3. HYDRAULIC SYSTEMS

3.1. Introduction

Hydraulic fluids are commonly used aboard as power transmission drives (actuating fluids) instead of electric power, pneumatic or, occasionally, steam.

Usage of hydraulic fluids is most common when transmission of power to slow speed and large force (or torque) driven equipment is required; hence, hydraulic actuating fluids working at high pressures are commonly utilized.

Major components of hydraulic systems used aboard are as follows:

a) Hydraulic pumps with a sump or reservoir;

b) Hydraulic piping, which may be either flexible or rigid type;

c) Discharge and compensation tanks, and filtering elements (at pump discharge line) or strainers (at pump suction line);

d) Hydraulic actuator (either rotative motor or axial cylinder - ram);

e) Heat exchangers (specially, in case of constant-flow hydraulic system);

f) Other accessories (valves, manometers, accumulators, etc.);

g) Command and control console.

Hydraulic systems, if well designed, allow variable speed operation and easy inversion of the driven equipment.

Hydraulic fluids must have adequate lubrication characteristics, should have low abrasive characteristics, and good thermal and chemical stability characteristics.

Basically, there are two different types of load control systems:

- Constant-flow hydraulic system;

- Constant-pressure hydraulic system.

As illustrated in Figure 3.1, the major components of a constant-flow hydraulic system are the fixed-delivery pump, unloading valve, and control valve. When the pump is started, fluid is sometimes delivered to an accumulator, which is usually of the pneumatic type, until the accumulator pressure equals that of the unload valve setting. If there is no demand on the system, the unloading valve opens to bypass the pump discharge back to the tank, and the pump runs continuously. At the same time, the check valve holds the fluid pressure in the system so that it is immediately available upon demand.

When a throttle or control valve to an attached hydraulic actuator is opened, the system immediately provides fluid to this load and the discharge causes a drop in system pressure. In turn, the unloading valve closes and the pump discharge is returned to the active system. Upon a decrease in the load demand, the system pressure again rises to actuate the unloading valve, and the load cycle is complete. The sump tank is replenished by returns from the exhaust lines of the attached loads.

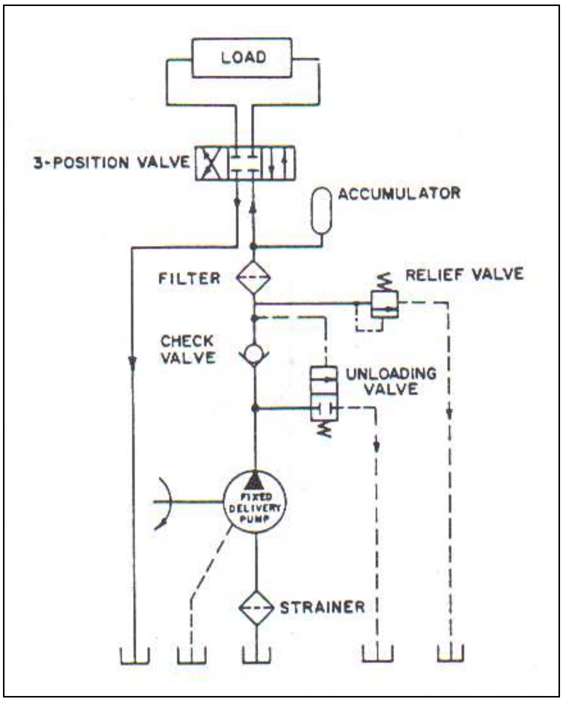


Fig. 3.1 – Single-wire diagram of a constant-flow hydraulic system.

Constant-pressure hydraulic systems, as shown in Figure 3.2, generally use one or more variable-delivery pressure-compensated pumps that supply hydraulic fluid at a substantially constant pressure to either a system of multiple loads or to a single load, such as a hydraulic elevator hoist. The constant-pressure hydraulic system pump takes suction from a sump tank and discharges directly into the main supply piping. When the pumping capacity exceeds the load requirements, the system pressure increases to a predetermined value, at which point the pressure compensator acts to take the pump off stroke, thus stopping or reducing the flow of hydraulic fluid. A relief valve is provided to protect against overpressurization in the event that the pressure compensator fails to properly reduce the stroke of the pump. When the system demands cause the system pressure to drop to a preset value, the pressure compensator acts to put the pump on stroke, thus restoring the flow of the hydraulic fluid to recharge the accumulator, if installed, and to maintain the system operating pressure. The fluid flow to the individual loads may be controlled by a variety of types of valves, in addition to the three-position valve shown for simplicity in Figure 3.2.

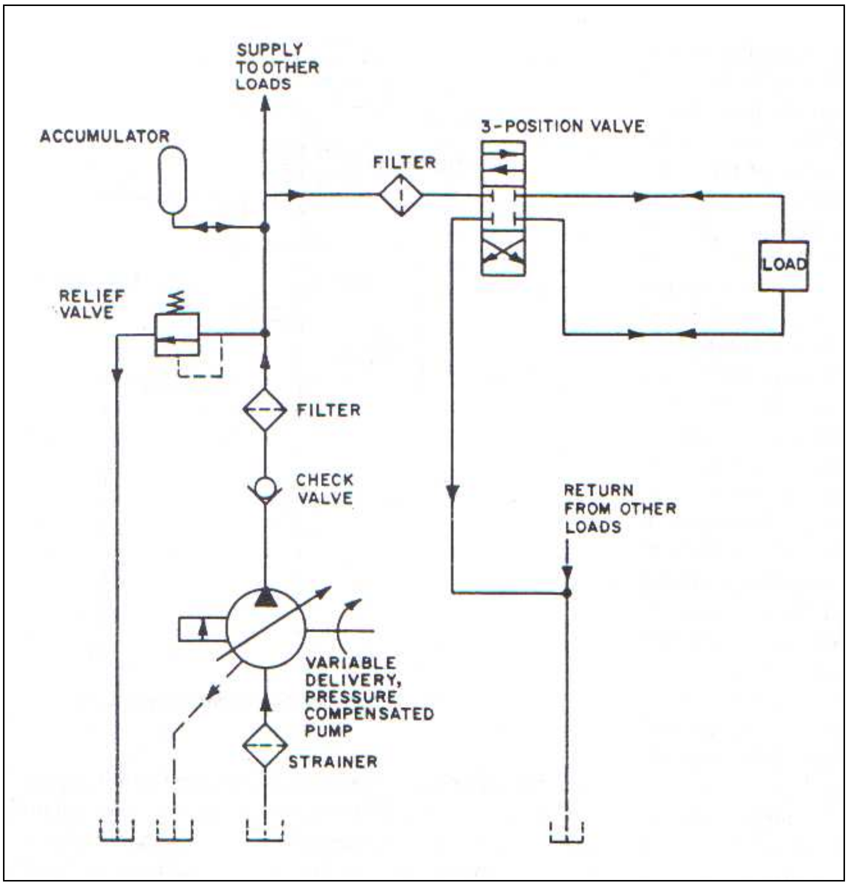


Fig. 3.2 – Single-wire diagram of a constant-pressure hydraulic system.

The following set of items: pressurizing device (pump), discharge tank, heat exchanger and accessories, is usually designated as Hydraulic Power Unit (HPU).

As mentioned before, pumps installed aboard are usually powered by electrical motors or, directly from main propulsion system by means of a Power Take-Off (PTO).

Hydraulic systems onboard, specially, those of the constant-flow type need to have an heat exchanger do dissipate heat from hydraulic fluid, which is generated by means friction losses. In some cases where the HPU is installed outside the ship, there is no specific unit such as an heat exchanger, and therefore heat exchange with outside is conducted at the walls of the discharge tank, which must be fitted with fins or be of the corrugated type.

Hydraulic systems use only displacement pumps, which might be of different types such as: piston displacement pump, rotary vane displacement pump, or screw displacement pump. The pumps commonly used aboard are of the axial piston or radial piston (Hele-Shaw). Motors having high rotational speed and low torque are of the axial piston type.

Hydraulic motors are very similar to hydraulic pumps, where, essentially some differences may be found at the compression and aspiration valves. Hydraulic motors with low speed and high torque are distinct from those electric having the similar torque characteristics, namely, there is no need to install a reduction gearbox and fine tuning of speed regulation is far more accurate.

Control equipment of hydraulic systems might be mechanical, hydraulic, or electrical. Servo-valves are extremely important units of the control equipment, since these allow remote command and control of all the hydraulic circuits.

Ship's equipments which are more often hydraulically powered are the steering gear, controllable pitch propellers, deck machinery equipment (such as hatch covers onboard container vessels, bow doors onboard RO-RO vessels, actuators of watertight doors, capstans and windlasses – see Figure 3.3), cargo movement equipment (such as elevator hoists, cranes and cargo booms), and propulsion auxiliary devices (such as bow and stern thrusters).

There are some rare occasions where hydraulic motors have been used for main propulsion systems. This hydraulic powering solution presents several features similar to electric propulsion, since propeller shaft lines are eliminated, and position of the propulsive HPU aboard is less restrained.

As illustrated in Figure 3.4, hydraulic systems are also used as command and control other systems, or other type of equipments.

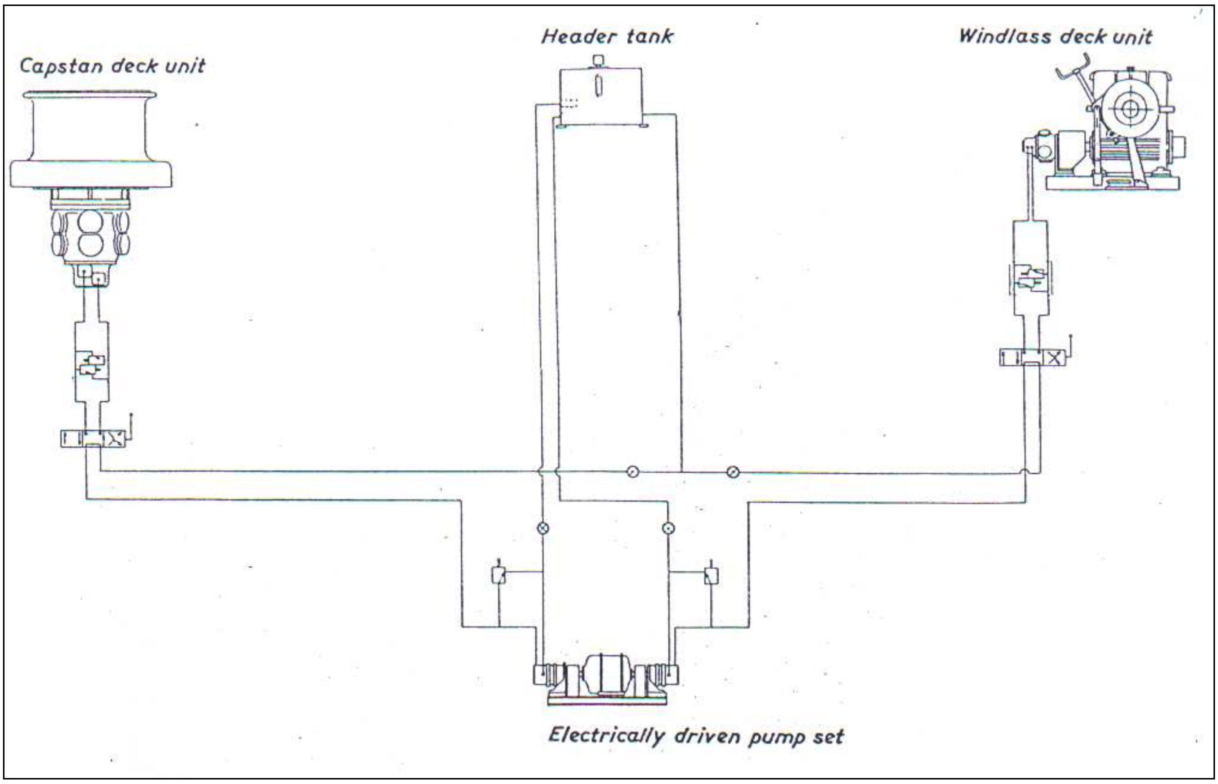


Fig. 3.3 – Piping arrangement for hydraulic windlass and capstan installation.

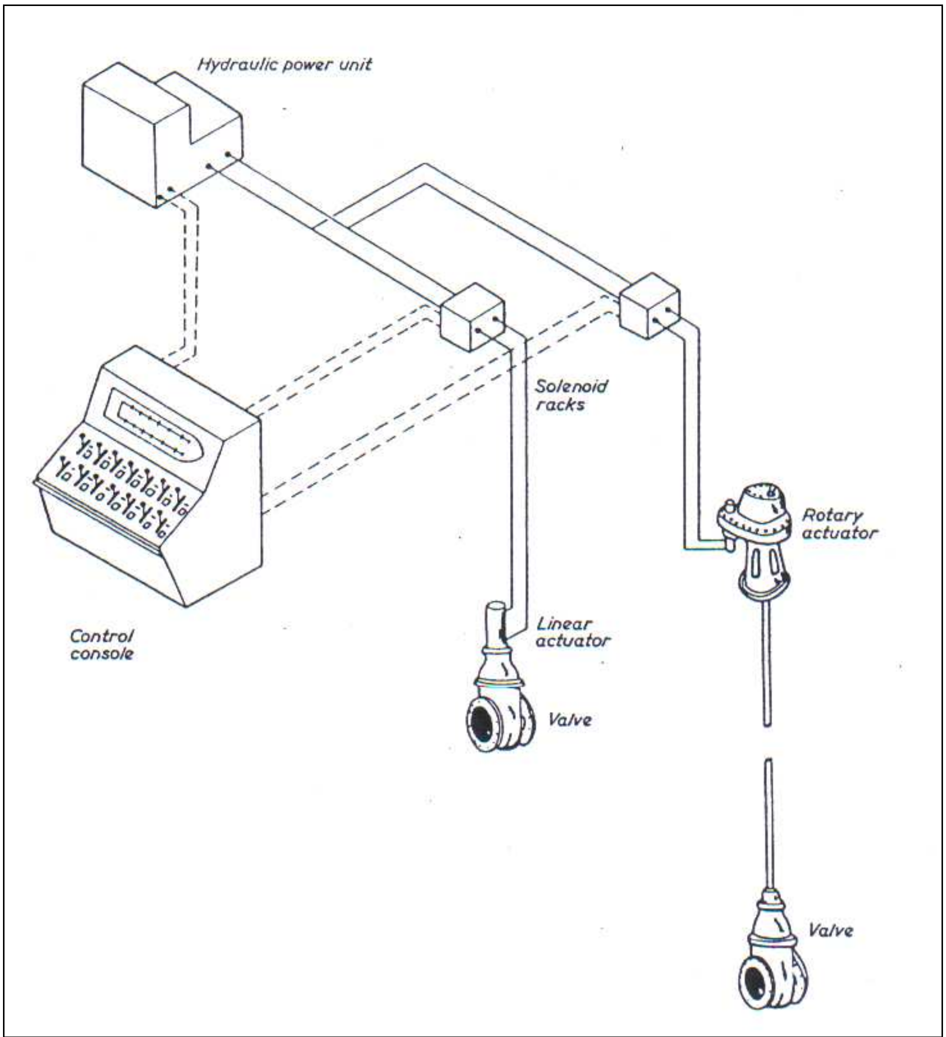


Fig. 3.4 – Typical cargo valve actuating system – schematic arrangement.

