Hauling device and hydraulic engineering for fishing boats

Complied by

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Forward

At present human resources and labor cost for onboard fishing activities is serious situation. As recently auxiliary fishing machinery are used to or installed on board fishing boats are being necessary for fishing vessels. Therefore, the mounted of fishing machinery onboard is comforted while fishing operation at sea. There are not only saving for labor cost, fishermen can get more profit during the period at sea they are more time remains for fish preservation handling and net or gears repairs.

Onboard fishing machinery is comes very important and quite difficulty maters for handle by fishermen. However, SEAFDEC training department has developed and promoted on sustain technique from simply local for small scale fishing boats to commercial fishery in the Southeast Asian region from previous as presently.

Thereby, the installation and utilize of an auxiliary fishing machinery and its application of onboard in combine with simply or appropriate machine or equipment should be impart and conveys to fishermen and technician concerned in Southeast Asian Countries in order to handle of fishery machinery on board with safe and capable to handle of maintenances and trouble shooting. In addition to reduce of labor cost and number of personal on board on together with reliable used with good performance at sea and no pollution impacts to the nature and fishing ground.

Dr. Siri Ekmaharaj
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Manpower Source

1. Introduction

A hand hauling device may be regarded as any hand operated aid fitted to a fishing boat which helps or assists the operator to make easier the setting or hauling of fishing gear. These devices may be divided into two groups:

- Any form of roller, drum or sheave used to assist in reducing friction as the fishing gear passes over the side of the boat. When the gear is being set or retrieved, the friction of the moving gear over the roller drum of sheave causes rotation of the device. This is a non-mechanical operation.

- Any machine enabling the fisherman to increase his hauling power by mechanical means which as a result of his effort causes the fishing gear to be moved by a reel, drum of sheave is called a hauler.

2. The simple devices

2.1. Roller

A simple roller to reduce the friction when handling fishing nets or ropes can be used in almost any type of fishing boat, depending on the size and stability of the craft. The aid can be in the form of a wide or narrow roller to suit the fishing gear, and may be fitted to a boat either at the bow, mid-ships or stern, or at a height to suit the boat. As well as helping to reduce friction, it also changes the direction of the effort needed to lift the gear over the side of the boat.

On a craft of suitable stability the roller may be mounted on a bracket, so that the person hauling pulls towards him rather than lifting upwards. The weight of his body is used to haul, rather than just the strength of his arms. See Fig. 1.3.

The diameter of the roller should be as large as practical to reduce friction and make it easier for the person pulling.

Fig. 1.3 pulling style without and with roller
2.2. Simple hauler

To make a simple hauler, all that is necessary is to fit a handle to a roller. The fisherman is able to pull the fishing gear aboard by turning the handle instead of using his hands to pull on the nets, line or rope.

The effort needed to haul his fishing gear can be made less by making the shank or the handle longer – the longer the shank the greater the lever effect and less effort will be required to haul in the gear.

In practice, the hauler can be bolted to an upright post or pedestal, and fitted to the bow, stern or to the side as required.

![Pedestal and side mounted hauler](image)

**Fig. 1.5 Pedestal and side mounted hauler**

2.3 Hand reel

A reel is a device on which fishing lines are wound and stored. The design and dimension of the reel may vary according to the type of fishing, the fishing depth, etc. The reel may be attached to a hand-held fishing rod or mounted directly to the boat sides, or stern. An example developed in Norway it has been very successful when fishing for cod. See **Fig. 1.6**.

![Hand reel](image)

**Fig. 1.6 Hand reel**
The reel can be manufactured from wood, aluminum, galvanized or stainless steel and secured to a tube clamped to or set into the gunwale of the boat. The center bolt through the boss is used as a brake when paying out the line, and for clamping the reel when hauling in the trace.

2.4. Gurdy

A gurdy as used in part of Southwest England for mackerel fishing is a simple hand hauled reel for pulling in and storing mackerel long lines. It is made of a welded mild steel reel mounted on a short pedestal, which is clamped to the gunwale of the boat. The handle can be fitted to either side of the reel for left or right handed operation.

![Fig. 1.8 Two gurdy on one boat](image)

2.5. Hauler with hand rim

An alternative to a handle is to make a hauler with a larger diameter rim, which can be easily turned, hand over hand.

