2

Hydraulic pumps and pressure regulation

A hydraulic pump (Figure 2.1) takes oil from a tank and delivers it to the rest of the hydraulic circuit. In doing so it raises oil pressure to the required level. The operation of such a pump is illustrated in Figure 2.la. On hydraulic circuit diagrams a pump is represented by the symbol of Figure 2.1b, with the arrowhead showing the direction of flow.

Hydraulic pumps are generally driven at constant speed by a three phase AC induction motor rotating at 1500 rpm in the UK (with a 50 Hz supply) and at 1200 or 1800 rpm in the USA (with a 60 Hz supply). Often pump and motor are supplied as one combined unit. As an AC motor requires some form of starter, the complete arrangement illustrated in Figure 2.1c is needed.

There are two types of pump (for fluids) or compressor (for gases) illustrated in Figure 2.2. Typical of the first type is the centrifugal pump of Figure 2.2a. Fluid is drawn into the axis of the pump, and flung out to the periphery by centrifugal force. Flow of fluid into the load maintains pressure at the pump exit. Should the pump stop, however, there is a direct route from outlet back to inlet and the pressure rapidly decays away. Fluid leakage will also occur past the vanes, so pump delivery will vary according to outlet pressure. Devices such as that shown in Figure 2.2a are known as hydrodynamic pumps, and are primarily used to shift fluid from one location to another at relatively low pressures. Water pumps are a typical application.

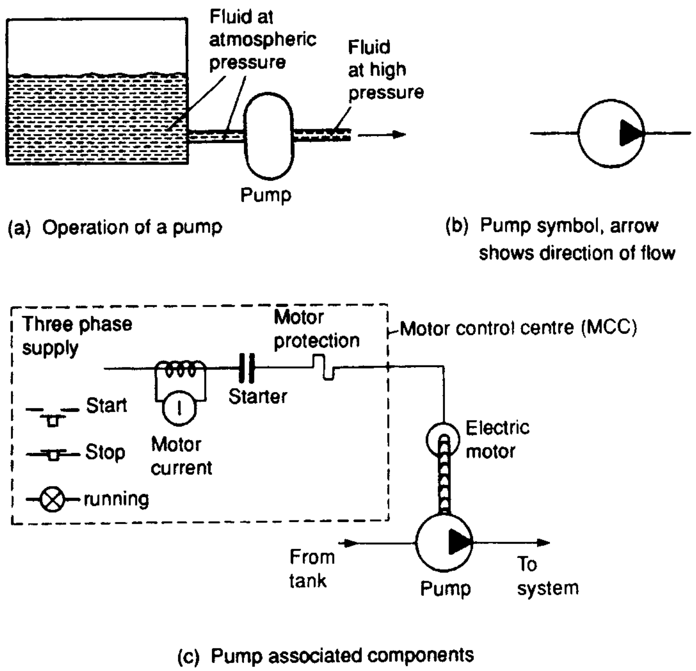


Figure 2.1 *The hydraulic pump*

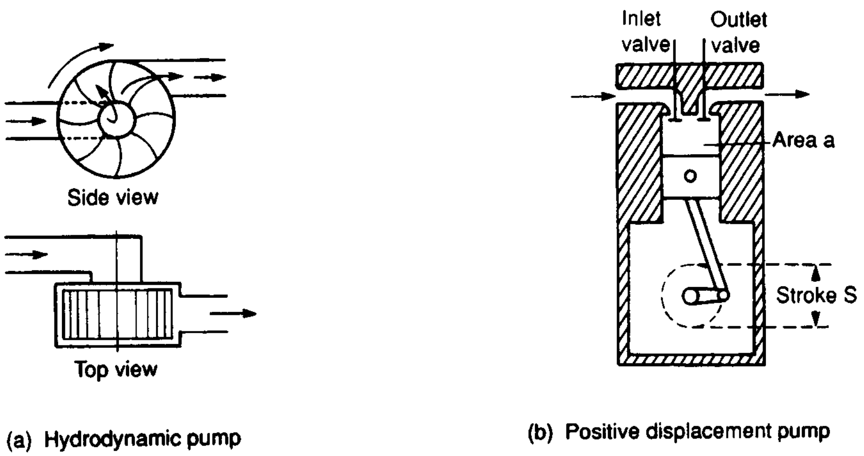


Figure 2.2 *Types of hydraulic pump*

Figure 2.2b shows a simple piston pump called a positive displacement or hydrostatic pump. As the piston is driven down, the inlet valve opens and a volume of fluid (determined by the cross section area of the piston and the length of stroke) is drawn into the cylinder. Next, the piston is driven up with the inlet valve closed and the outlet valve open, driving the same volume of fluid to the pump outlet.

Should the pump stop, one of the two valves will always be closed, so there is no route for fluid to leak back. Exit pressure is therefore maintained (assuming there are no downstream return routes).

More important, though, is the fact that the pump delivers a fixed volume of fluid from inlet to outlet each cycle regardless of pressure at the outlet port. Unlike the hydrodynamic pump described earlier, a piston pump has no inherent maximum pressure determined by pump leakage: if it drives into a dead end load with no return route (as can easily occur in an inactive hydraulic system with all valves closed) the pressure rises continuously with each pump stroke until either piping or the pump itself fails.

Hydraulic pumps are invariably hydrostatic and, consequently, require some method of controlling system pressure to avoid catastrophic pipe or pump failure. This topic is discussed further in a later section.

A hydraulic pump is specified by the flow rate it delivers (usually given in litres min^{-1} or gallons min^{-1}) and the maximum pressure the pump can withstand. These are normally called the pump capacity (or delivery rate) and the pressure rating.

Pump data sheets specify required drive speed (usually 1200, 1500 or 1800 rpm corresponding to the speed of a three phase induction motor). Pump capacity is directly related to drive speed; at a lower than specified speed, pump capacity is reduced and pump efficiency falls as fluid leakage (called slippage) increases. Pump capacity cannot, on the other hand, be expected to increase by increasing drive speed, as effects such as centrifugal forces, frictional forces and fluid cavitation will drastically reduce service life.

Like any mechanical device, pumps are not 100% efficient. The efficiency of a pump may be specified in two ways. First, volumetric efficiency relates actual volume delivered to the theoretical maximum volume. The simple piston pump of Figure 2.2b, for example, has a theoretical volume of $A \times s$ delivered per stroke, but in practice the small overlap when both inlet and outlet valves are closed will reduce the volume slightly.