3.2. Pumps

a) Pump Types

There are three main classes of pump in marine use (not hydraulic specific): displacement, axial flow and centrifugal. A number of different arrangements are possible for displacement and centrifugal pumps to meet particular system characteristics.

a.1) Displacement

The displacement pumping action is achieved by the reduction or increase in volume of a space causing the liquid (or gas) to be physically moved. The method employed is either a piston in a cylinder using a reciprocating motion, or a rotating unit using vanes, gears or screws.

A reciprocating displacement pump is shown diagrammatically in Figure 3.5, to demonstrate the operating principle. The pump is double-acting, that is liquid is admitted to either side of the piston where it is alternately drawn in and discharged. As the piston moves upwards, suction takes place below the piston and liquid is drawn in, the valve arrangement ensuring that the discharge valve cannot open on the suction stroke. Above the piston, liquid is discharged and the suction valve remains closed. As the piston travels down, the operations of suction and discharge occur now on opposite sides.

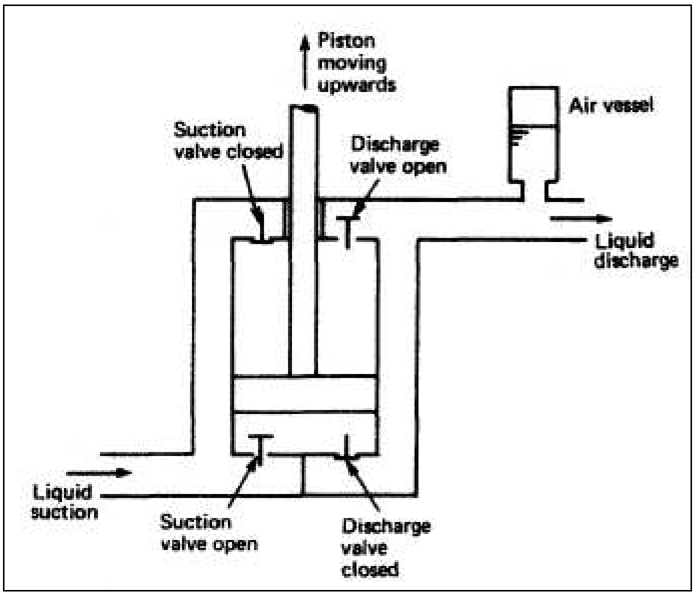


Fig. 3.5 – Diagrammatic reciprocating displacement pump.

An air vessel (accumulator) is usually fitted in the discharge pipework to dampen out the pressure variations during discharge. As the discharge pressure rises the air is compressed in the vessel, and as the pressure falls the air expands. The peak pressure energy is thus 'stored' in the air and returned to the system when the pressure falls. Air vessels are not fitted on reciprocating boiler feed pumps since they may introduce air into the de-aerated feedwater.

A relief valve is always fitted between the pump suction and discharge chambers to protect the pump should it be operated with a valve closed in the discharge line.

Reciprocating displacement pumps are self priming, will accept high suction lifts, produce the discharge pressure required by the system and can handle large amounts of vapor or entrained gases. They are, however, complicated in construction with a number of moving parts requiring attention and maintenance.

When starting the pump the suction and discharge valves must be opened. It is important that no valves in the discharge line are closed, otherwise either the relief valve will lift or damage may occur to the pump when it is started. The pump is self priming, but where possible to reduce wear or the risk of seizure it should be flooded with liquid before starting. An electrically driven pump needs only to be switched on, when it will run erratically for a short period until liquid is drawn into the pump.

Most of the moving parts in the pump will require examination during overhaul. The pump piston, rings and cylinder liner must also be thoroughly checked. Ridges will eventually develop at the limits of the piston ring travel and these must be removed. The suction and discharge valves must be refaced or ground in as required.

Two different rotary displacement pumps are shown in Figure 3.6. The action in each case results in the trapping of a quantity of liquid (or air) in a volume or space which becomes smaller at the discharge or outlet side. It should be noted that the liquid does not pass between the screw or gear teeth as they mesh but travels between the casing and the teeth.

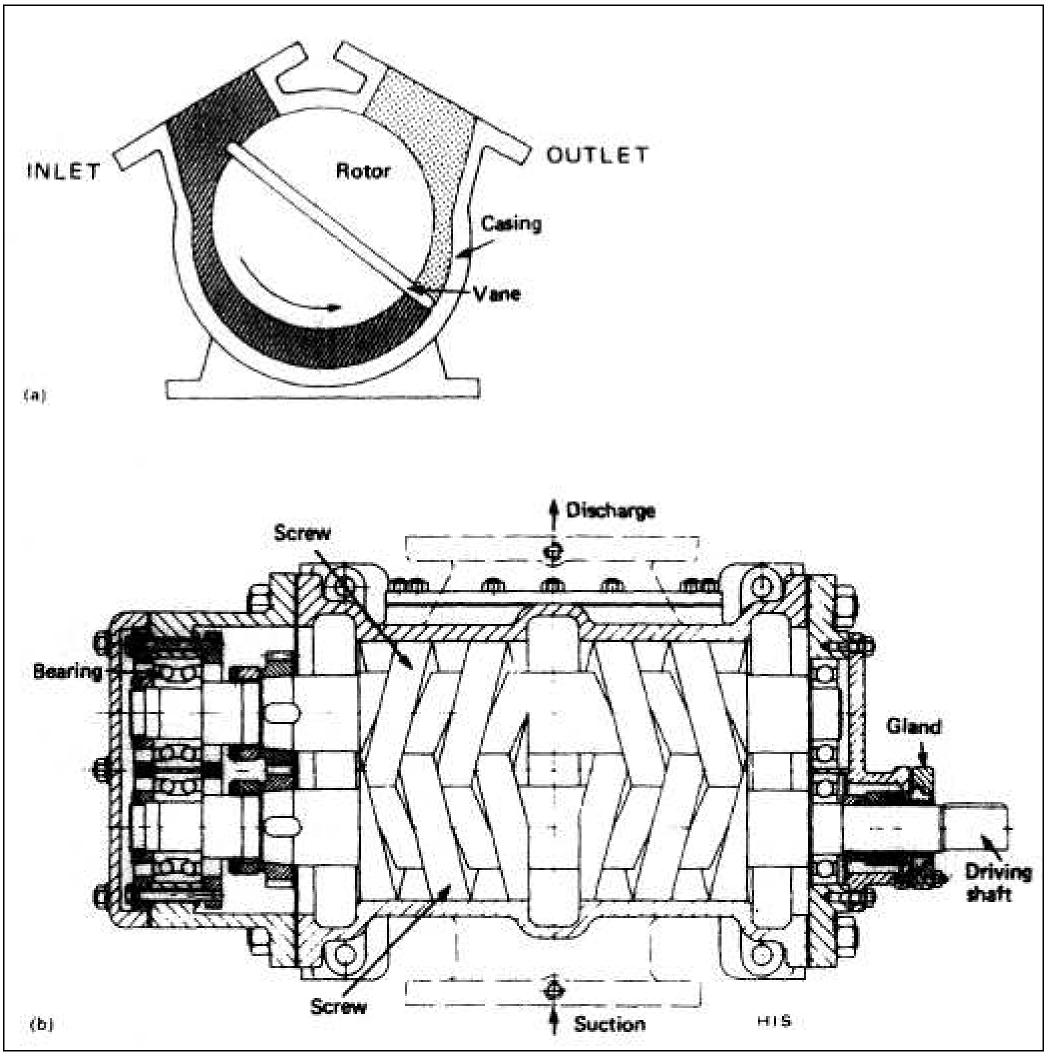


Fig. 3.6 – Rotary displacement pumps: (a) rotary vane displacement pump, (b) screw displacement pump.

The starting procedure is similar to that for the reciprocating displacement pump. Again a relief valve will be fitted between suction and discharge chambers. The particular maintenance problem with this type of pump is the shaft sealing where the gland and packing arrangement must be appropriate for the material pumped. The rotating vane type will suffer wear at a rate depending upon the liquid pumped and its freedom from abrasive or corrosive substances. The screw pump must be correctly timed and if stripped for inspection care should be taken to assemble the screws correctly.

A special type of rotary displacement pump has a particular application in steering gear and is described in section 3.6.

<u>Design of Piston Type Pumps:</u>

Delivered power of a piston type pump is given by:

$$P = M\omega = pQ \qquad (3.1)$$

, where:

M = pump torque [Nm];

ω = rotational speed [rad/s];

p = pressure [Pa];

Q = flow rate [m^3/s].

Displaced volume or flow rate per each pump rotation is expressed by Q_c, then pump torque is given by:

$$M = \frac{pQ_c}{2\pi} \tag{3.2}$$

As illustrated in Figures 2.36 and 3.7, considering a single effect piston pump the instantaneous pump flow rate expressed by Q_{inst} this given by:

$$Q_{inst} = Q_{Max} \sin\theta \tag{3.3}$$

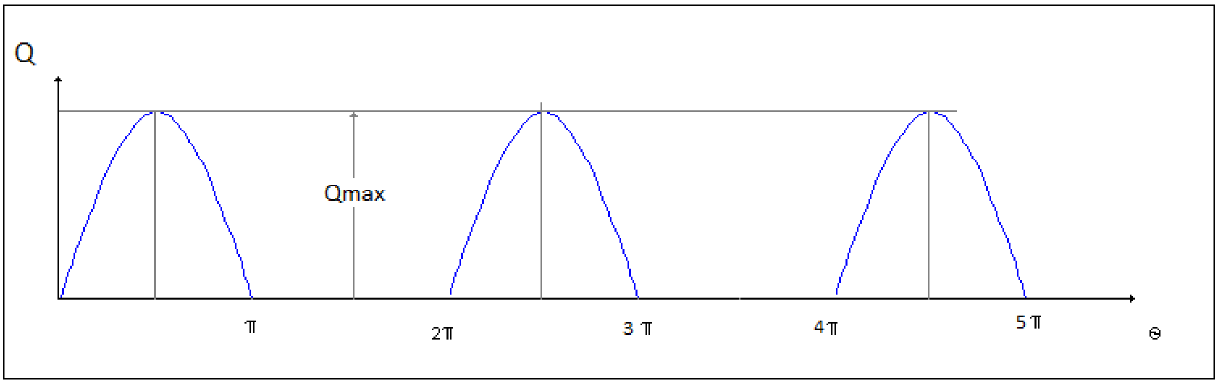


Fig. 3.7 – Axial cylinder single-effect flow-rate.

Different types of piston pumps - essentially there are two different types of piston pumps:

- axial cylinders;

- radial cylinders.

Pumps with radial cylinders can also be sub-divided into:

- pump with fixed cylinders and eccentric interior;

- pump with rotational cylinders.

Axial Cylinders Pump

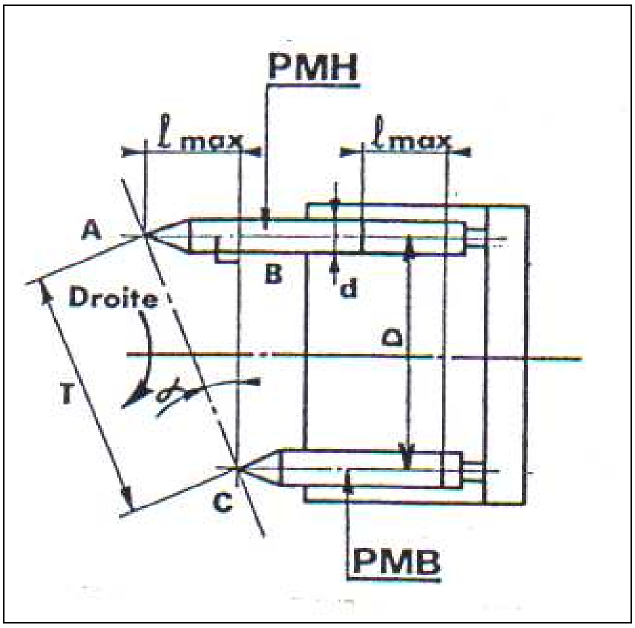


Fig. 3.8 – Axial cylinders pump flow-rate.

As shown in Figure 3.8, the axial piston stroke of a pump is given by:

$$\overline{AB} = \overline{BC}\tan\alpha = D\tan\alpha \tag{3.4}$$

, where:

D = distance between two piston diameters [m].

Now the displaced volume of the pump is given by:

$$Q_c = n.D.\tan\alpha.s \tag{3.5}$$

, where:

n = number of pistons;

$s = \dfrac{\pi}{4}d^2$ = piston surface area [m^2];

d = piston diameter [m].

<u>Design of Vane Type Pumps:</u>

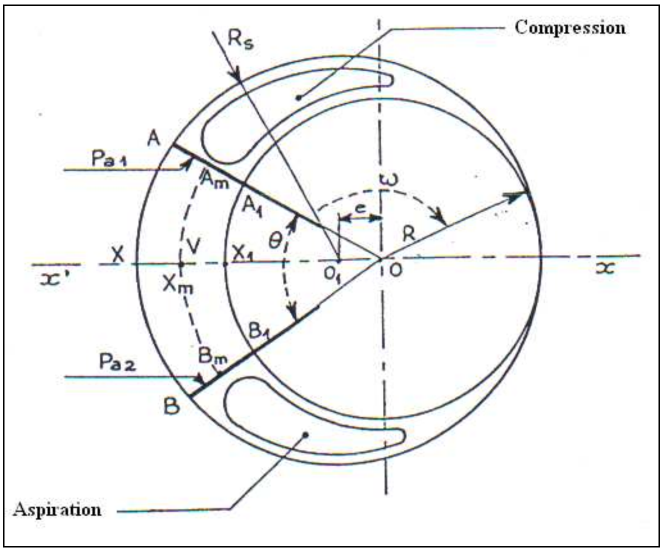


Fig. 3.9 – Rotary vane pump flow-rate.