![Fig. 1.9 Hauler with hand rim](image)
2.6 Rectangular frame reel

A fishing reel does not have to be circular in shape, but may be made in rectangular form, with a central point and handle.

The pivot is fitted to a pedestal as shown in Fig 1.10.

![Fig. 1.10 Rectangular hand reel](image)

2.7. Pole or davit mounted hand reel

A hand reel may be fitted to a pole or davit and mounted on the side of a fishing boat so that the fishing line is haul clear of the boat sides by a pulley block fitted to the extended arms, as shown in Fig. 1.11.

![Fig. 1.11 Pole or davit mounted hand reel](image)
2.8. Metal reel and davit for deep water fishing

An example of a locally is made metal fishing reel and davit are shown in Fig. 1.14 and 1.15.

![Fig. 1.14 Detail of metal hand reel](image1)

![Fig. 1.15 Metal hand reel davit](image2)

2.9. Hand squid jigging

Fishing for squid is carried out in various parts of the world; notably Japan, using a method called “Jigging”.

Hooked lines being “jerked” so that the hooks tear into the body of the animal rather than the mouth, as with most other fish caught by hook catch squid. This jerking a typical unit is shown in Fig. 1.16.

![Fig. 1.16 Hand jigging reel system](image3)
2.10. Sheave

A solid hub of hardwood or cast aluminum, with a deep V-shaped groove is machined so that it will grip the line, the groove being made to suit the thickness of the line.

The shape of the sheave is important, and the dimensions have to be matched to the size of line to be hauled. One method is to make the sheave up of two discs, bolted together by sandwiching a distance piece between them. Various thickness of distance pieces may be added or taken away to suit lines of different sizes. The sheave is made to turn from between 50-150 rpm as necessary to give the required hauling speed in meters/min (or ft/min).

The diameter of the line to be hauled may range from 6 mm to 20 mm, depending on the type of fishing.

The line is prevented from wrapping around the sheave by an “ejector-knife” which fits closely into the groove to lead the line off the sheave, and allow it to coil down on the deck. The ejector-knife is often made from bronze to prevent wear of the sheave.

Fig. 3.14 Line hauler sheave – two disc type

To improve the grip on the line a guide roller may be fitted to increase amount of contact of the line with the sheave. A second roller may also be added to further improve the grip, and to ensure that the line leaves the sheave without wrapping around it. See Fig. 1.20 and 1.21.

Fig. 1.20 Sheave with guide roller Fig. 1.21 Sheave with two rollers
2.11. Sheave with pressure roller

It is possible to obtain a positive grip on the line by fitting a covered rubber roller as shown in Fig. 1.22. The roller is pressed on to the sheave, by mounting the rubber roller on an ‘over-center’ lever, which exerts pressure on the line in the sheave. The profile of the covered rubber roller must match the sheave, and fit as closely as possible to prevent the line jumping between them.

![Fig. 1.22 Sheave with pressure roller](image)

2.12. Net drum

Fishing nets may be hauled in and stored on a hand operated drum, which can be fitted to haul over the bow, sides or stern of the boat as required.

The fish have to be taken out of the net before it is reeled on the drum, so there must be enough space to do this between the point where the net comes over the boat sides and the drum.
2.13. Simple winch

The unit may be made of mild steel bar fitted with a slightly larger diameter pipe or hollow wooden handle, e.g. bamboo. The bar is bent into a crank supported by simple bearings on each side. A drum of drums to wind the rope may be fitted inboard or outboard of the bearings, as prepared.

The unit may be bolted or screwed to the boat sides or made to drop into existing holes for rowlocks, if provided.

The rope may be hauled by passing a few turns around the drum, used as a capstan as in Fig. 1.26 or wound on the drum and stored as shown in Fig. 1.27 and 1.28.
The method of operation is to use a light rope fastened to a large anchor at one end, and to the hand winch at the other.

By dropping the anchor and paying out all the line, the boat, together with a small oyster dredge is then hand-wound or cranked up to the anchor. In this way the dredge collects the oysters or shells until the boat are winched up to the anchor. It is then lifted, reset, and the cycle starts again.