Second, efficiency may be specified in terms of output hydraulic power and input mechanical (at the drive shaft) or electrical (at the motor terminals) power.

Typical efficiencies for pumps range from around 90% (for cheap gear pumps) to about 98% for high quality piston pumps. An allowance for pump efficiency needs to be made when specifying pump capacity or choosing a suitable drive motor.

The motor power required to drive a pump is determined by the pump capacity and working pressure. From expression 1.6:

$$\text{Power} = \frac{\text{work}}{\text{time}}$$

$$= \frac{\text{force} \times \text{distance}}{\text{time}}.$$

In Figure 2.3, a pump forces fluid along a pipe of area A against a pressure P, moving fluid a distance d in time T. The force is PA, which, when substituted into expression 2.1, gives:

$$\text{Power} = \frac{P \times A \times d}{T}$$

but A × d/T is flow rate, hence:

$$\text{Power} = \text{pressure} \times \text{flow rate}. \qquad (2.2)$$

Unfortunately, expression 2.2 is specified in impractical SI units (pressure in pascal, time in seconds, flow in cubic metres). We may adapt the expression to use more practical units (pressure in bar, flow rate in litres min^{-1}) with the expression:

$$\text{Power} = \frac{\text{pressure} \times \text{flow rate}}{600} \text{ Kw}. \qquad (2.3)$$

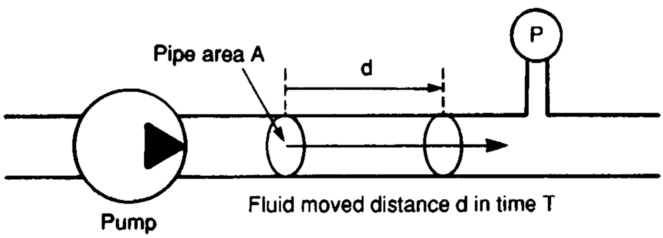


Figure 2.3 *Derivation of pump power*

For Imperial systems (pressure in psig, flow rate in gallons min^{-1}), the expression becomes:

$$\text{Power} = \frac{\text{pressure} \times \text{flow rate}}{1915} \text{ Kw.} \tag{2.4}$$

For fully Imperial systems, motor power in horsepower can be found from:

$$\text{Horsepower} = 0.75 \times \text{power in Kw.} \tag{2.5}$$

Hydraulic pumps such as that in Figure 2.1 do not require priming because fluid flows, by gravity, into the pump inlet port. Not surprisingly this is called a self-priming pump. Care must be taken with this arrangement to avoid sediment from the tank being drawn into the pump.

The pump in Figure 2.4 is above the fluid in the tank. The pump creates a negative (less than atmospheric) pressure at its inlet port causing fluid to be pushed up the inlet pipe by atmospheric pressure. This action creates a fluid lift which is, generally, incorrectly described as arising from pump suction. In reality fluid is *pushed* into the pump.

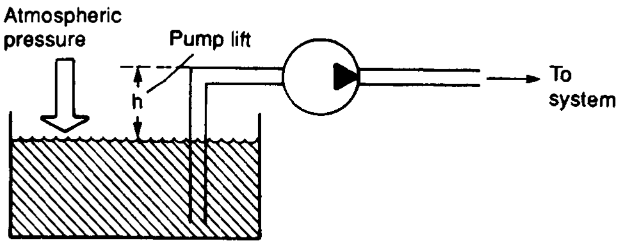


Figure 2.4 *Pump lift*

Maximum pump lift is determined by atmospheric pressure and is given by expressions 1.3 and 1.4. In theory a lift of about 8 m is feasible but, in practice, would be accompanied by undesirable side effects such as cavitation (formation and destructive collapse of bubbles from partial vaporisation of fluid). The lift should be as small as possible and around 1 m is a normal practical limit.

Fluid flow in the inlet line always takes place at negative pressure, and a relatively low flow velocity is needed to reduce these side effects. The design should aim for a flow velocity of around 1 m s^{-1}. Examination of any hydraulic system will always reveal pump inlet pipes of much larger diameters than outlet pipes.

Pressure regulation

Figure 2.5a shows the by now familiar system where a load is raised or lowered by a hydraulic cylinder. With valve V_1 open, fluid flows from the pump to the cylinder, with both pressure gauges P_1 and P_2 indicating a pressure of F/A. With valves V_1 closed and V_2 open, the load falls with fluid being returned to the tank. With the load falling, gauge P_2 will still show a pressure of F/A, but at P_1 the pump is dead-ended leading to a continual increase in pressure as the pump delivers fluid into the pipe.

Obviously some method is needed to keep P_1 at a safe level. To achieve this, pressure regulating valve V_3 has been included. This is normally closed (no connection between P and T) while the pressure is below some preset level (called the cracking pressure). Once the cracking pressure is reached valve V_3 starts to open, bleeding fluid back to the tank. As the pressure increases, valve V_3 opens more until, at a pressure called the full flow pressure, the valve is

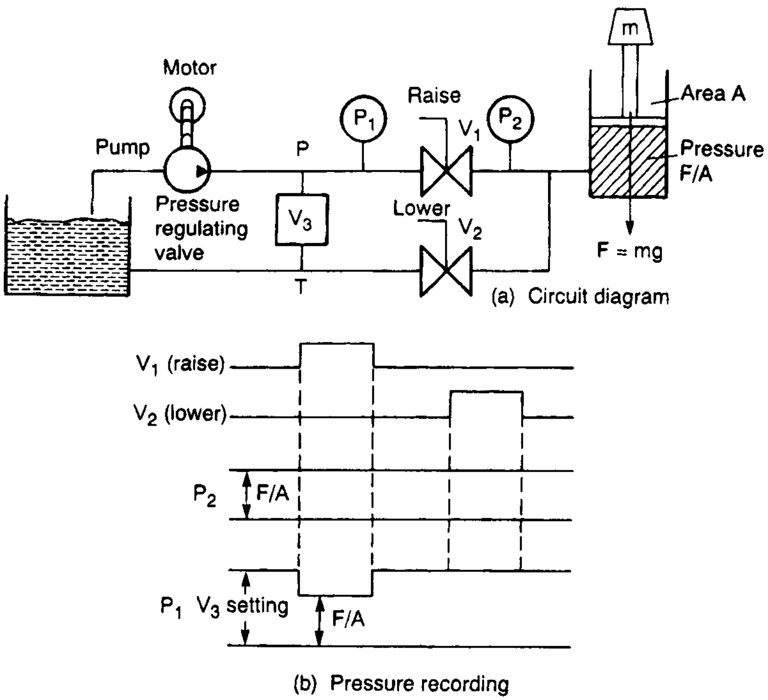


Figure 2.5 *Action of pressure regulation*

fully open. With valve V_1 closed, all fluid from the pump returns to the tank via the pressure regulating valve, and P_1 settles somewhere between the cracking and full flow pressures.