As illustrated in Figure 3.9, displaced volume of a vane type pump is given by:

$$Q = n.N.V = 4.\pi.e.R.l.N \tag{3.6}$$

, where:

n = number of vanes;

V = volume between vanes [m^3];

R = rotor radius [m];

N = rotational speed [rpm];

e = eccentricity between rotor and pump case [m];

l = length of the vanes, measured along the pump axis [m].

As illustrated in Figure 3.9, displaced volume between two consecutive vanes is given by:

$$V = \overline{AA_1}.\overline{A_m B_m}.l \tag{3.7}$$

Noticing that the following relations can be derived:

$$\overline{AA_1} = \overline{XX_1} = 2.e \tag{3.8}$$

$$\overline{A_m B_m} = \theta.\overline{OA_m} = \frac{2\pi}{n}(R+e) \tag{3.9}$$

$$\overline{OA_m} \approx \overline{OX_m} = \left(\overline{OX} + \overline{OX_1}\right)\Big/2 = \left(R_s + e + R\right)\Big/2 = \left(2R + 2e\right)\Big/2 = R + e \tag{3.10}$$

Hence, displaced volume is given by:

$$V = 2.e.\frac{2\pi}{n}.(R+e).l \approx \frac{4\pi}{n}.e.R.l \quad , e << R \tag{3.11}$$

a.2) Axial-flow pump

An axial-flow pump uses a screw propeller to axially accelerate the liquid. The outlet passages and guide vanes are arranged to convert the velocity increase of the liquid into a pressure.

A reversible axial flow pump is shown in Figure 3.10. The pump casing is split either horizontally or vertically to provide access to the propeller.

A mechanical seal prevents leakage where the shaft leaves the casing. A thrust bearing of the tilting pad type is fitted on the drive shaft. The prime mover may be an electric motor or a steam turbine.

The axial flow pump is used where large quantities of water at a low head are required, for example in condenser circulating. The efficiency is equivalent to a low lift centrifugal pump and the higher speeds possible enable a smaller driving motor to be used. The axial-flow pump is also suitable for supplementary use in a condenser scoop circulating system since the pump will offer little resistance to flow when idling.

With scoop circulation the normal movement of the ship will draw in water; the pump would be in use only when the ship was moving slowly or stopped.

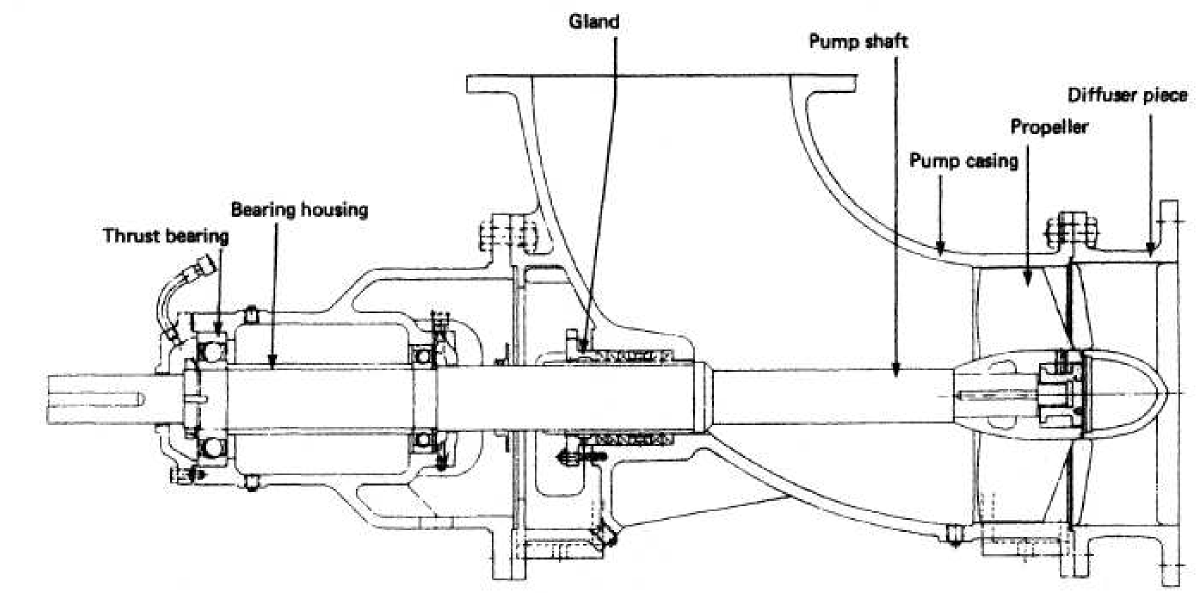


Fig. 3.10 – Axial-flow pump.

a.3) Centrifugal pump

In a centrifugal pump liquid enters the centre or eye of the impeller and flows radially out between the vanes, its velocity being increased by the impeller rotation. A diffuser or volute is then used to convert most of the kinetic energy in the liquid into pressure. The arrangement is shown diagrammatically in Figure 3.11.

A vertical, single stage, single entry, centrifugal pump for general marine duties is shown in Figure 3.12. The main frame and casing, together with a motor support bracket, house the pumping element assembly. The pumping element is made up of a top cover, a pump shaft, an impeller, a bearing bush and a sealing arrangement around the shaft. The sealing arrangement may be a packed gland or a mechanical seal and the bearing lubrication system will vary according to the type of seal. Replaceable wear rings are fitted to the impeller and the casing.

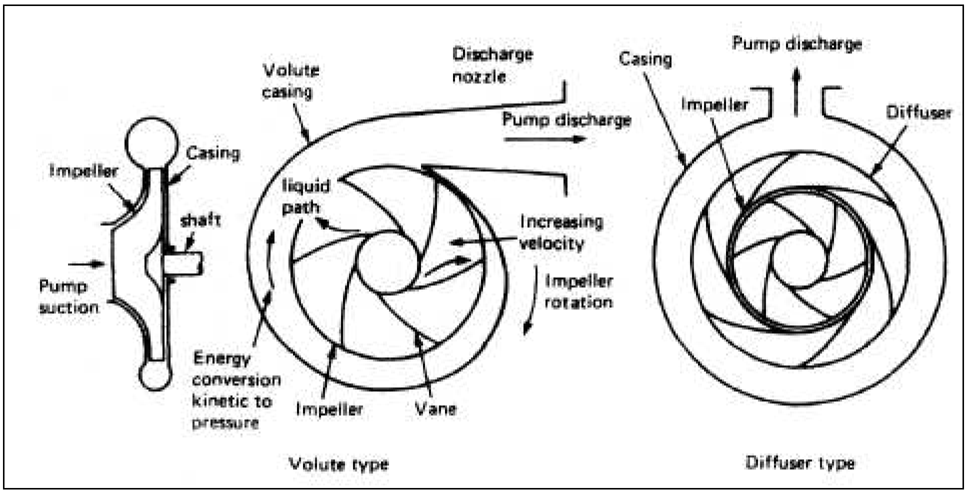


Fig. 3.11 – Centrifugal pump schematic.

The motor support bracket has two large apertures to provide access to the pumping element, and a coupling spacer is fitted between the motor and pump shaft to enable the removal of the pumping element without disturbing the motor.

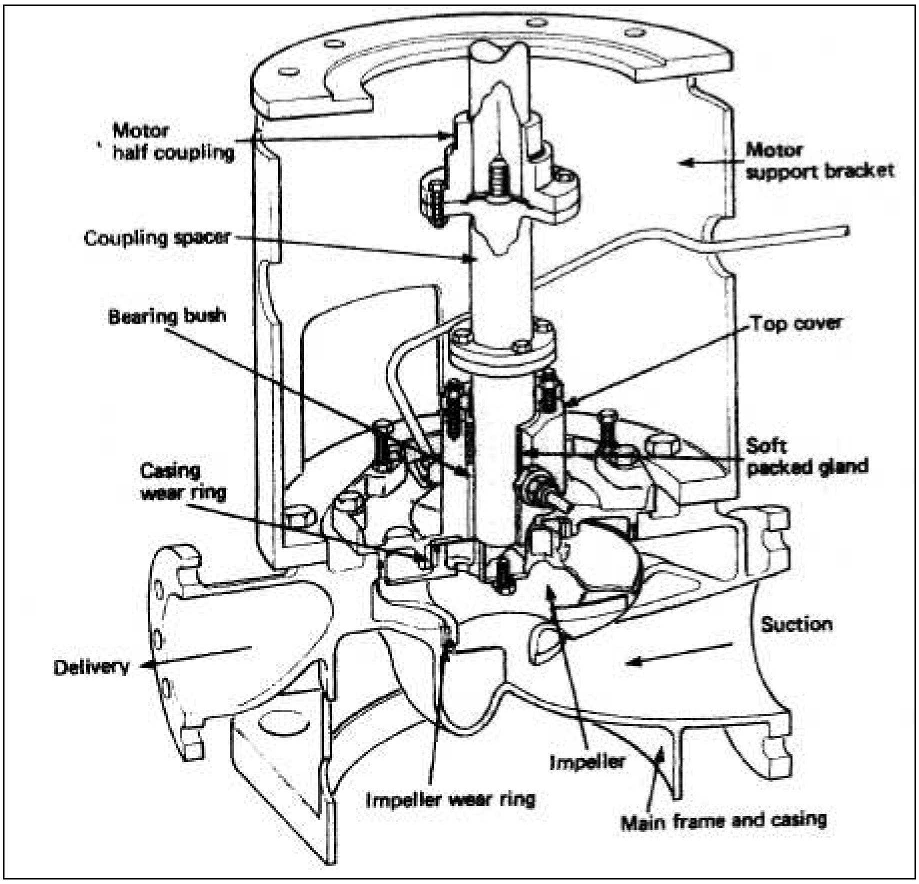


Fig. 3.12 – Single stage centrifugal pump.

3.3. Hydraulic actuators

Hydraulic actuators convert hydraulic into mechanical power.

There are two different types of hydraulic actuators:

a) The cylinders (or rams), whose mechanical power is transmitted in the form of a linear (axial) displacement;

b) The rotational actuators, whose mechanical power is transmitted in the form of a rotational displacement.

Hydraulic cylinders may be classified as:

a.1) Single effect piston – hydraulic fluid action is exerted on a single direction and return to initial position is achieved by means of a mechanical spring (see Figure 3.13);

a.2) Double effect piston – hydraulic fluid action is exerted on both directions (see Figure 3.14);

a.3) Differential effect piston – likewise the double effect piston, the hydraulic fluid action is exerted on both directions, however, since the two piston action areas are not exactly the same, then different working pressures are provided to upper and lower sides of the piston (see Figure 3.15).

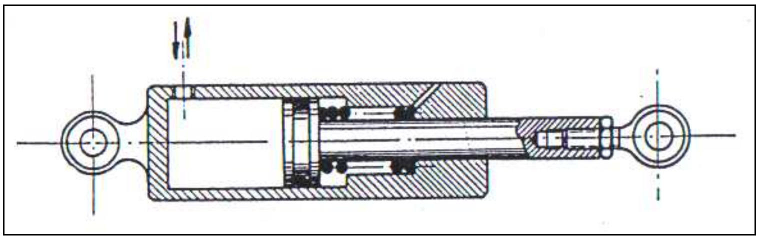


Fig. 3.13 – Single effect piston.

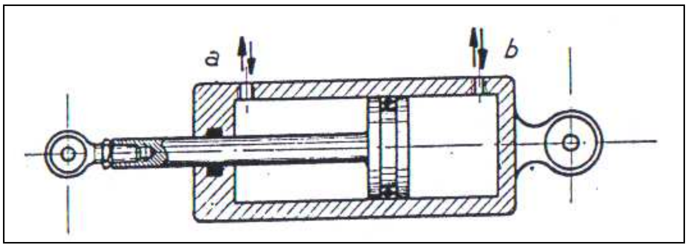


Fig. 3.14 – Double effect piston.

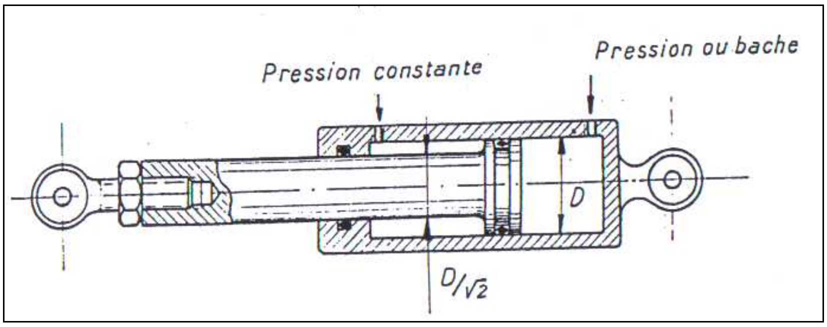


Fig. 3.15 – Differential effect piston.

A cylinder is usually made of a sleeve, a piston and sealing devices.

The rotational actuators, as suggested by its name, are actuated to provide a certain rotational displacement, which might correspond to a quite small angle. Actually, these rotational actuators are most of the times applied when a small angular displacement and a large torque is required, like it happens in the case of a steering gear.

Hydraulic motors may be divided into four categories:

- Radial piston motors, usually these are low speed motors (see Figure 3.16);

- Axial piston motors, usually these are high speed motors;

- Rotary gear motors;

- Rotary vanes motors.