A simple deck mounted hand winch for light loads may be made, as shown in Fig. 1.32. This consists of two end brackets fastened to the deck and which support the shaft carrying the rope drum. A handle at each end of the shaft is used to turn the winch and a chain or rope strop fastened at one end to the winch is looped over the handle to stop the drum turning.
2.15. Methods of transmission power

In order to handle more fishing gear or to fish at a greater depth a hand hauler with gearing is required to be able to haul the increased weight. By using a hand hauler designed to use a system of pulleys and belt, sprockets and chain or gears it is possible to increase the effective hand power of the fisherman when hauling.

It is necessary to understand the basic principles involved and to be familiar with the various components of any machine which will give a “mechanical advantage”.

1) Pulleys and Belt If two equal size pulleys are connected by a belt, and one pulley is turned by hand the other pulley will turn at the same speed.

![Fig. 1.34 Equal size pulleys](image)

If the pulleys are different size, connected by a belt and the large pulley A is turned, the smaller pulley B will turn faster. See Fig. 1.35.

![Fig. 1.35 Different size pulleys](image)

Conversely, if pulley B is turned, pulley A will run much more slowly. The object of using pulleys of different sizes is to improve the mechanical advantage of the hauler, to make it easier for the man hauling in the net or rope.
2) **Sprockets and chain** Exactly the same principle applies when using two sprockets and a chain as with two pulleys and a belt.

A large sprocket with 40 teeth connected by chain to a sprocket with 20 teeth will give a ratio of 2:1.

![Fig. 1.36 Reduction using chain and sprockets](image)

3) **Spur and helical gears** The simplest type of gear is called a SPUR gear, which means that the teeth are cut parallel to the shaft on which it fits. Simple spur gears as fitted to hand turned machinery are often heavy, and made from cast iron machined to produce the teeth. When two or more gears are to be meshed together, the teeth on the gears must have the same shape and be the same distance apart. If not they will not mesh properly and cannot be made to work.

![Fig. 1.37 Spur gears](image)  ![Fig. 1.38 Helical gears](image)

Helical gears have teeth cut at an angle to the shaft on which it fits, and the teeth are cut in a curved shape. They run more smoothly than spur gears are more expensive and difficult to make. Helical gears and spur gears cannot be mixed or meshed together.

![Fig. 1.39 Simple reduction gear](image)
In Fig. 1.39 (A) if a handle is fitted to the small gear wheel, the larger gear wheel turns at \( \frac{1}{4} \) speed, the gear ratio is 4:1, and the hauler can pull four times the weight for the same effort. In (B), the ratio is only 3:1 so the hauler can pull three times the weight for the same effort.

### 4) Gear trains

Extra reduction can be made possibly by the use of another gear in the train. See Fig. 1.40.

![Fig. 1.40 Gear train](image)

The ratio between gears A and B = 48:16, or 3:1
The ratio between gears B and C = 16:8, or 2:1
The overall ratio between A and C is 3*2 = 6:1

If gear C is turned 6 times, gear B will turn 3 times and gear A will turn one revolution (once).

Note that gears A and C turn in the same direction.

The most common gear train design is using 4 gears, but with 2 of them on the same shaft, so that they turn together (at the same speed) as in Fig. 1.41.

![Fig. 1.41 Gear train with handle](image)

The ratio between A and B = 4:1
The ratio between C and D = 4:1
The overall ratio = 4*4=16:1

Note: B and C turn at the same speed. A and D turn in the same direction.
In same case, a choice of cranking speed can be provided by fitting the handle on the shaft carrying gears B and C. The handle will then have to be turned in the opposite direction. A second handle can be fitted on the other side of the winch if the shaft through gear A is long enough.

![Diagram of winch with reduction gear train](image)

**Fig. 1.42 Principle of winch with reduction gear train**

By applying the theory of gearing, as it is possible to design a more efficient hand hauler, preferably on which can be built from parts available from local sources.

As the hauler or winch is developed and improved from basic designs, it becomes necessary to provide the operator with additional controls to ensure that the hauler can be used effectively and with safety.

![Diagram of geared hand hauler](image)

**Fig. 1.43 Geared hand hauler**

One possible source for finding the reduction gears is to use oil engine parts. On many engines there is either a set of meshing gears, of a set of chain wheels and chain to drive the camshaft from the crankshaft. These are called the TIMING GEARS or TIMING SPROCKETS AND CHAIN.
Fig. 1.44 Engine showed timing chain drive

If the chain drive type is to be used, the chain should run inside some kind of cover so that it is kept oiled and therefore does not rust (especially when fishing in sea water). This principle is shown in Fig. 1.45.