Cracking pressure of a relief valve *must* be higher than a system's working pressure, leading to a fall in system pressure as valve V_1 opens and external work is performed. Valve positions and consequent pressure readings are shown in Figure 2.5b.

The simplest form of pressure regulation valve is the ball and spring arrangement of Figure 2.6a. System pressure in the pipe exerts a force of $P \times a$ on the ball. When the force is larger than the spring compressive force the valve will crack open, bypassing fluid back to the tank. The higher the pipe pressure, the more the valve opens. Cracking pressure is set by the spring compression and in practical valves this can be adjusted to suit the application.

The difference between cracking and full flow pressure is called the pressure override. The steady (non-working) system pressure will lie somewhere within the pressure override, with the actual value determined by pipe sizes and characteristics of the pressure regulating valve itself.

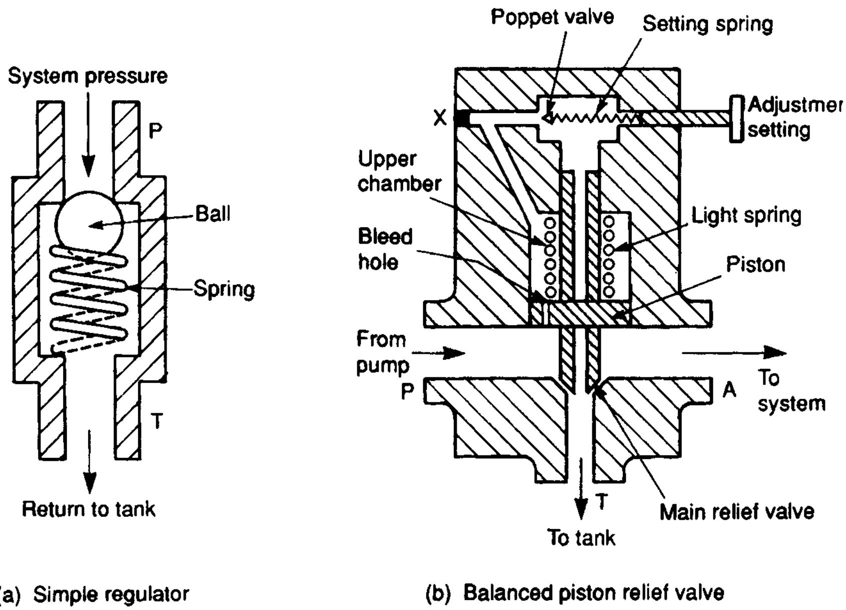


Figure 2.6 *Pressure regulation*

If the quiescent pressure is required to be precisely defined, a small pressure override is needed. This pressure override is related to spring tension in a simple relief valve. When a small, or precisely defined, override is required, a balanced piston relief valve (shown in Figure 2.6b) is used.

The piston in this valve is free moving, but is normally held in the lowered position by a light spring, blocking flow to the tank. Fluid is permitted to pass to the upper chamber through a small hole in the piston. The upper chamber is sealed by an adjustable spring-loaded poppet. In the low pressure state, there is no flow past the poppet, so pressure on both sides of the piston are equal and spring pressure keeps the valve closed.

When fluid pressure rises, the poppet cracks and a small flow of fluid passes from the upper chamber to the tank via the hole in the piston centre. This fluid is replenished by fluid flowing through the hole in the piston. With fluid flow there is now a pressure differential across the piston, which is acting only against a light spring. The whole piston lifts, releasing fluid around the valve stem until a balance condition is reached. Because of the light restoring spring a very small override is achieved.

The balanced piston relief valve can also be used as an unloading valve. Plug X is a vent connection and, if removed, fluid flows from the main line through the piston. As before, this causes the piston to rise and flow to be dumped to the tank. Controlled loading/unloading can be achieved by the use of a finite position valve connected to the vent connection.

When no useful work is being performed, *all* fluid from the pump is pressurised to a high pressure then dumped back to the tank (at atmospheric pressure) through the pressure regulating valve. This requires motor power defined earlier by expression 2.3 and 2.4, and represents a substantial waste of power. Less obviously, energy put into the fluid is converted to heat leading to a rise in fluid temperature. Surprisingly, motor power will be higher when no work is being done because cracking pressure is higher than working pressure.

This waste of energy is expensive, and can lead to the need for heat exchangers to be built into the tank to remove the excess heat. A much more economic arrangement uses loading/unloading valves, a topic discussed further in a later section.

Pump types

There are essentially three different types of positive displacement pump used in hydraulic systems.

Gear pumps

The simplest and most robust positive displacement pump, having just two moving parts, is the gear pump. Its parts are non-reciprocating, move at constant speed and experience a uniform force. Internal construction, shown in Figure 2.7, consists of just two close meshing gear wheels which rotate as shown. The direction of rotation of the gears should be carefully noted; it is the *opposite* of that intuitively expected by most people.

As the teeth come out of mesh at the centre, a partial vacuum is formed which draws fluid into the inlet chamber. Fluid is trapped between the outer teeth and the pump housing, causing a continual transfer of fluid from inlet chamber to outlet chamber where it is discharged to the system.