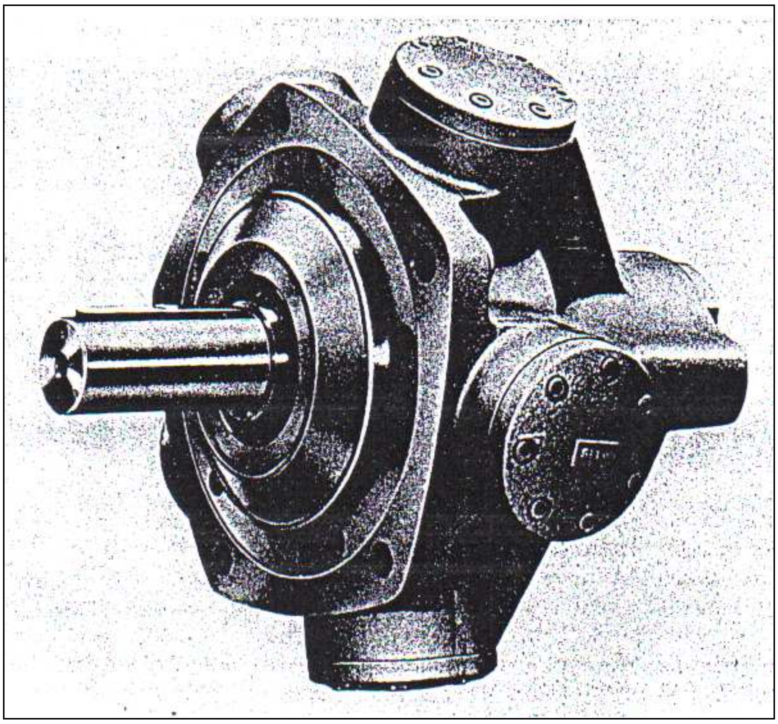


Fig. 3.16 – Five piston radial slow speed high torque motor .

3.4. Hydraulic power units

Hydraulic Power Units (HPU) convert mechanical into hydraulic power, supplied in the form of an highly pressurized hydraulic oil flow.

There are four distinct configuration of HPU, namely:

a) HPU with a constant-flow pump and unloading and control valve. These are utilized when a moderate pressure and flow are required (see Figure 3.17);

b) HPU with a variable-delivery pressure-compensated pump. These are utilized when a pressure up to 200 [bar] and power up to 50 [KW] are required (see Figure 3.18);

c) HPU with a variable-delivery pump and feeding pump (see Figure 3.19);

d) HPU with accumulator and constant-flow pump control (see Figure 3.20).

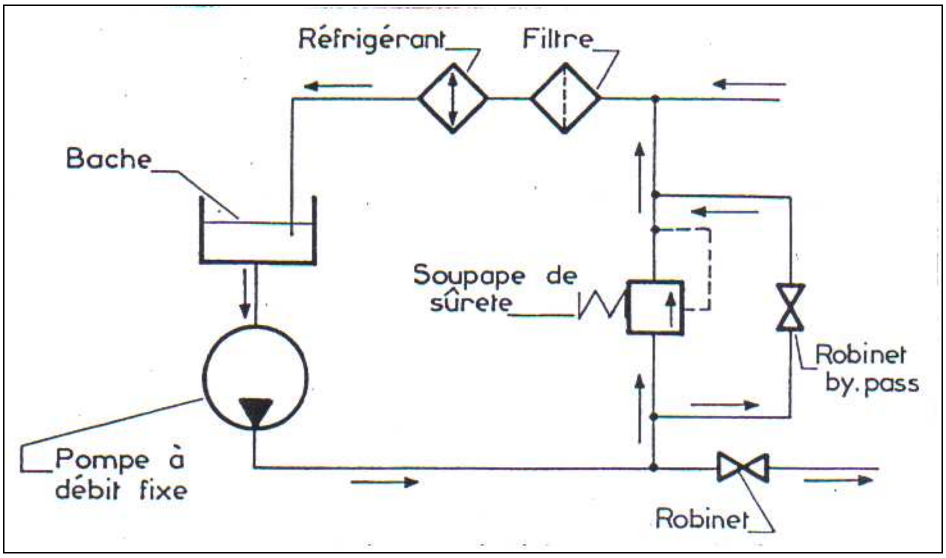


Fig. 3.17 – Hydraulic power unit with a constant-flow pump and unloading and control valve.

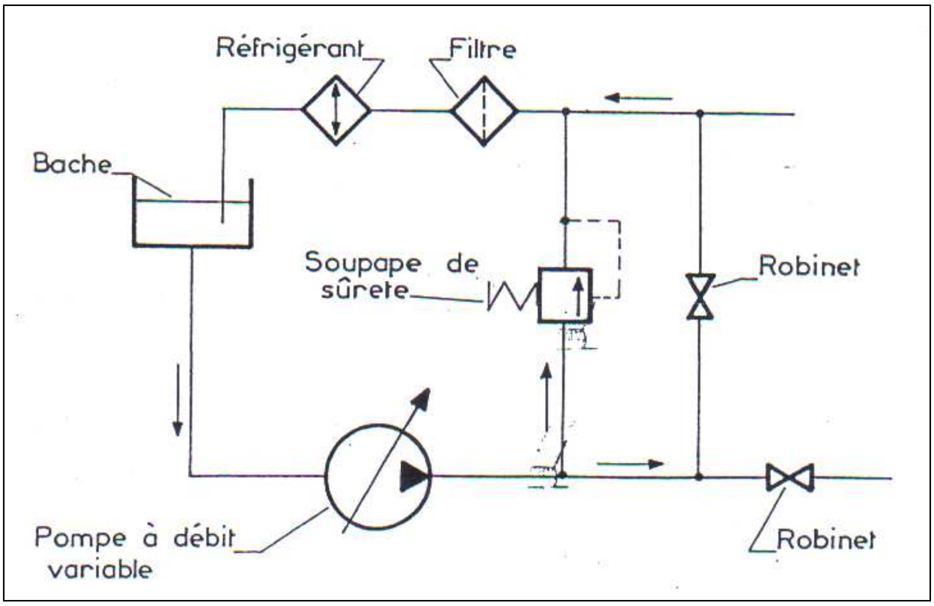


Fig. 3.18 – Hydraulic power unit with a variable-delivery pressure-compensated pump.

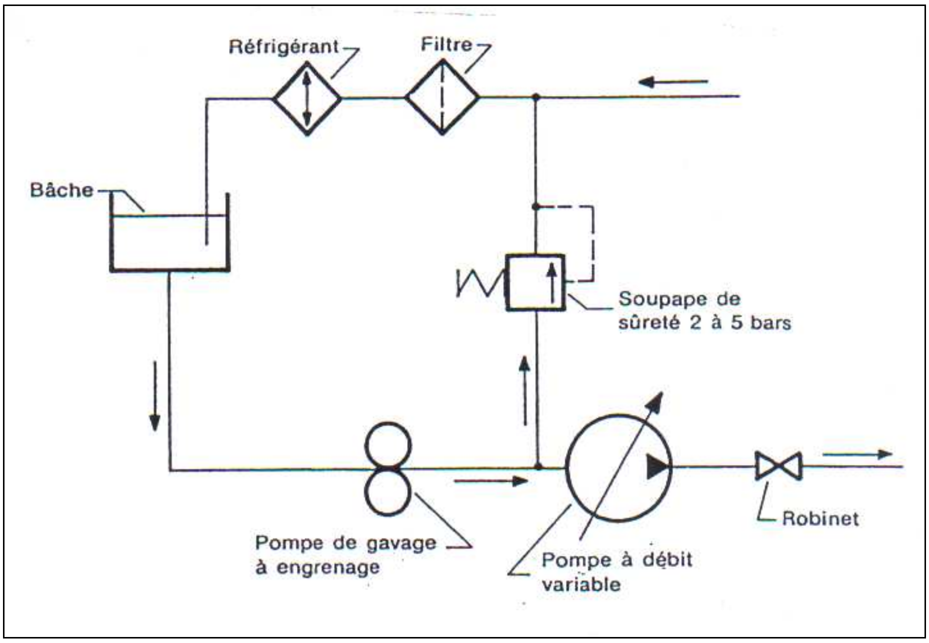


Fig. 3.19 – Hydraulic power unit with a variable-delivery pump and feeding pump.

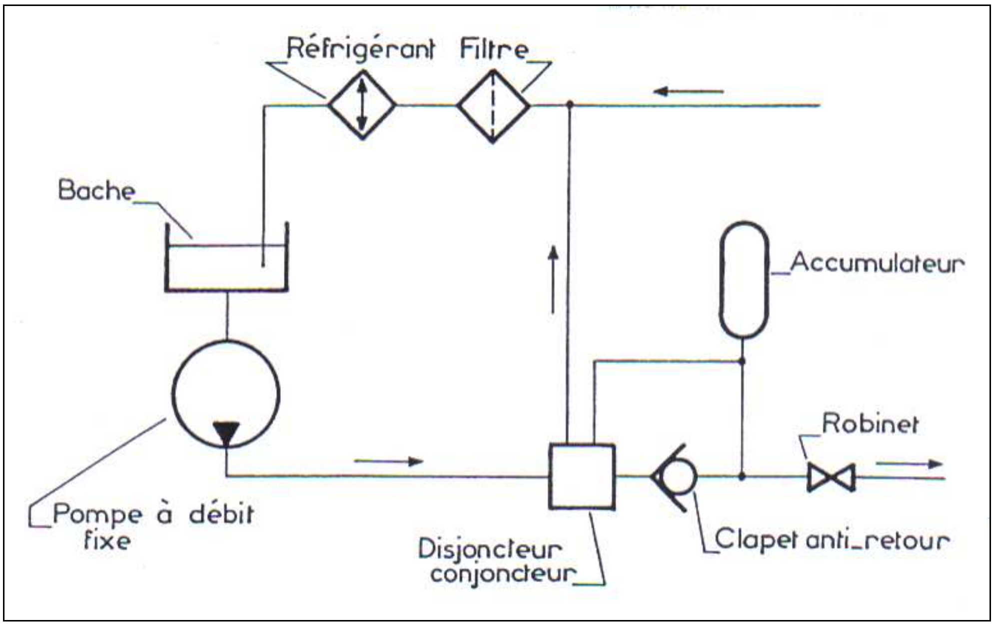


Fig. 3.20 – Hydraulic power with accumulator and constant-flow pump control.

3.5. Steering gear

The steering gear provides a movement of the rudder in response to a signal from the bridge. The total system may be considered made up of three parts:

a) control equipment;

b) power unit;

c) transmission to the rudder stock.

As illustrated in Figures 3.21 and 3.22, the control equipment conveys a signal of desired rudder angle from the bridge and activates the power unit and transmission system until the desired angle is reached. The power unit provides the force, when required and with immediate effect, to move the rudder to the desired angle. The transmission system, also called the steering gear, is the means by which the movement of the rudder is accomplished.

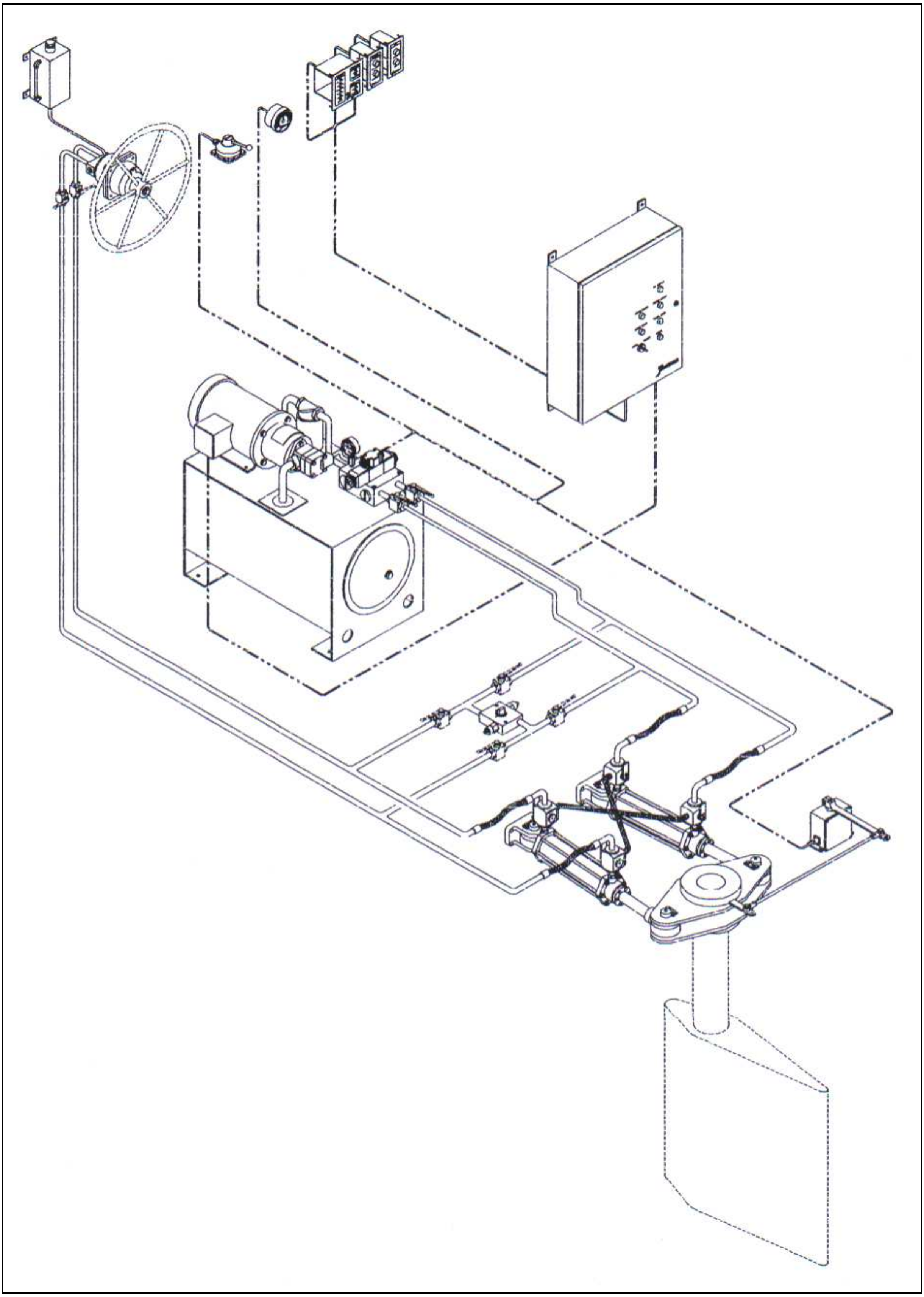


Fig. 3.21 – Typical steering gear arrangement of a large vessel.

