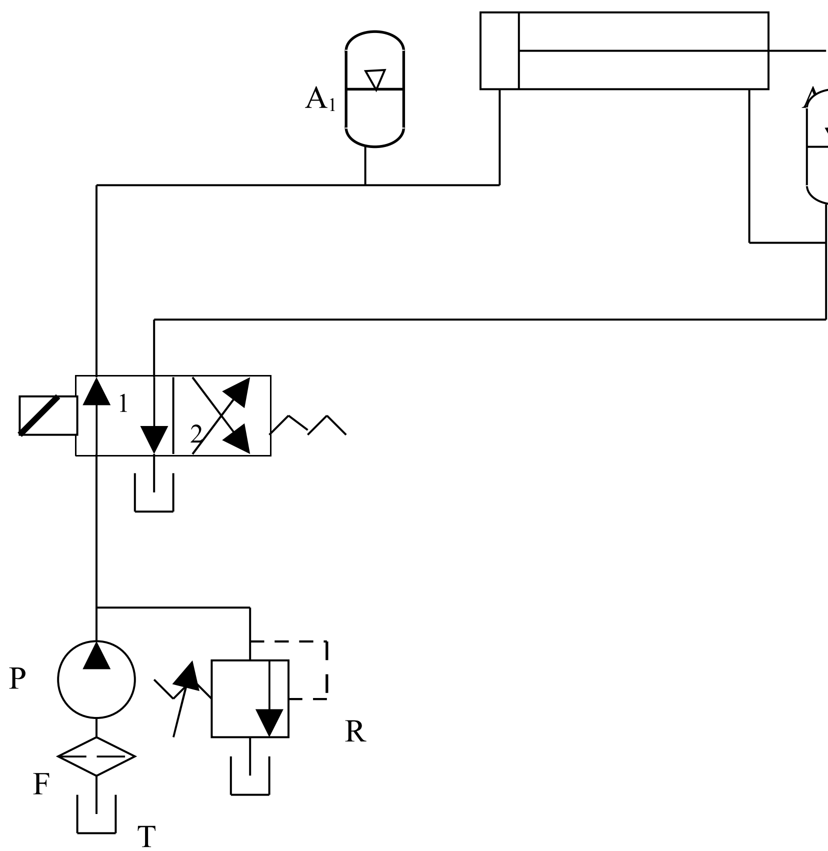


Fig 5.32. **Accumulator as an Shock Absorber**

C = Double acting cylinder

A_1, A_2 = Accumulators

P = Pump; T = Tank ; F = Filter

R = Relief Valve ,

D = 2/4 Solenoid actuated, spring return DCV

One of the most important industrial application of accumulator is the elimination or reduction of high pressure pulsation or hydraulic shock Hydraulic shock is caused by the sudden stoppage, sudden impact load, or reversal with heavy loads. Hydraulic shock load may be reduced considerably if the deceleration time of the flowing fluid mass can be reduced. The accumulator (Fig 5.32) should be installed as close to the shock source as possible. Here 2 accumulators are installed near the entry to the cylinder. The oil from the pump flow to the accumulator first and when accumulator is filled, the oil moves to the cylinder and piston starts moving.