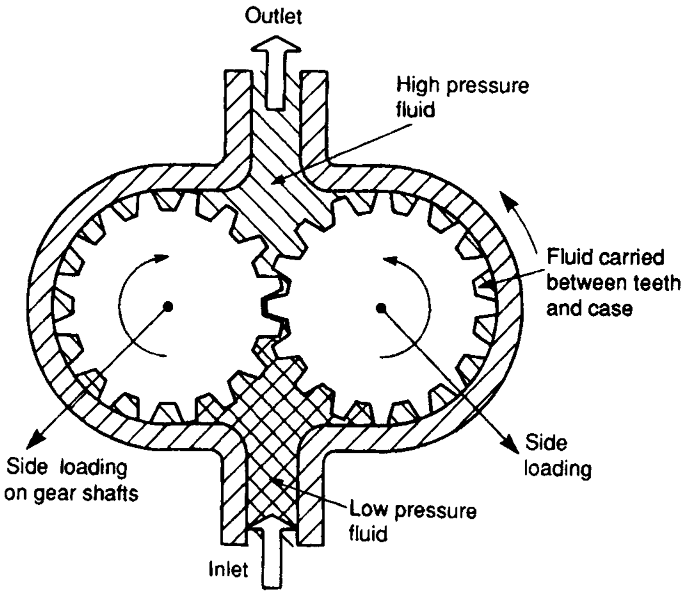


Figure 2.7 *Gear pump*

Pump displacement is determined by: volume of fluid between each pair of teeth; number of teeth; and speed of rotation. Note the pump merely delivers a fixed volume of fluid from inlet port to outlet port for each rotation; outlet port pressure is determined solely by design of the rest of the system.

Performance of any pump is limited by leakage and the ability of the pump to withstand the pressure differential between inlet and outlet ports. The gear pump obviously requires closely meshing gears, minimum clearance between teeth and housing, and also between the gear face and side plates. Often the side plates of a pump are designed as deliberately replaceable wear plates. Wear in a gear pump is primarily caused by dirt particles in the hydraulic fluid, so cleanliness and filtration are particularly important.

The pressure differential causes large side loads to be applied to the gear shafts at 45° to the centre line as shown. Typically, gear pumps are used at pressures up to about 150 bar and capacities of around 150 gpm (6751 min^{-1}). Volumetric efficiency of gear pumps at 90% is lowest of the three pump types.

There are some variations of the basic gear pump. In Figure 2.8, gears have been replaced by lobes giving a pump called, not surprisingly, a lobe pump.

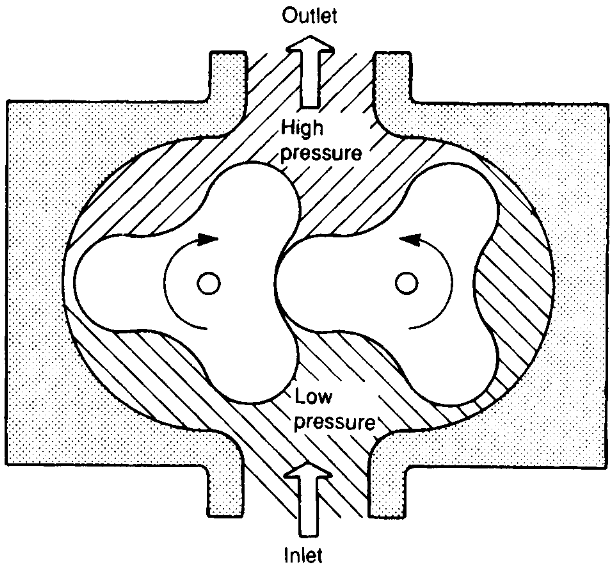
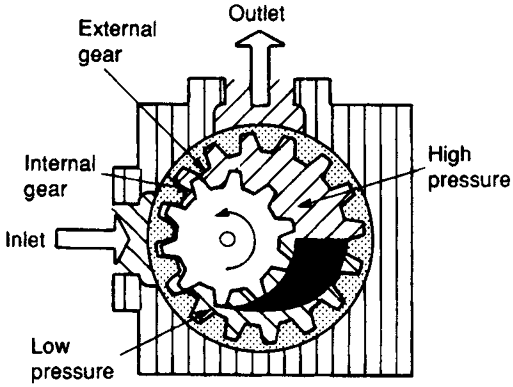
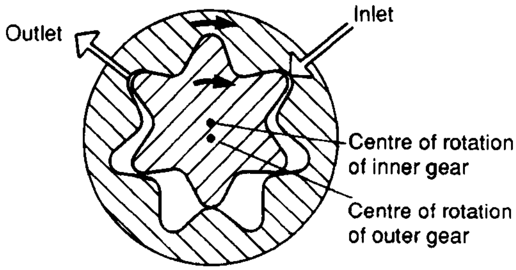


Figure 2.8 *The lobe pump*



(a) Internal gear pump



(b) Gerotor pump

Figure 2.9 *Further forms of gear pump*

Figure 2.9a is another variation called the internal gear pump, where an external driven gear wheel is connected to a smaller internal gear, with fluid separation as gears disengage being performed by a crescent-shaped moulding. Yet another variation on the theme is the gerotor pump of Figure 2.9b, where the crescent moulding is dispensed with by using an internal gear with one less tooth than the outer gear wheel. Internal gear pumps operate at lower capacities and pressures (typically 70 bar) than other pump types.

Vane pumps

The major source of leakage in a gear pump arises from the small gaps between teeth, and also between teeth and pump housing. The vane pump reduces this leakage by using spring (or hydraulic) loaded vanes slotted into a driven rotor, as illustrated in the two examples of Figure 2.10.