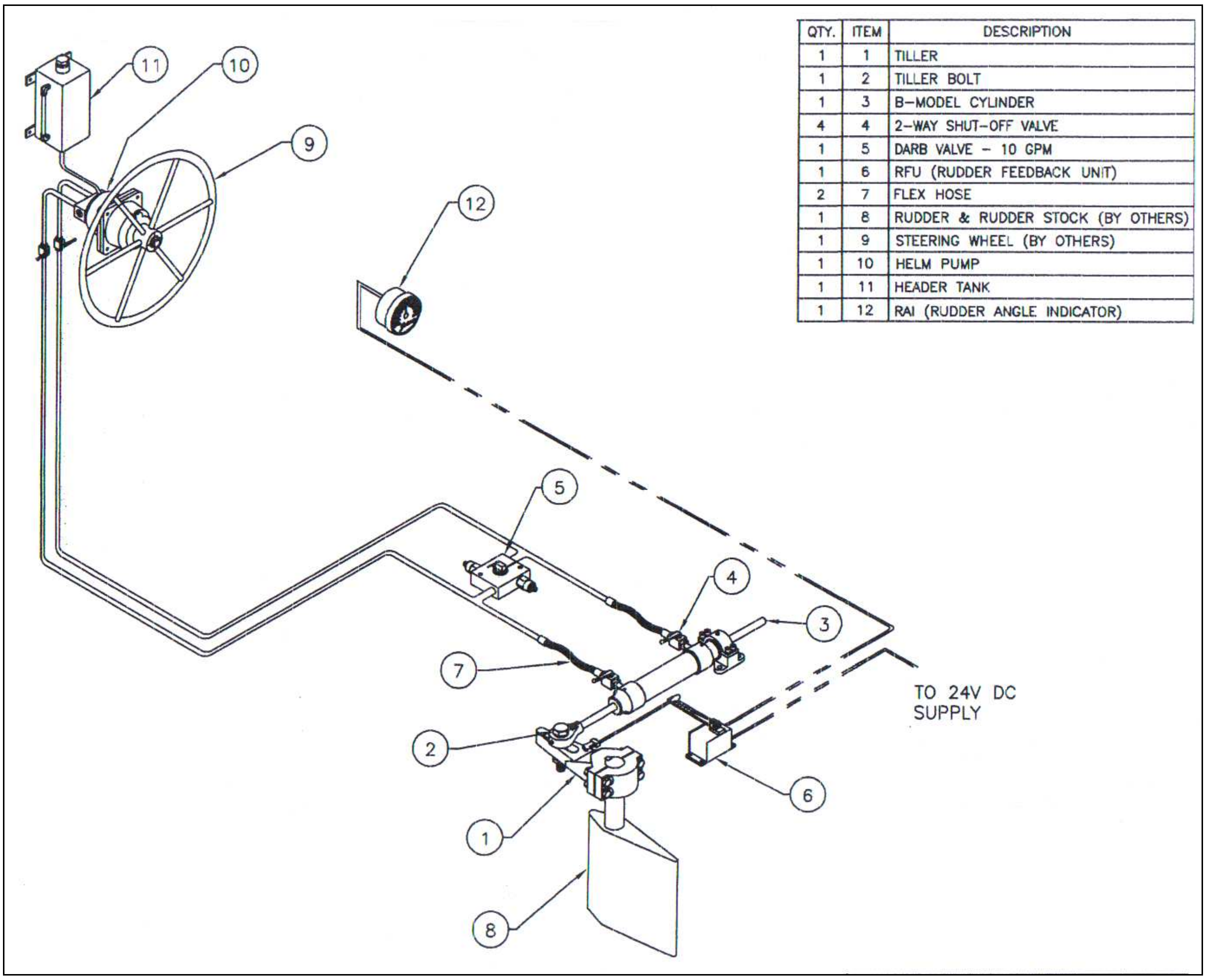


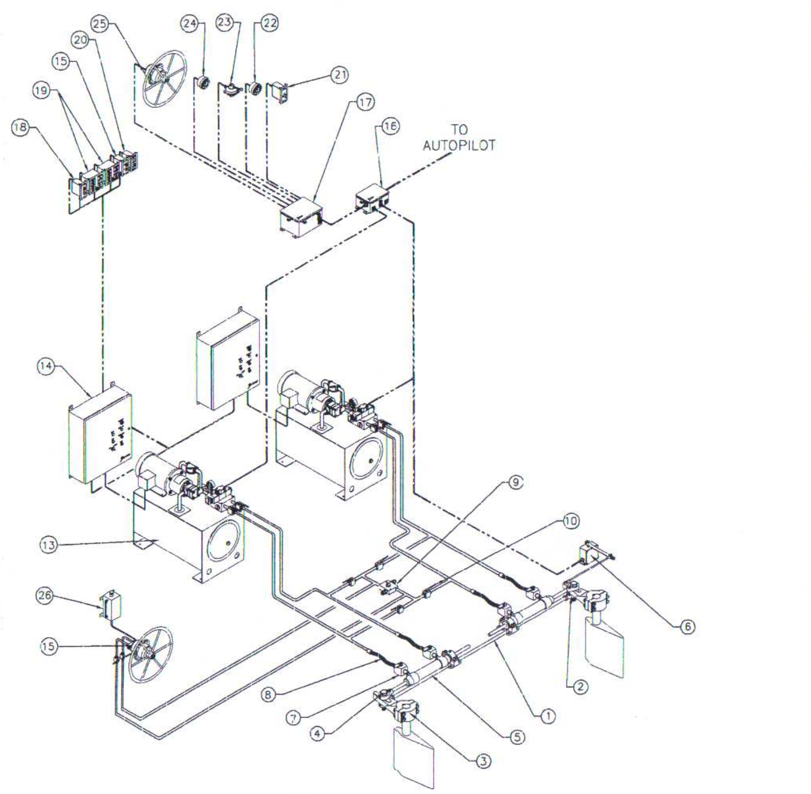
Fig. 3.22 – Typical steering gear arrangement of a small boat.

Certain requirements, established by IMO Protocol 1978 and Classification Societies rules, must currently be met by a ship's steering system. There must be two independent means of steering, although where two identical power units are provided an auxiliary unit is not required. The power and torque capability must be such that the rudder can be swung from 35° one side to 35º the other side with the ship at maximum speed, and also the time to swing from 35° one side to 30° the other side must not exceed 28 seconds. The system must be protected from shock loading and have pipework which is exclusive to it as well as be constructed from approved materials. Control of the steering gear must be provided in the steering gear compartment.

Tankers of 10,000 ton gross tonnage and upwards must have two independent steering gear control systems which are operated from the bridge. Where one fails, changeover to the other must be immediate and achieved from the bridge position. The steering gear itself must comprise two independent systems where a failure of one results in an automatic changeover to the other within 45 seconds. Any of these failures should result in audible and visual alarms on the bridge.

The design pressure of all the steering gear components should be assumed to be at least equal to the greater of the 1.25 times the maximum working pressure or the relief valve setting.

Steering gears can be arranged with hydraulic control equipment known as a 'telemeter', or with electrical control equipment (see Figure 3.23). The power unit may in turn be hydraulic or electrically operated. Each of these units will be considered in turn, with the hydraulic unit pump being considered first. A pump is required in the hydraulic system which can immediately pump fluid in order to provide a hydraulic force that will move the rudder. Instant response does not allow time for the pump to be switched on and therefore a constantly running pump is required which pumps fluid only when required. A variable delivery pump provides this facility.



QTY.	ITEM	DESCRIPTION
1	1	TIE BAR ASSEMBLY (BY OTHERS)
2	2	ROD END
2	3	TILLER ASSEMBLY
2	4	TILLER BOLT – EXTRA LONG
2	5	B-MODEL CYLINDER
1	6	RFU-2000 (RUDDER FEEDBACK UNIT)
4	7	2-WAY SHUT OFF VALVE
4	8	FLEX HOSE
1	9	DARB (DOUBLE ACTING RELIEF & BYPASS) VALVE
6	10	SHUT-OFF VALVE (BY OTHERS)
1	11	H58 (HELM PUMP)
1	12	HEADER TANK
2	13	HPU (HYDRAULIC POWER UNIT)
2	14	MSA (MOTOR STARTER & ALARM) BOX
1	15	AP-36 (ALARM PANEL)
1	16	SCA-200 (STEERING CONTROL AMPLIFIER)
1	17	WHJB (WHEELHOUSE JUNCTION BOX)
1	18	TP36-2 (TEST PANEL)
2	19	AP-66-2 (ALARM PANEL)
1	20	SP36-2 (STARTER PANEL)
1	21	CP36-1 (MODE SELECTOR PANEL)
1	22	RAI-300 (RUDDER ANGLE INDICATOR)
1	23	J0100 (JOG LEVER)
1	24	ROI-300 (RUDDER ORDER INDICATOR)
1	25	EW200 (ELECTRIC WHEEL)
1	26	HEADER TANK

NOTE:
–SOLID LINES REPRESENT PIPING
–DASH LINES REPRESENT WIRING FUNCTION
(NOT CABLE REQUIREMENT)

Fig. 3.23 – Steering gear with electric helm control.

3.5.1 Variable delivery pumps

A number of different designs of variable delivery pump exist. Each has a means of altering the pump stroke so that the amount of oil displaced will vary from zero to some designed maximum value. This is achieved by use of a floating ring, a swash plate or a slipper pad.

a) The radial cylinder (Hele-Shaw) pump

The radial cylinder (Hele-Shaw) pump is shown in Figure 3.24.

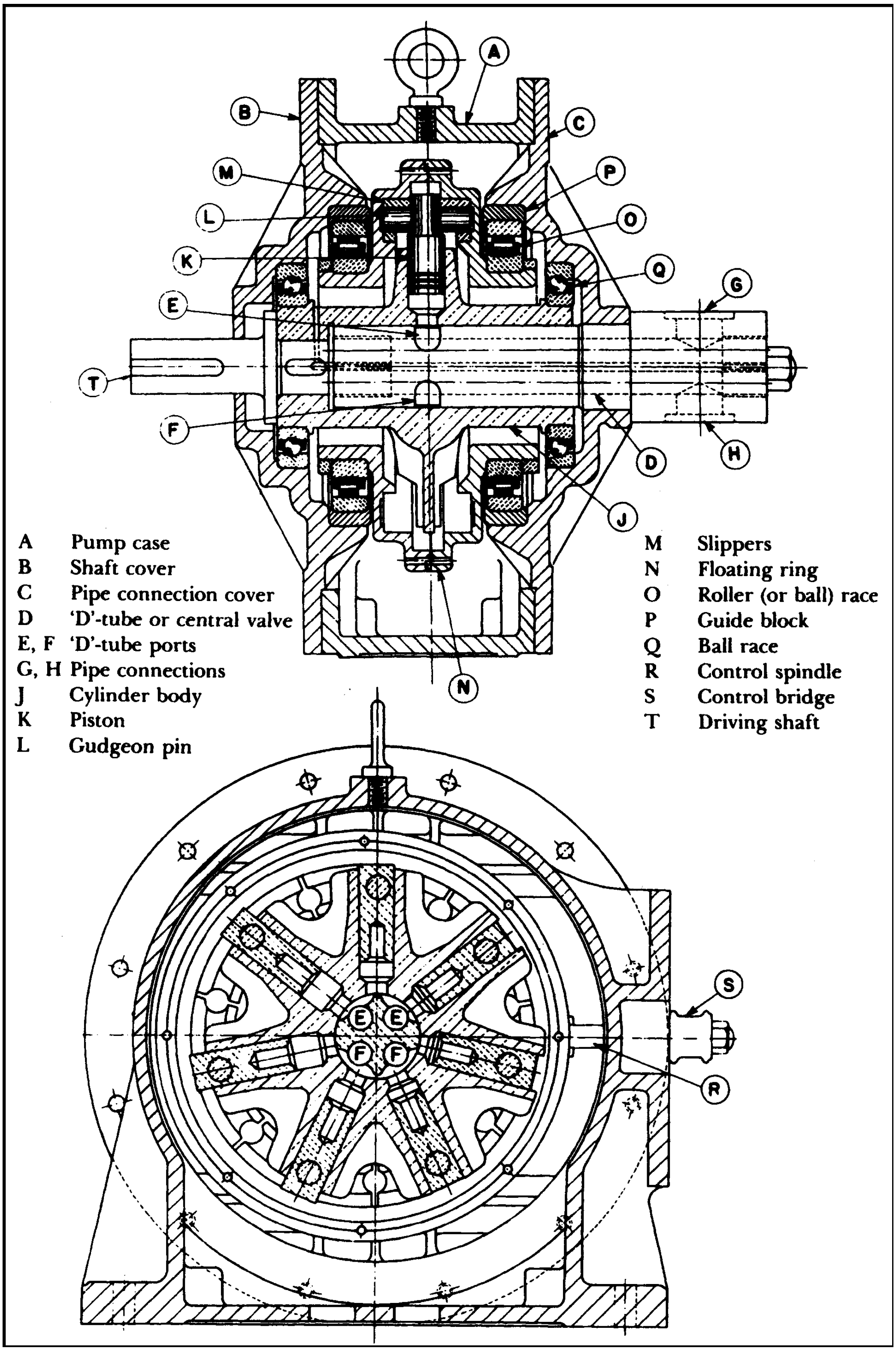


Fig. 3.24 – Hele Shaw pump.

Within the casing a short length of shaft drives the cylinder body which rotates around a central valve or tube arrangement and is supported at the ends by ball bearings. The cylinder body is connected to the central valve arrangement by ports which lead to connections at the outer casing for the supply and delivery of oil. A number of pistons fit in the radial cylinders and are fastened to slippers by a gudgeon pin. The slippers fit into a track in the circular floating ring. This ring may rotate, being supported by ball bearings, and can also move from side to side since the bearings are mounted in guide blocks. Two spindles which pass out of the pump casing control the movement of the ring.

The operating principle will now be described by reference to Figure 3.25. When the circular floating ring is concentric with the central valve arrangement the pistons have no relative reciprocating motion in their cylinders (Figure 3.25(a)). As a result no oil is pumped and the pump, although rotating, is not delivering any fluid. If however the circular floating ring is pulled to the right then a relative reciprocating motion of the pistons in their cylinders does occur (Figure 3.25(b)). The lower piston, for instance, as it moves inwards will discharge fluid out through the lower port in the central valve arrangement. As it continues past the horizontal position the piston moves outwards, drawing in fluid from the upper port. Once past the horizontal position on the opposite side, it begins to discharge the fluid. If the circular floating ring were pushed to the left then the suction and discharge ports would be reversed (Figure 3.25(c)).