Fig. 1.45 Hand hauler using timing chain and sprockets

5) Pole or davit mounted hand reel
A development of the simple hand reel is shown in Fig. 1.46 so that the fishing line can be retrieved very quickly. The gear ratio between the two pulleys is designed so that the reel containing the line turns much faster than the pulley lifted with the handle. The drive from one pulley to the other is by belt.
In this example there is very little weight or load on the fishing line, so the gear ratio provides a ‘step’ as opposed to a step-down or reduction which is normally the case.

6) Hand winch with reduction gear
The winch consists of a framework of mild steel angle welded or bolted to a base. The winch drum is mounted on a large diameter shaft carried on a pair of bearings, one on each side.

The drum may be of wood or mild steel tubing or fabricated by bending mild steel into a cylinder and welding the seam. Flanges of mild steel are welded at each end of the tube, with the drum shaft passing through the center.

A large diameter gear is then bolted to one flange of the drum and concentric with it. A shaft with a bearing at each end and carrying a small diameter gear to mesh with the gear on the drum is bolted to the framework.

A reduction of from between 10:1 and 40:1 ratio, depending on the gears selected or available may be achieved with two gear as shown in Fig. 1.47.

7) Winch brake
A brake is necessary to control the speed of paying out of the warp if the load is very heavy. Very high speed paying out can be dangerous and it should be controlled. Operator is easy to control it by using the lever acting on the drum, particularly if there is a ‘flywheel’ to the drum, or large flat surface on which a brake can act. See Fig. 1.49.

Another type of brake used on larger winches utilizes a brake band around the smooth surface of the flywheel. When the brake lever is applied (or a wheel
turned) the brake band clamps on to the flywheel. See Fig. 1.50. A brake lever may be hand or foot operated.

**Fig. 1.50 Brake band acting on drum**

8) **Winch ratchet assembly**
A winch ratchet is necessary when hauling in heavy loads. Without it, the weight being pulled may overcome the fisherman as he becomes tired. This device allows the fisherman to rest temporarily, or to re-adjust his position ready for another haul, while the ratchet assembly takes the weight.

The ratchet assembly consists of two parts: the pawl, which is fitted to the frame of the winch, and the ratchet wheel, which is fitted to the drum. As the drum of the winch is being turned, the pawl fitted above the ratchet wheel lifts and drops by its own weight on to the ratchet. When drum stop turned, the square shoulder on the cam or ratchet wheel butts against the pawl, to preventing drum from turning backwards due to weight being hauled. See Fig. 1.51.

**Fig. 1.51 Pawl and ratchet assembly** **Fig. 1.52 Pawl and ratchet assembly**
9) Spooling device

If the drum of the winch has to carry a large capacity of rope, a method of spooling the rope evenly on the drum should be designed and made. A simple way of doing this is to provide a long bar, pivoted on the base of the winch in the center of the drum. The bar has a yoke fitted above the drum center line, the length of the yoke should be at least the depth of the drum when fully wound with rope, when the bar pivots at the maximum width of the drum.

The rope is fed through the yoke, and as the drum winds on the warp a crewman moves the bar left or right to keep the warp spooling on to the drum neatly and evenly.

Fig. 1.53 Hand winch with spooling device
IC Engine Power Source

1. Introduction

The use of hand hauling devices is a very useful and often necessary ‘first step’ in the improvement of traditional fishing methods.

However, in many cases the development can be carried further by using an engine to drive a hauling device, instead of using manpower. This opens up a much wider range of fishing methods and gear available to the fisherman so that he can then consider the use of more nets, longer lines, even a different type of fishing operation as a result of having a powered hauler.

One of the disadvantages of an engine driven hauling device is the increased cost involved in buying the equipment, particularly if it is made in another country and has to be paid for in foreign currency. In order to get around the problems of high cost and any exchange controls involving foreign currency, it may be possible to produce the equipment locally.

It is necessary therefore to consider various basic ideas of hauler designs and their components, so that a suitable machine can be made. Powered haulers can be divided into two main types:-

a) Driven from the main engine of the fishing boat by a power-take-off system (P.T.O. drive); The advantage of the hauler driven by the main engine is that there is no need to buy and maintain another engine. The drive arrangement is taken of the engine in such a way that it can be ‘clutched’ in or out of drive by the operator, leaving the engine free to drive the boat in the normal way independently of the hauler.