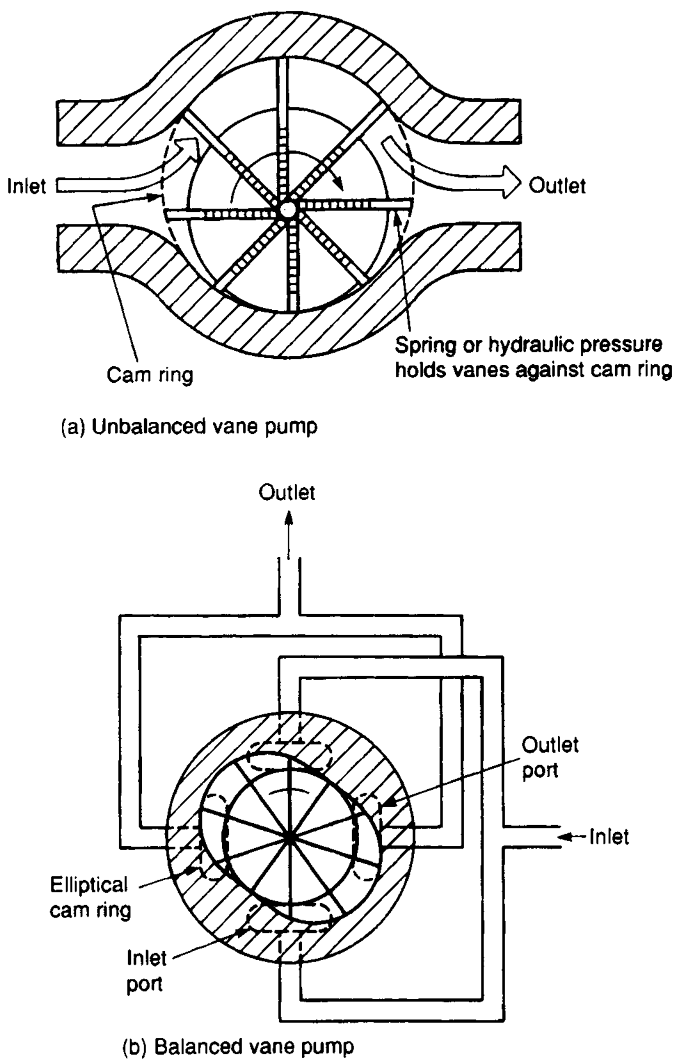


Figure 2.10 *Vane pumps*

In the pump shown in Figure 2.10a, the rotor is offset within the housing, and the vanes constrained by a cam ring as they cross inlet and outlet ports. Because the vane tips are held against the housing there is little leakage and the vanes compensate to a large degree for wear at vane tips or in the housing itself. There is still, however, leakage between rotor faces and body sides. Pump capacity is determined by vane throw, vane cross sectional area and speed of rotation.

The difference in pressure between outlet and inlet ports creates a severe load on the vanes and a large side load on the rotor shaft which can lead to bearing failure. The pump in Figure 2.10a is consequently known as an unbalanced vane pump. Figure 2.10b shows a balanced vane pump. This features an elliptical cam ring together with two inlet and two outlet ports. Pressure loading still occurs in the vanes but the two identical pump halves create equal but opposite forces on the rotor, leading to zero net force in the shaft and bearings. Balanced vane pumps have much improved service lives over simpler unbalanced vane pumps.

Capacity and pressure ratings of a vane pump are generally lower than gear pumps, but reduced leakage gives an improved volumetric efficiency of around 95%.

In an ideal world, the capacity of a pump should be matched exactly to load requirements. Expression 2.2 showed that input power is proportional to system pressure and volumetric flow rate. A pump with too large a capacity wastes energy (leading to a rise in fluid temperature) as excess fluid passes through the pressure relief valve.

Pumps are generally sold with certain fixed capacities and the user has to choose the next largest size. Figure 2.11 shows a vane pump with adjustable capacity, set by the positional relationship between rotor and inner casing, with the inner casing position set by an external screw.

Piston pumps

A piston pump is superficially similar to a motor car engine, and a simple single cylinder arrangement was shown earlier in Figure 2.2b. Such a simple pump, however, delivering a single pulse of fluid per revolution, generates unacceptably large pressure pulses into the system. Practical piston pumps therefore employ multiple cylinders

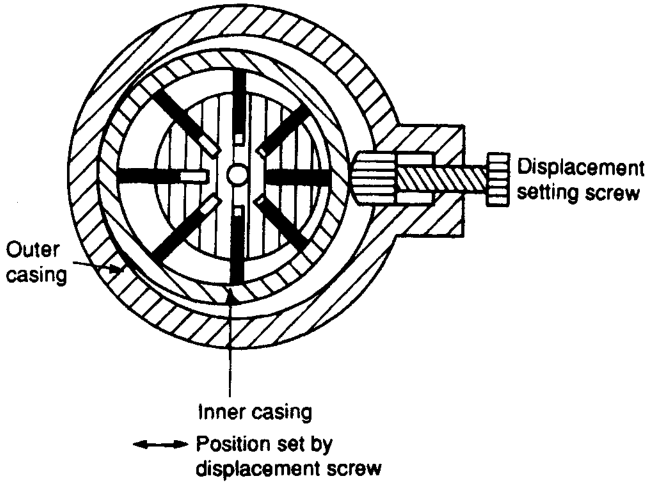


Figure 2.11 *Variable displacement vane pump*

and pistons to smooth out fluid delivery, and much ingenuity goes into designing multicylinder pumps which are surprisingly compact.

Figure 2.12 shows one form of radial piston pump. The pump consists of several hollow pistons inside a stationary cylinder block. Each piston has spring-loaded inlet and outlet valves. As the inner cam rotates, fluid is transferred relatively smoothly from inlet port to the outlet port.