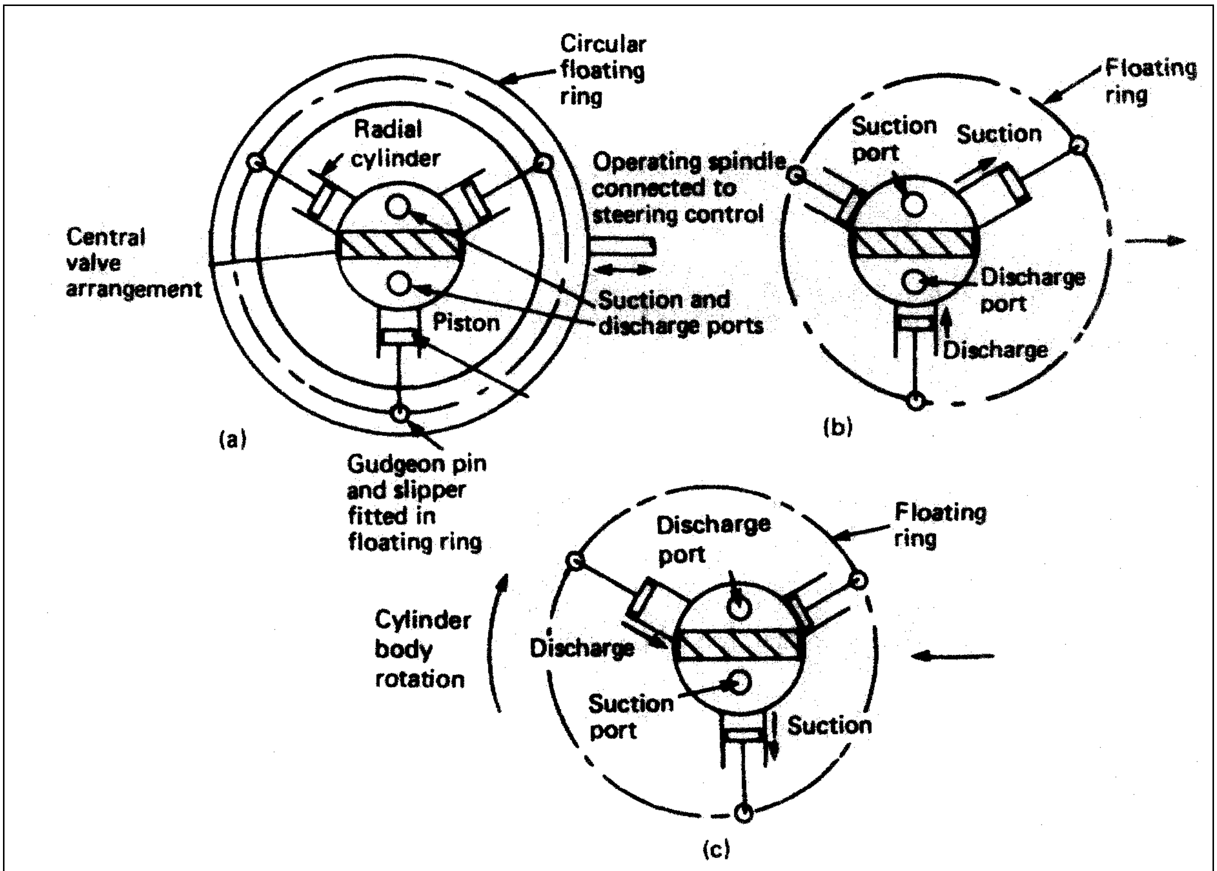


Fig. 3.25 – Hele Shaw pump – operating principle.

This pump arrangement therefore provides, for a constantly rotating unit, a no-flow condition and infinitely variable delivery in either direction. The pump is also a positive displacement unit. Where two pumps are fitted in a system and only one is operating, reverse operation might occur. Non-reversing locking gear is provided as part of the flexible coupling and is automatic in operation. When a pump is stopped the locking gear comes into action; as the pump is started the locking gear releases.

b) The swash plate axial cylinder pump

This pump (Figure 3.26) has a circular cylinder block with axial cylinders disposed on a pitch circle around a central bore which is machined with splines to suit the input shaft with which it revolves. The individual cylinders are parallel with the shaft, with one of each terminating in a drilled port at the end face of the block. This face bears against a stationary valve plate and is maintained in contact by spring pressure. Semi circular ports in the valve plate, in line with those from the cylinders, are connected by external pipes to the steering cylinders for the ram type configuration.

Each cylinder contains a piston, connected by a double ball-ended rod to a swash plate driven by the input shaft.

When the swash plate is vertical, cylinder barrel and pistons revolve in the same plane and the pistons have no stroke. As the swash plate is tilted stroke is given to the pistons at each half revolution, the length of the stroke being determined by the angle of tilt.

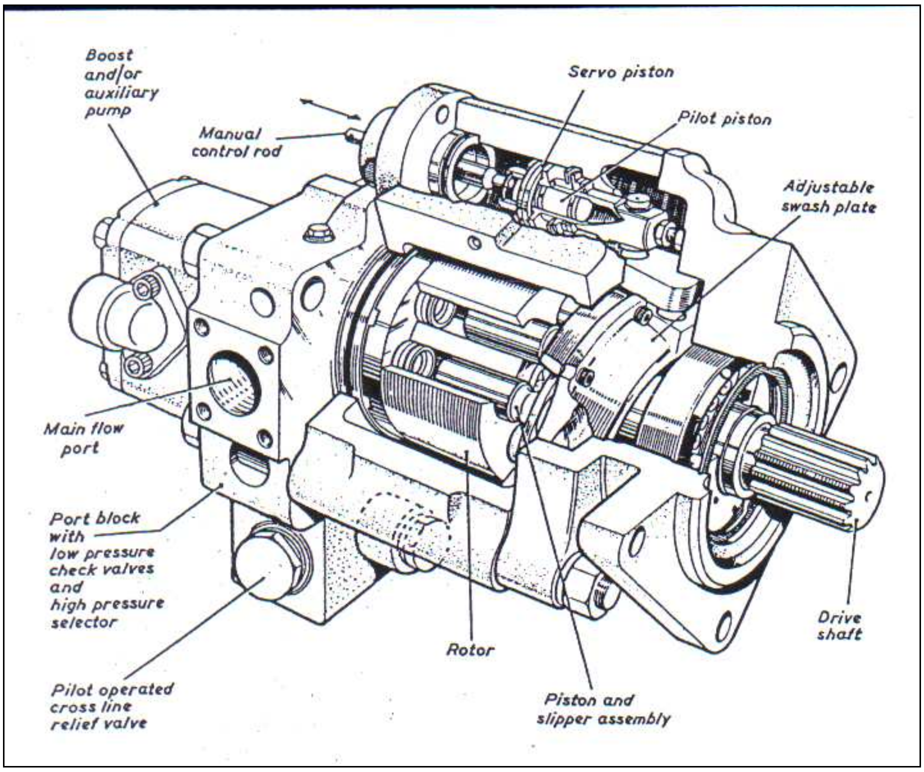


Fig. 3.26 – Swash plate axial cylinder pump.

c) The slipper pad axial cylinder pump

This is another development of the pump described above, suitable for the higher pressures demanded as steering gear and fin stabilizer systems were developed. The connecting rods are replaced by slipper pads in the swash plate, the spherical ends of the pistons being carried in the pads. Figure 3.27 shows a cut-away section.

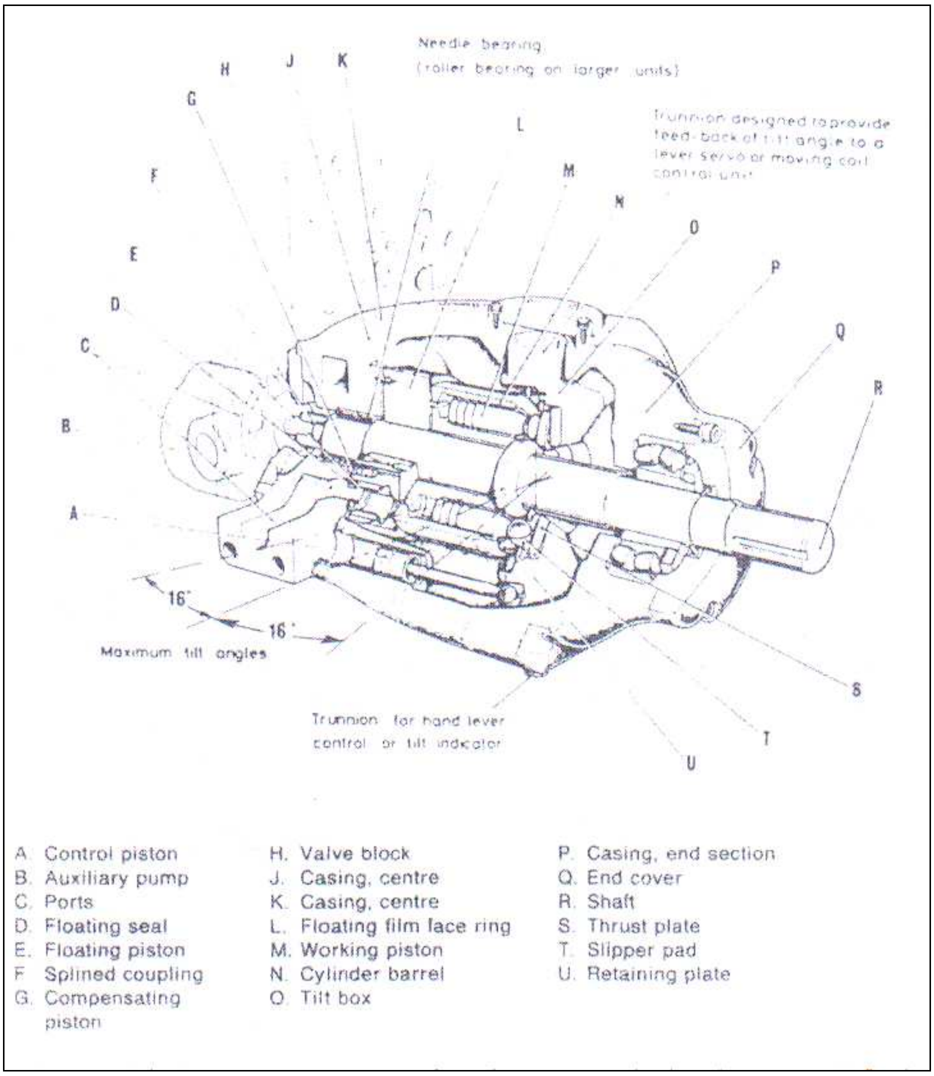
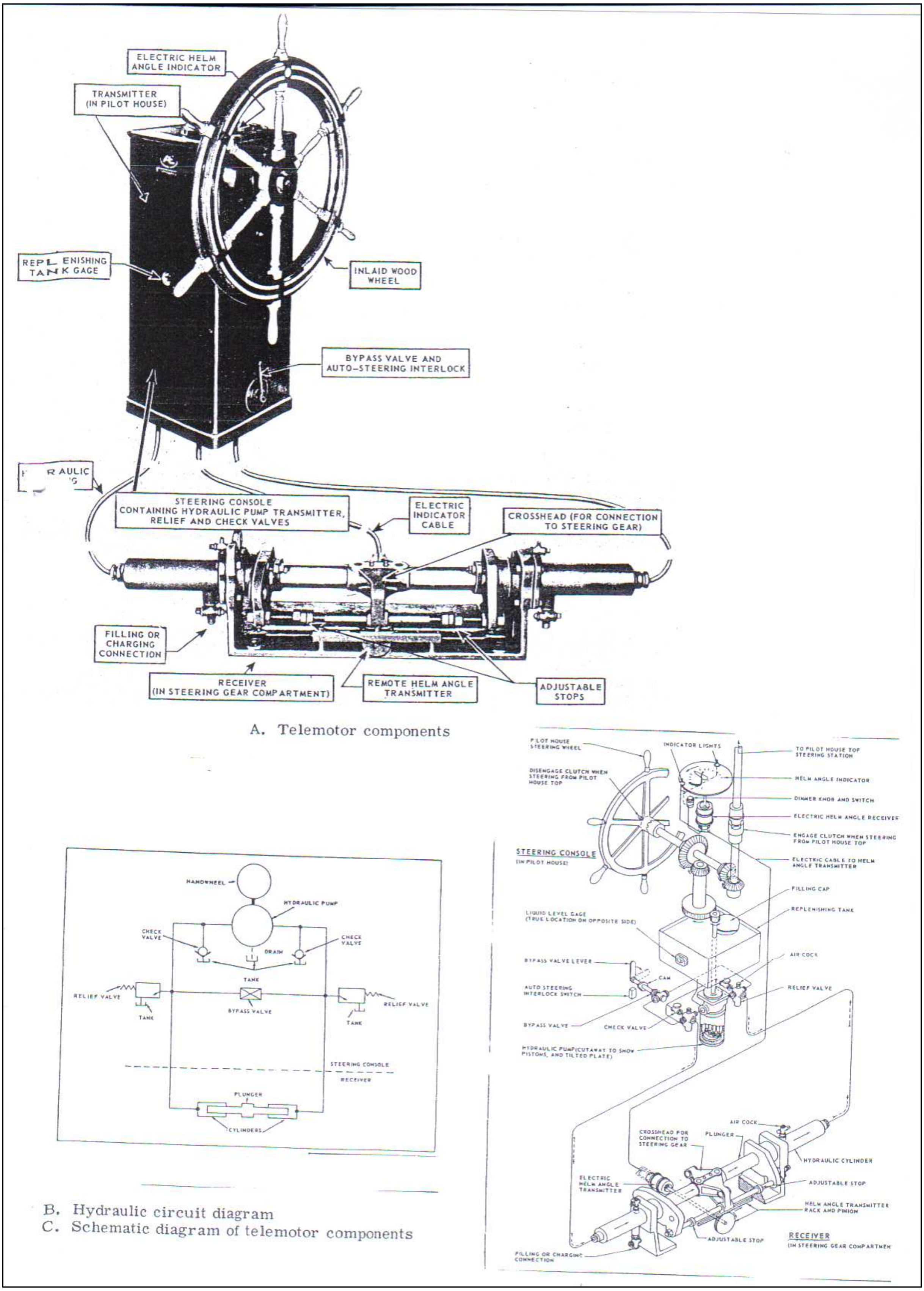


Fig. 3.27 – Slipper pad axial cylinder pump.

3.5.2 Telemotor control

Telemotor control is a hydraulic control system employing a transmitter, a receiver, pipes and a charging unit. The transmitter, which is built into the steering wheel console, is located on the bridge and the receiver is mounted on the steering gear. The charging unit is located near to the receiver and the system is charged with a non-freezing fluid.

The telemotor system is shown in Figure 3.28. Two rams are present in the transmitter which move in opposite directions as the steering wheel is turned. The fluid is therefore pumped down one pipe line and drawn in from the other. The pumped fluid passes through piping to the receiver and forces the telemotor cylinder unit to move. The suction of fluid from the opposite cylinder enables this movement to take place.



A. Telemotor components

B. Hydraulic circuit diagram
C. Schematic diagram of telemotor components

Fig. 3.28 – Hydraulic telemotor control.

In normal operation the working pressure of about 20 to 30 [bar], or the manufacturer's given figure, should not be exceeded. The wheel should not be forced beyond the 'hard over' position as this will strain the gear. The replenishing tank should be checked regularly and any lubrication points should receive attention. Any leaking or damaged equipment must be repaired or replaced as soon as possible. The system should be regularly checked for pressure tightness. The rudder response to wheel movement should be checked and if sluggish or slow then air venting undertaken. If, after long service, air venting does not remove sluggishness, it may be necessary to recharge the system with new fluid.

3.5.3 Power transmission units

Three types of hydraulically powered transmission units or steering gears are in common use, namely:

a) Ram type;

b) Rotary vane type;

c) Spherical type.

A two-ram steering gear is shown in Figure 3.29.

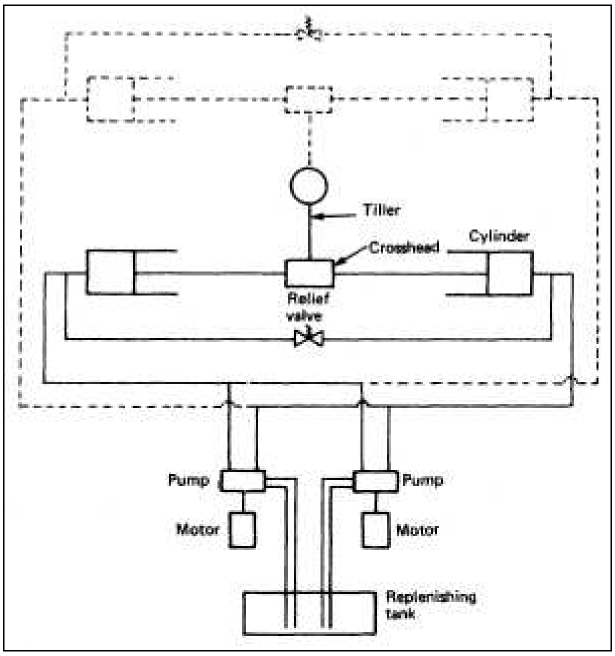


Fig. 3.29 – Diagrammatic arrangement of two-ram steering gear (additional items for four-ram system shown dotted).

The rams acting in hydraulic cylinders operate the tiller by means of a swivel crosshead carried in a fork of the rams. A variable delivery pump is mounted on each cylinder and the slipper ring is linked by rods to the control spindle of the telemotor receiver. The variable delivery pump is piped to each cylinder to enable suction or discharge from either. A replenishing tank is mounted nearby and arranged with non-return suction valves which automatically provide make-up fluid to the pumps.

A bypass valve is combined with spring-loaded shock valves which open in the event of a very heavy sea forcing the rudder over. In moving over, the pump is actuated and the steering gear will return the rudder to its original position once the heavy sea has passed. A spring-loaded return linkage on the tiller will prevent damage to the control gear during a shock movement.

During normal operation one pump will be running. If a faster response is required, for instance in confined waters, both pumps may be in use. The pumps will be in the no-delivery state until a rudder movement is required by a signal from the bridge telemotor transmitter.

The telemotor receiver cylinder will then move: this will result in a movement of the floating lever which will move the floating ring or slipper pad of the pump, causing a pumping action. Fluid will be drawn from one cylinder and pumped to the other, thus turning the tiller and the rudder. A return linkage or hunting gear mounted on the tiller will reposition the floating lever so that no pumping occurs when the required rudder angle is reached.

Ram type hydraulic steering gear can also be sub-divided into different configurations:

a.1) Rapson slide (see Figure 3.30);

a.2) Articulated rod or link (see Figure 3.31).

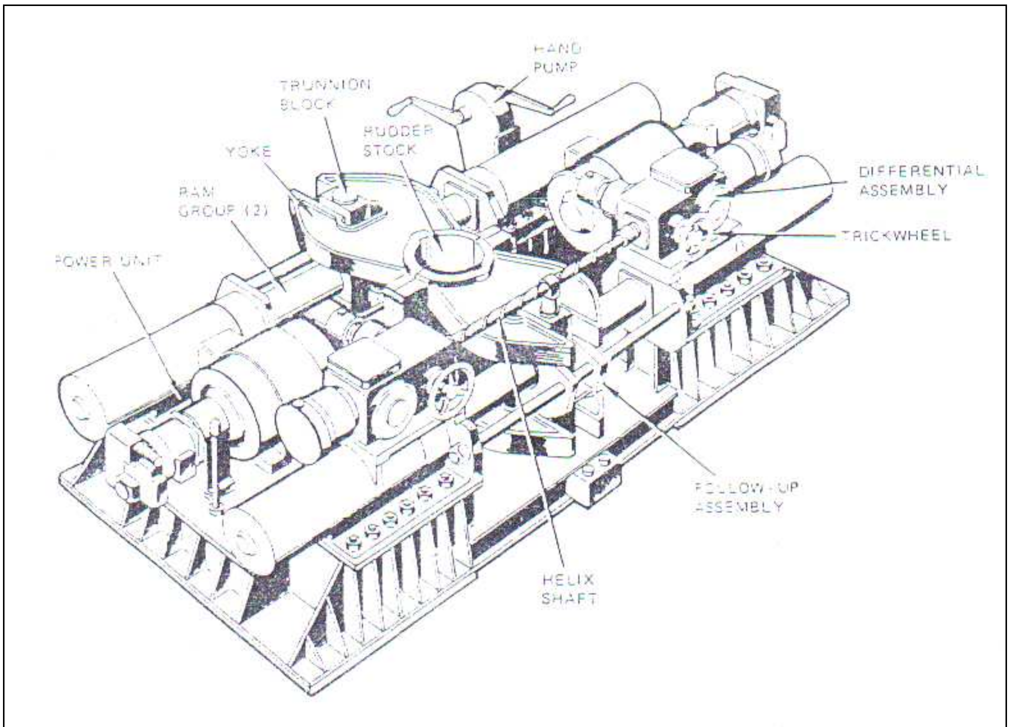


Fig. 3.30 – Rapson slide type steering gear.

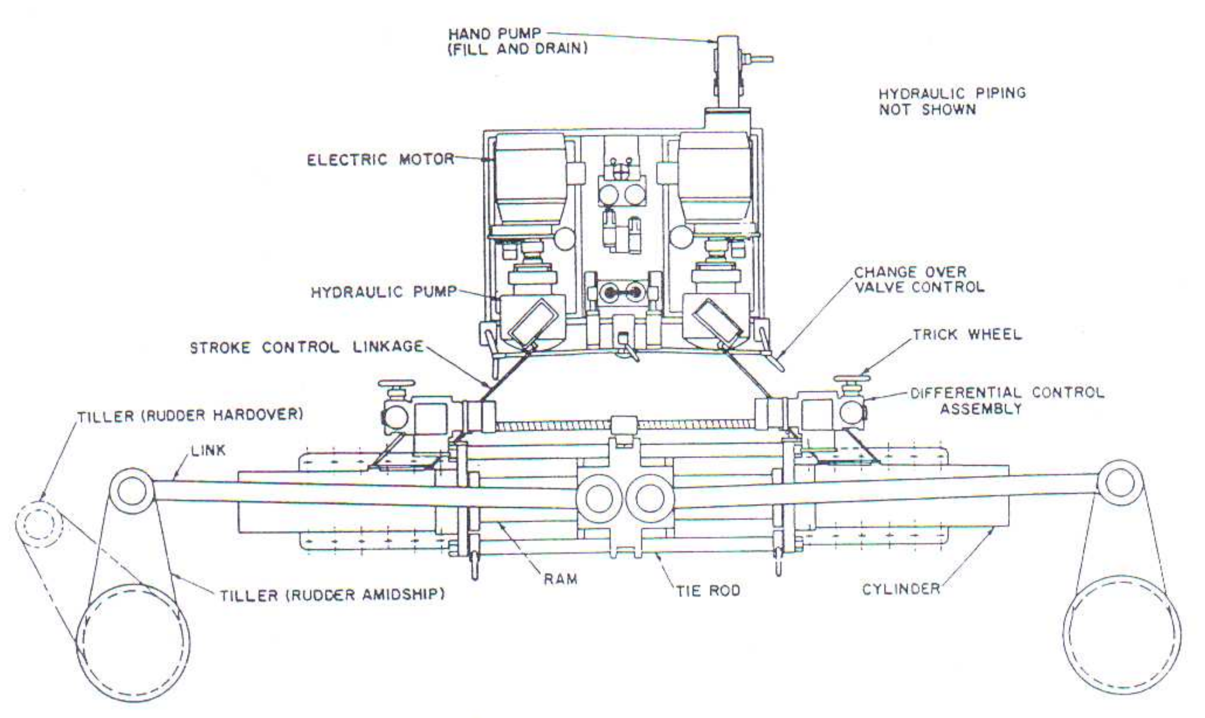


Fig. 3.31 – Articulated rod or link type steering gear for a twin-rudder ship.

Rapson slide is particularly well suited for higher rudder torque ratings because its arrangement provides an increasing mechanical advantage at larger rudder angles (see Figure 3.33). On the other hand, link-type steering gears generally have a decreasing, rather than increasing, mechanical advantage at larger rudder angles (see Figure 3.32). However, there are two cases for which a link type steering gear is ideally suited. One is where there is insufficient space around the rudderstock to permit the installation of rams. By comparing Figures 3.30 and 3.31, it will be seen that the space required in way of the rudderstock is considerably different for the two types. The other case is for twin-rudder ship, which can be arranged so that one link-type steering gear can serve both rudders, as shown in Figure 3.31.

<u>Rudder torque rating:</u>

One of the more difficult aspects of ship design process is the calculation of the maximum rudder torque for which the steering gear should be designed. Although ship specifications may stipulate the maximum design rudder torque, in some instances they usually state that the steering gear shall be capable of moving the rudder at a prescribed rate when the ship is proceeding at maximum ahead speed. With the specification written in this manner, the shipbuilder has the responsibility of determining the maximum design rudder torque. Assuming the ram force may be determined for each rudder angle, based on a specific NACA profile, the torque developed by a single-ram link arrangement is given by:

$$Q = F.R.\cos\alpha \qquad (3.12)$$

, where:

Q = torque developed, in [Nm];

F = ram force, in [N];

R = tiller radius, in [m];

α = rudder angle, in [deg].

As may be seen from Figure 3.33, an increasing mechanical advantage is obtained at larger rudder angles with the Rapson-slide type of mechanism. The torque developed by a Rapson-slide arrangement is given by:

$$Q = F'.R' = \frac{F}{\cos\alpha}.\frac{R}{\cos\alpha} = \frac{F.R}{\cos^2\alpha} \qquad (3.13)$$

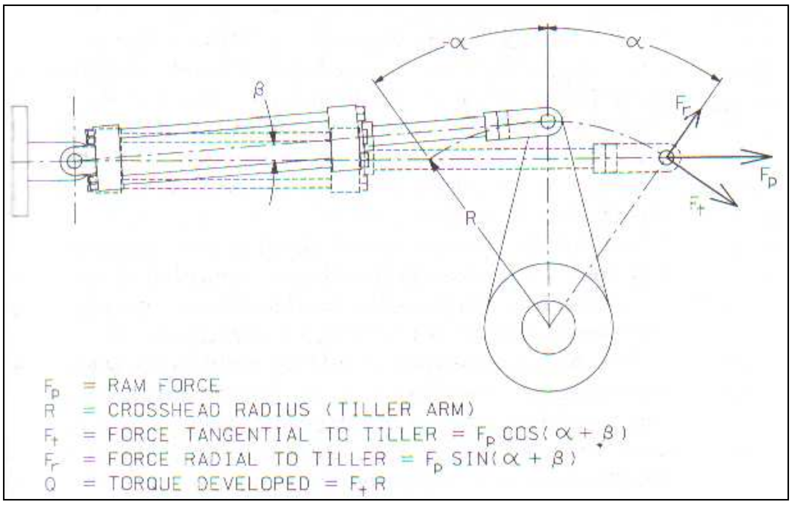


Fig. 3.32 – Force diagram for a link type steering gear.

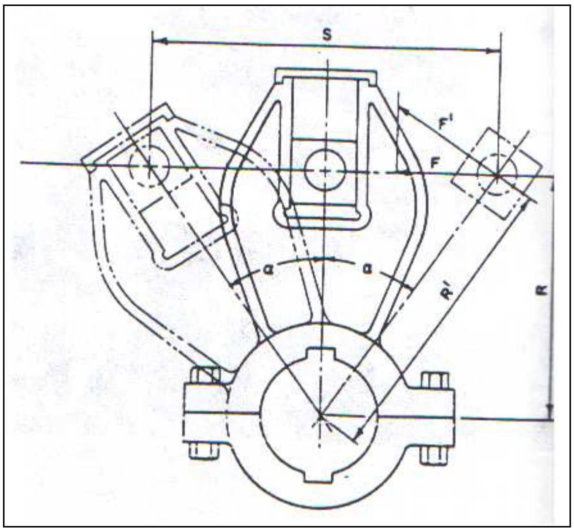


Fig. 3.33 – Force diagram for a Rapson-slide type steering gear.

Hence, for the same ram force and tiller radius, the torque that can be developed by a Rapson-slide arrangement is greater than the capability of a link arrangement by a factor of $1/\cos^3\alpha$. At a rudder angle of 35º, this factor is 55%. In view of the mechanical advantage offered by a Rapson-slide arrangement, it may appear that a link arrangement need not to be considered; however, such is not the case as discussed earlier.

b) Rotary vane type:

With this type of steering gear a vaned rotor is securely fastened onto the rudder stock (see Figures 3.34 and 3.35). The rotor is able to move in a housing which is solidly attached to the ship's structure.