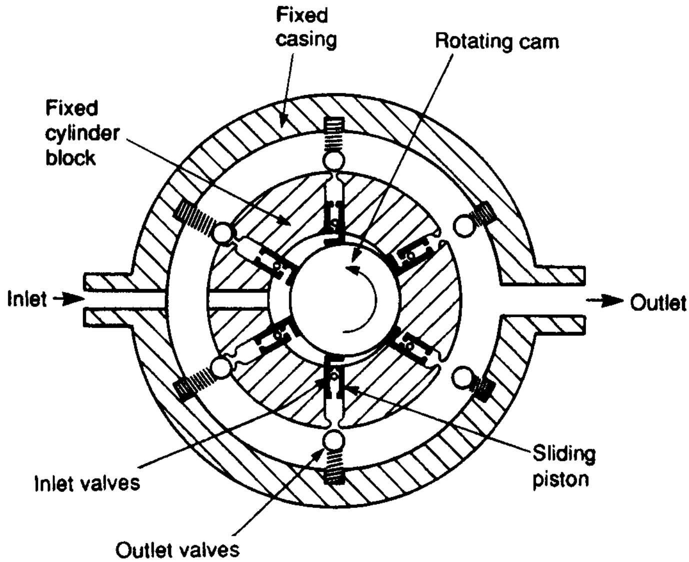


Figure 2.12 *Radial piston pump*

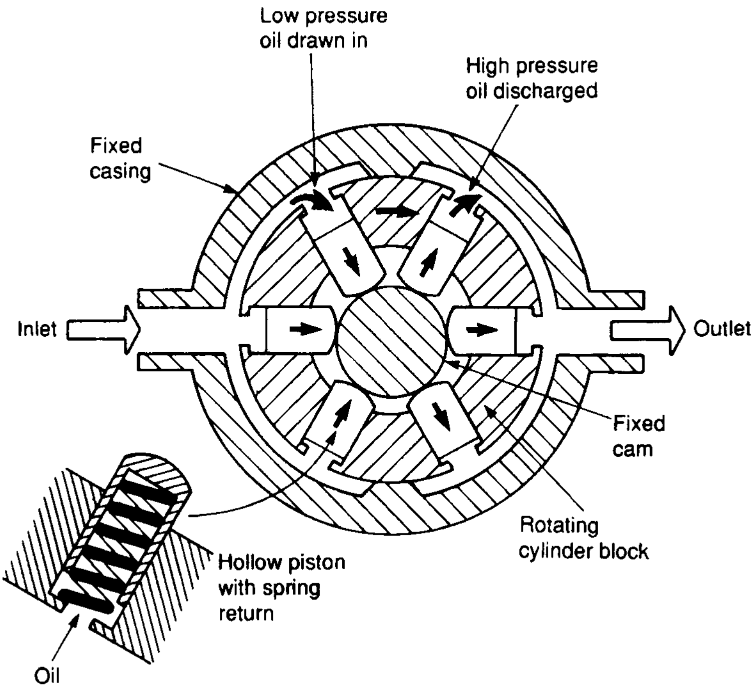


Figure 2.13 *Piston pump with stationary cam and rotating block*

The pump of Figure 2.13 uses the same principle, but employs a stationary cam and a rotating cylinder block. This arrangement does not require multiple inlet and outlet valves and is consequently simpler, more reliable, and cheaper. Not surprisingly most radial piston pumps have this construction.

An alternative form of piston pump is the axial design of Figure 2.14, where multiple pistons are arranged in a rotating cylinder. The pistons are stroked by a fixed angled plate called the swash plate. Each piston can be kept in contact with the swash plate by springs or by a rotating shoe plate linked to the swash plate.

Pump capacity is controlled by altering the angle of the swash plate; the larger the angle, the greater the capacity. With the swash plate vertical capacity is zero, and flow can even be reversed. Swash plate angle (and hence pump capacity) can easily be controlled remotely with the addition of a separate hydraulic cylinder.

An alternative form of axial piston pump is the bent axis pump of Figure 2.15. Stroking of the pistons is achieved because of the angle between the drive shaft and the rotating cylinder block. Pump capacity can be adjusted by altering the drive shaft angle.

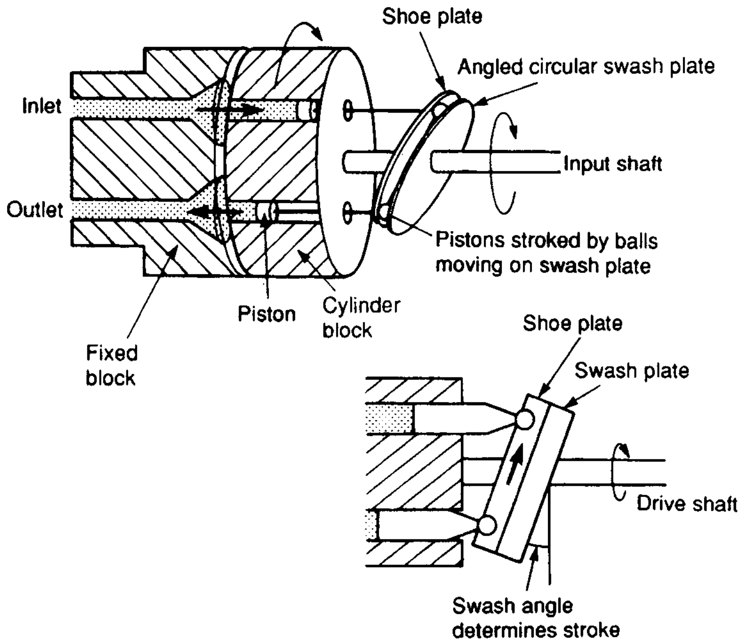


Figure 2.14 *Axial pump with swash plate*

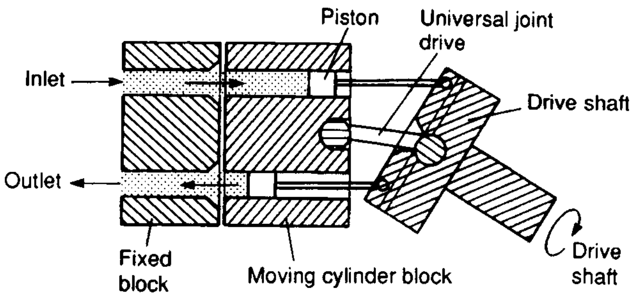


Figure 2.15 *Bent axis pump*

Piston pumps have very high volumetric efficiency (over 98%) and can be used at the highest hydraulic pressures. Being more complex than vane and gear pumps, they are correspondingly more expensive. Table 2.1 gives a comparison of the various types of pump.

Table 2.1 Comparison of hydraulic pump types

Type	*Maximum pressure (bar)*	*Maximum flow (l/min)*	*Variable displacement*	*Positive displacement*
Centrifugal	20	3000	No	No
Gear	175	300	No	Yes
Vane	175	500	Yes	Yes
Axial piston (port-plate)	300	500	Yes	Yes
Axial piston (valved)	700	650	Yes	Yes
In-line piston	1000	100	Yes	Yes

Specialist pumps are available for pressures up to about 7000 bar at low flows. The delivery from centrifugal and gear pumps can be made variable by changing the speed of the pump motor with a variable frequency (VF) drive.

Combination pumps

Many hydraulic applications are similar to Figure 2.16, where a workpiece is held in place by a hydraulic ram. There are essentially two distinct requirements for this operation. As the cylinder extends or retracts a large volume of fluid is required at a low pressure (sufficient just to overcome friction). As the workpiece is gripped, the requirement changes to a high pressure but minimal fluid volume.

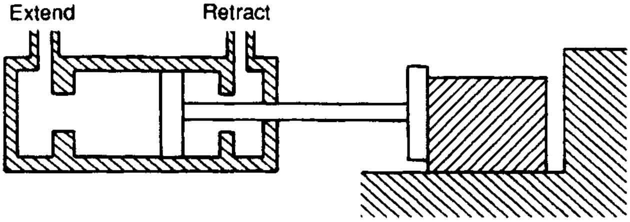


Figure 2.16 *A clamping cylinder. A large flow, but low pressure, is needed during extension and retraction, but zero flow and high pressure are needed during clamping*

This type of operation is usually performed with two separate pumps driven by a common electric motor as shown in Figure 2.17. Pump P_1 is a high pressure low volume pump, while pump P_2 is a high volume low pressure pump. Associated with these are two relief valves RV_1 and RV_2 and a one-way check (or non-return)

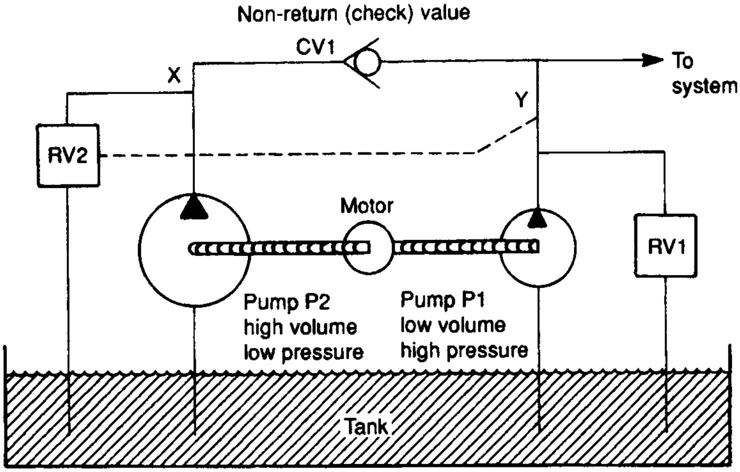


Figure 2.17 *Combination pump*

valve which allows flow from left to right, but blocks flow in the reverse direction.

A normal (high pressure) relief valve is used at position RV_1 but relief valve RV_2 is operated not by the pressure at point X, but remotely by the pressure at point Y. This could be achieved with the balanced piston valve of Figure 2.6. In low pressure mode both relief valves are closed and both pumps P_1 and P_2 deliver fluid to the load, the majority coming from pump P_2 because of its higher capacity.

When the workpiece is gripped, the pressure at Y rises, and relief valve RV_2 opens causing all the fluid from pump P_2 to return straight to the tank and the pressure at X to fall to a low value. Check valve CV_1 stops fluid from pump P_1 passing back to the tank via relief valve RV_2, consequently pressure at Y rises to the level set by relief valve RV_1.

This arrangement saves energy as the large volume of fluid from pump P_2 is returned to the tank at a very low pressure, and only a small volume of fluid from pump P_1 is returned at a high pressure. Pump assemblies similar to that shown in Figure 2.17 are called combination pumps and are manufactured as complete units with motor, pumps, relief and check valves prefitted.

Loading valves

Expression 2.2 shows that allowing excess fluid from a pump to return to the tank by a pressure relief valve is wasteful of energy and can lead to a rapid rise in temperature of the fluid as the wasted energy is converted to heat. It is normally undesirable to start and stop the pump to match load requirements, as this causes shock loads to pump, motor and couplings.

In Figure 2.18, valve V_1 is a normal pressure relief valve regulating pressure and returning excess fluid to the tank as described in earlier sections. The additional valve V_2 is opened or closed by an external electrical or hydraulic signal. With valve V_2 open, all the pump output flow is returned to the tank at low pressure with minimal energy cost.

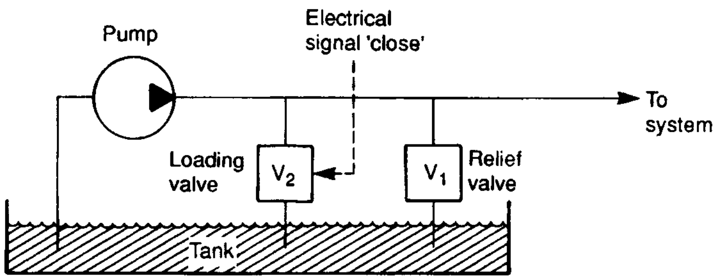


Figure 2.18 *Loading valve*

When fluid is required in the system the control signal closes valve V_2, pressure rises to the setting of valve V_1, and the system performs as normal. Valve V_2 is called a pump loading or a pump unloading valve according to the interpretation of the control signal sense.

Filters

Dirt in a hydraulic system causes sticking valves, failure of seals and premature wear. Even particles of dirt as small as 20 μ can cause damage, (1 micron is one millionth of a metre; the naked eye is just able to resolve 40 μ). Filters are used to prevent dirt entering the vulnerable parts of the system, and are generally specified in microns or meshes per linear inch (sieve number).

Inlet lines are usually fitted with strainers inside the tank, but these are coarse wire mesh elements only suitable for removing relatively large metal particles and similar contaminants Separate filters are needed to remove finer particles and can be installed in three places as shown in Figures 2.19a to c.

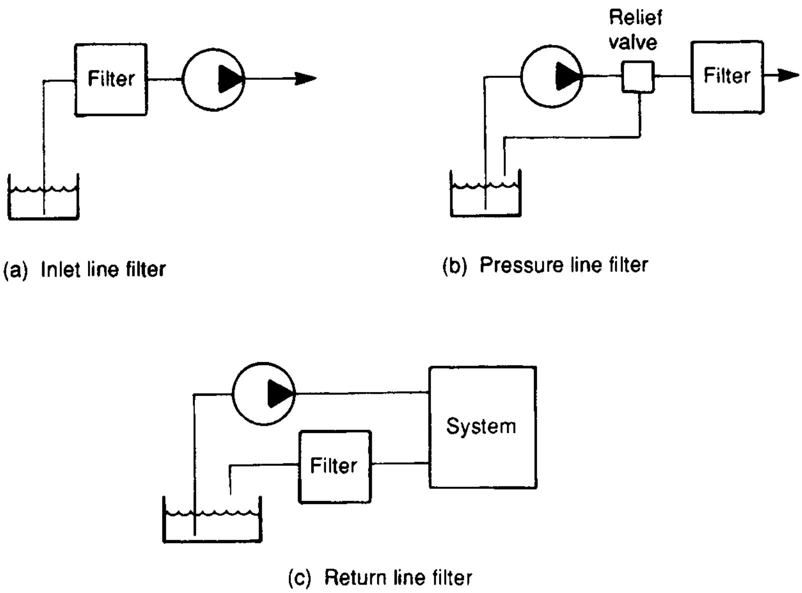


Figure 2.19 *Filter positions*

Inlet line filters protect the pump, but must be designed to give a low pressure drop or the pump will not be able to raise fluid from the tank. Low pressure drop implies a coarse filter or a large physical size.