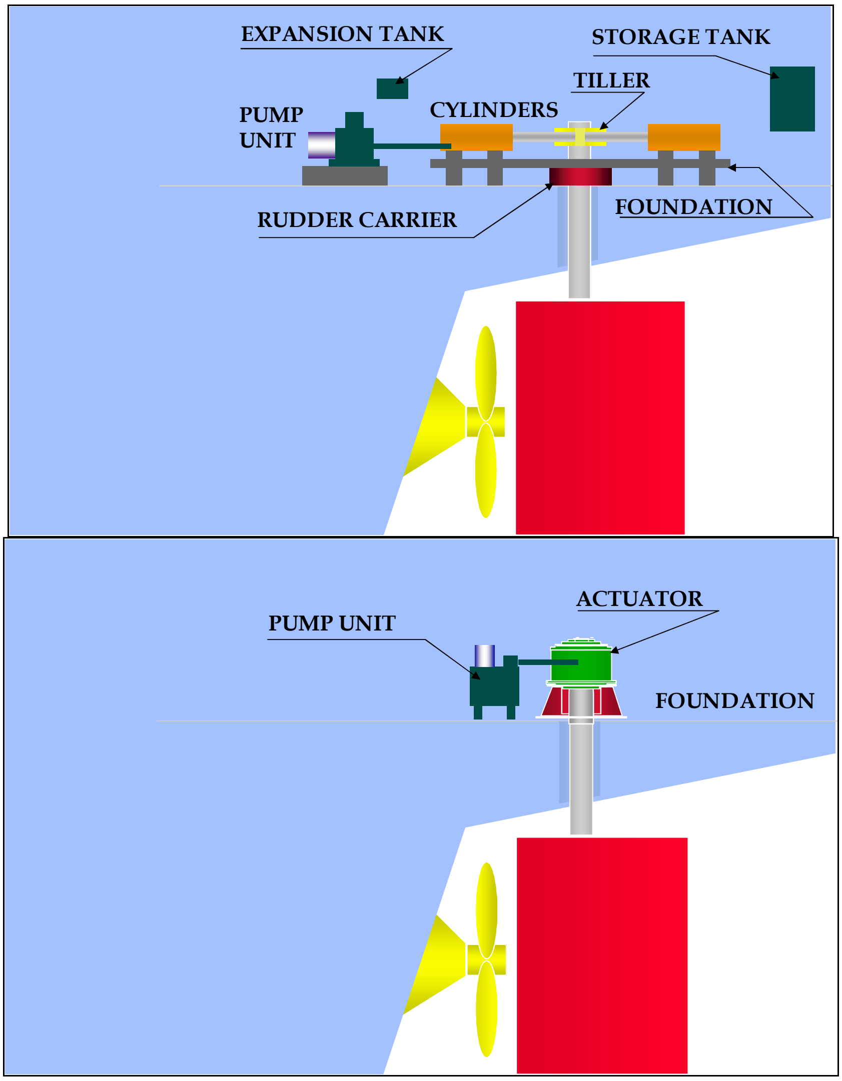


Fig. 3.34 – Schematic view of ram versus rotary vane steering gear.

As shown in Figure 3.35, chambers are formed between the vanes on the rotor and the vanes in the housing. These chambers will vary in size as the rotor moves and can be pressurized since sealing strips are fitted on the moving faces. The chambers either side of the moving vane are connected to separate pipe systems or manifolds. Thus by supplying hydraulic fluid to all the chambers to the left of the moving vane and drawing fluid from all the chambers on the right, the rudder stock can be made to turn anti-clockwise. Clockwise movement will occur if pressure and suction supplies are reversed.

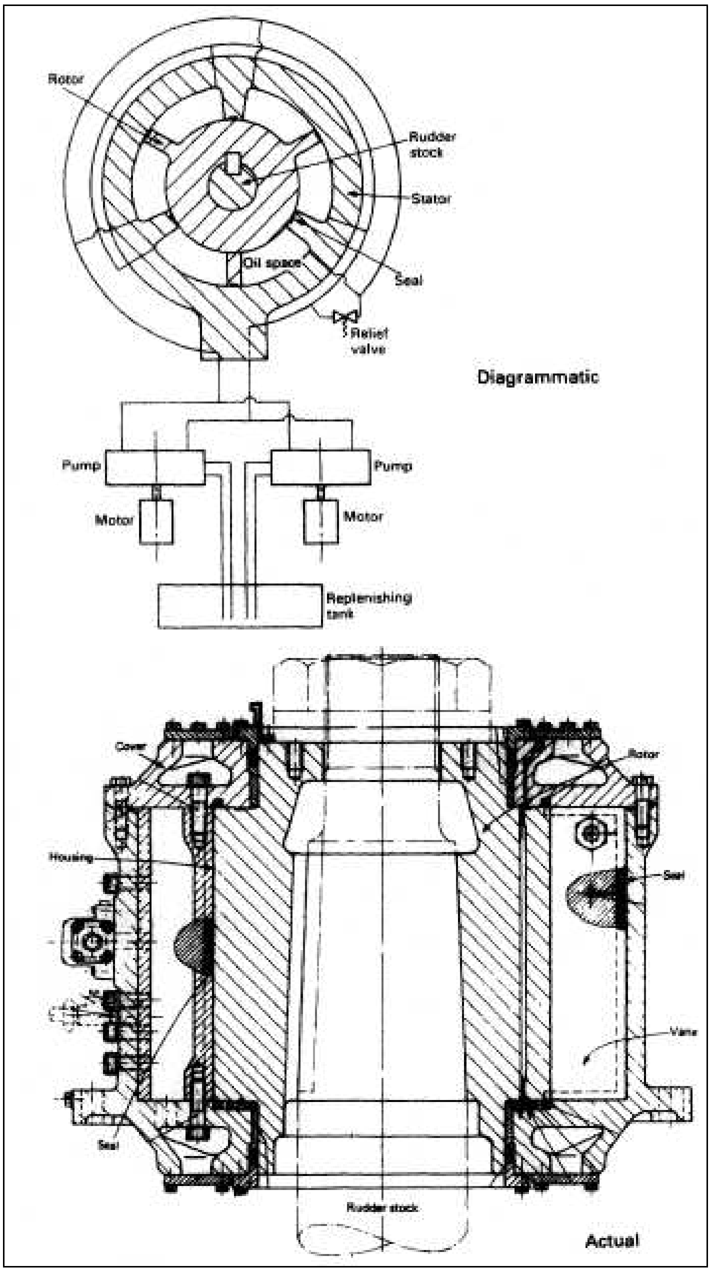


Fig. 3.35 – Rotary vane steering gear.

As shown in Figure 3.36, two or three vanes are usual and permit an angular movement of 70°: the vanes also act as stops limiting rudder movement. The hydraulic fluid is supplied by a variable delivery pump and control will be electrical, as described earlier. A relief valve is fitted in the system to prevent overpressure and allow for shock loading of the rudder.

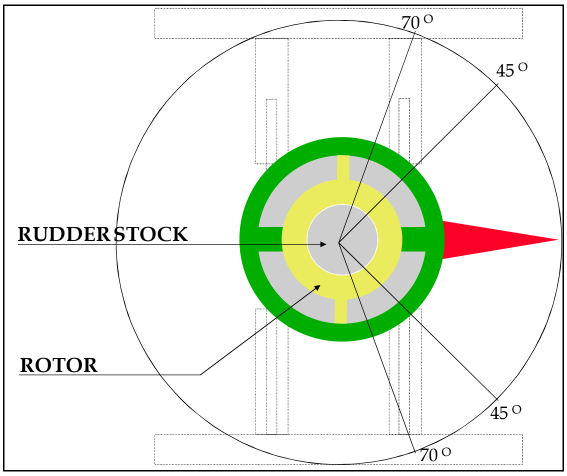


Fig. 3.36 – Ram versus rotary vane steering gear maximum angular amplitudes.

As illustrated on Figure 3.37, a major advantage of this type of steering gear actuator is the compactness (much less space required onboard). On the other hand, a major disadvantage of this type of steering gear actuator is that they do not allow redundancy as in the ram type actuators.

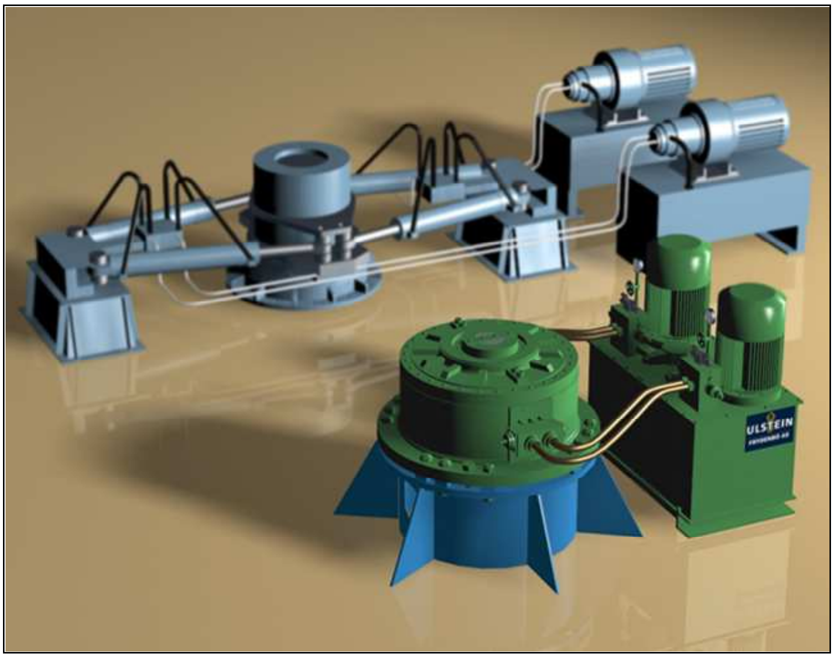


Fig. 3.37 – 3D view of ram versus rotary vane steering gear hydraulic systems.

More recently, Rolls-Royce™ developed spherical steering gears to improve reliability of the rotary vane concept by decreasing loads over the bearing around the rudder stock. Figure 3.38 shows a 3D cutaway of a rotor and actuator of a spherical steering gear. This new concept involves that rotation of rotor eliminates transfer of bending forces from rudder stock onto the actuator, since no edge pressure on bearings will occur due to the spherical design. According to manufacturer, this results in long bearing lifetime.

Fig. 3.38 – 3D cutaway of a spherical steering gear.

3.5.4 Rudder carrier bearing

The traditional rudder carrier bearing (see Figure 3.39) takes the weight of the rudder on a grease lubricated thrust face. The rudder stock is located by the journal beneath, also grease lubricated.

Support for the bearing is provided by framing beneath the steering gear deck. There is thicker deck plating in the area beneath the carrier bearing and the latter may be supported on steel chocks. The base of the carrier bearing is located by the side chocks welded to the deck. The carrier may be of meehanite with gunmetal thrust ring and brush. Carrier bearing components are split as necessary for removal or replacement. Screw down (hand) lubrication may be fitted but automatic lubricators are common. The grease used for lubrication is of water resistant type (calcium soap base with graphite).

The tiller (see Figure 3.39) is keyed to the rudder stock and is forged or cast steel with one (or two for a four ram gear) arms, machined smooth to slide in a swivel block arrangement designed to convert linear movement of the rams to the rotary movement of the tiller arms and rudder stock.

The usual limit for movement of the rudder is 35º each way from the mid position and this is controlled by the telemotor. External rudder stops if fitted, would limit the movement to, say 39º from the mid position. The steering gear itself will also impose a limit on rudder movement but with hydraulic oil loss and heavy weather, there may be severe damage to the gear.

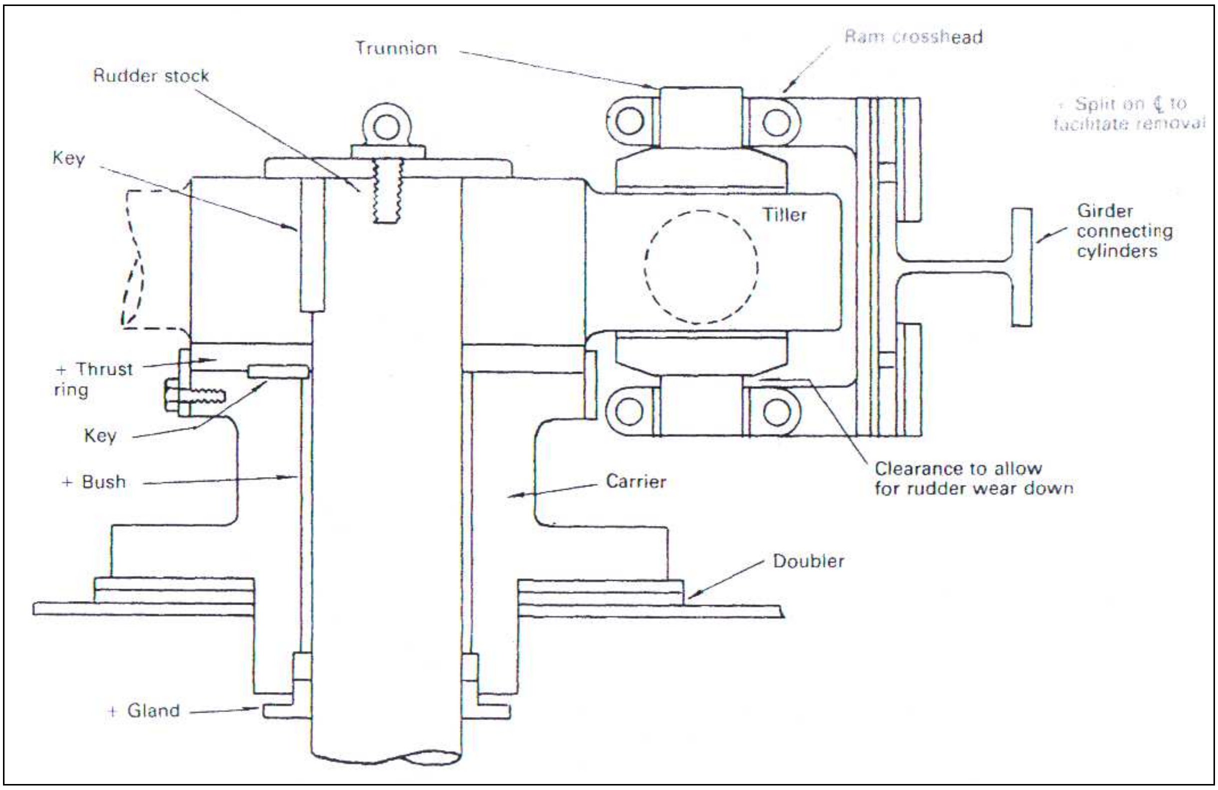


Fig. 3.39 – Rudder carrier bearing.