Pressure line filters placed after the pump protect valves and actuators and can be finer and smaller. They must, however, be able to withstand full system operating pressure. Most systems use pressure line filtering.

Return line filters may have a relatively high pressure drop and can, consequently, be very fine. They serve to protect pumps by limiting size of particles returned to the tank. These filters only have to withstand a low pressure. Filters can also be classified as full or proportional flow. In Figure 2.20a, all flow passes through the filter. This is obviously efficient in terms of filtration, but incurs a large pressure drop. This pressure drop increases as the filter becomes polluted, so a full flow filter usually incorporates a relief valve which cracks when the filter becomes unacceptably blocked. This is purely a safety feature, though, and the filter should, of course, have been changed before this state was reached as dirty unfiltered fluid would be passing round the system.

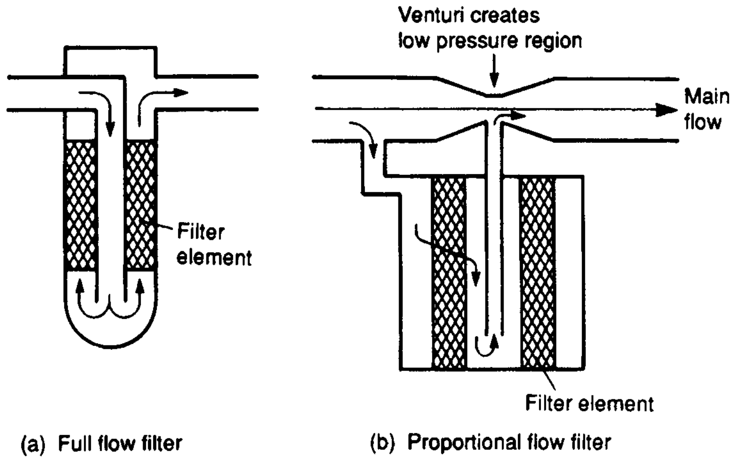


Figure 2.20 *Filter types*

In Figure 2.20b, the main flow passes through a venturi, creating a localised low pressure area. The pressure differential across the filter element draws a proportion of the fluid through the filter. This design is accordingly known as a proportional flow filter, as only a proportion of the main flow is filtered. It is characterized by a low pressure drop, and does not need the protection of a pressure relief valve.

Pressure drop across the filter element is an accurate indication of its cleanliness, and many filters incorporate a differential pressure meter calibrated with a green (clear), amber (warning), red (change overdue) indicator. Such types are called indicating filters.

Filtration material used in a filler may be mechanical or absorbent. Mechanical filters are relatively coarse, and utilise fine wire mesh or a disc/screen arrangement as shown in the edge type filter of Figure 2.21. Absorbent filters are based on porous materials such as paper, cotton or cellulose. Filtration size in an absorbent filter can be very small as filtration is done by pores in the material. Mechanical filters can usually be removed, cleaned and re-fitted, whereas absorbent filters are usually replaceable items.

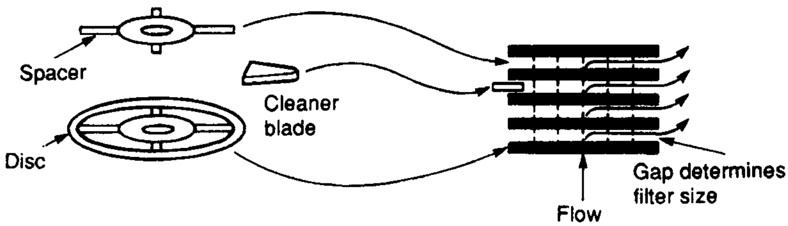


Figure 2.21 *Edge type filter*

In many systems where the main use is the application of pressure the actual draw from the tank is very small reducing the effectiveness of pressure and return line filters. Here a separate circulating pump may be used as shown on Figure 2.22 to filter and cool the oil. The running of this pump is normally a pre-condition for starting the main pumps. The circulation pump should be sized to handle the complete tank volume every 10 to 15 minutes.

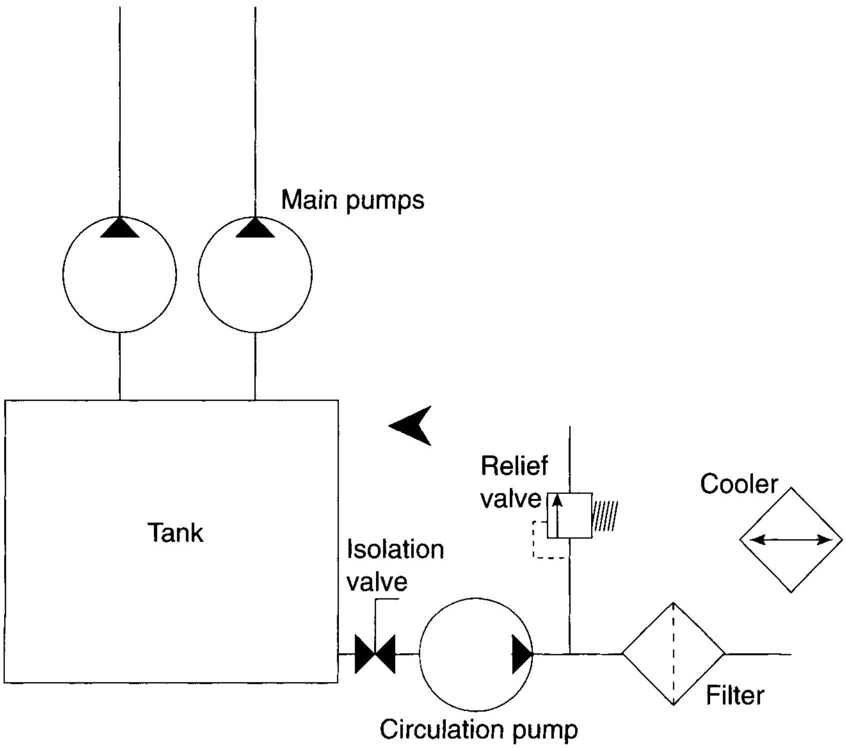


Figure 2.22 *A circulation pump used to filter and clean the fluid when the draw from the main pumps is small*

Note the pressure relief valve – this is included to provide a route back to tank if the filter or cooler is totally blocked. In a real life system additional hand isolation and non return valves would be fitted to permit changing the filter or cooler with the system running. Limit switches and pressure switches would also be included to signal to the control system that the hand isolation valves are open and the filter is clean.