With this type of drive the installation may be more complicated and the hauler can only be used in one (fixed) position in the vessel. There are some limitations with its use, as the engine drives either the hauler or the propeller shaft or both at once. The operator does not have the choice of independent speed variations; for example, he cannot make the propeller shaft turn slowly and at the same time speed the hauler up to run fast, or vice versa. If the engine runs fast, the hauler and the propeller shaft (when in gear) will also run fast.

b) Driven by separate engine completely independent of the main engine of the fishing boat.

The advantages of this type of hauler are that:-

1) The operator can have total control over the speed at which the fishing gear is being hauled, independent of the main engine and therefore the speed of the boat;

2) The engine and hauler can be in a unit, and moved to various places on the boat to suit the fishing operation.
The disadvantage is that there is another engine to buy, start and maintain. A hauler engine is usually fairly small, and often small engines are not so robust, can be temperamental to start and are exposed (on deck) to corrosion and damage.

2. Power take-off (P.T.O)

Power take-off (P.T.O) is the method by which power to drive auxiliary equipment is taken partly from the engine used for main propulsion purposes. In a marine application, the engine is installed primarily to propel the vessel by driving the propeller shaft and propeller, any other power required for auxiliaries, winch, net hauler, etc., may be derived from the engine by means of a P.T.O.

In practice, power is taken off an engine in the following ways:-
1) The front of crankshaft
2) The extended camshaft (at half engine crankshaft speed)
3) The flywheel
4) The power take-off points with clutch (fitted to the engine before deliver)

1) Crankshaft

On many engines the crankshaft is extended forward outside the engine casing to enable a drive to be take of. The crankshaft is usually provided with a parallel key way, so that a V-pulley, flat belt pulley or clutch unit can be attached and driven. See Fig. 2.1.

![Fig. 2.1 Crankshaft extension](image)

2) Camshaft

Many smaller engines have a long extension camshaft outside the engine casing which is keyed in the same manner as the crankshaft extension. It is important to remember that the camshaft turns at half engine speed. For example, on an engine that runs at 2000 rpm. the camshaft would turn at 1000 rpm. On some engines the camshaft is gear driven and therefore turns in the opposite direction of rotation (D.O.R.) to the crankshaft.

![Fig. 2.2 Camshaft extension](image)
3) Flywheel

On some models the engine flywheel is not encased within the engine, and provision is made for a power take-off point on the flywheel itself. Holes or tapped (screwed bolt holes are provided, so that an extra pulley can be easily fixed. See Fig. 2.3 A. On some small engines, the perimeter of the flywheel may be used to drive a flat or V-belt. See Fig. 2.3 B.

![Fig. 2.3 Flywheel pulley](image)

4) Power take-off points with clutch

Many engines are now built to include a power take-off point complete with clutch unit. The units are designed and built either by the engine manufacturers, or by specialist clutch manufacturers for a particular engine and therefore matched to it. Examples of engines complete with clutch unit either on the front or rear of the engine are shown in Fig. 2.4 and 2.5.

![Fig. 2.4 Engines fitted with front end P.T.O. clutches](image)  ![Fig. 2.5 Engine fitted with P.T.O. clutch on gear](image)

Some clutch units are built to incorporate a reduction gear, so it is important to know the gear ratio between the engine and P.T.O. output shaft. If is also necessary to know the Direction Of Rotation (D.O.R) of the P.T.O. shaft when compared with D.O.R. of the engine.

5) Power take-off limitations

It is important to note that engine and clutch manufacturers make very clear recommendations concerning how many horsepower is taken off the engine at the P.T.O. or clutch unit. In addition, there are often limits to the amount of side
loads, which may be exerted on the engine crankshaft, crankshaft and camshaft extensions, P.T.O. clutches, etc.

When a flat or V-belt pulley is fitted to the engine and belts are tensioned in order to drive the hauling equipment. The side thrusts may be so considerable that severe damage to the crankshaft, camshaft or P.T.O. clutch bearings and oil seals may result unless the installation is correctly designed for the expected loading.

The method of preventing damage by side loads on the engine unit is to install a short P.T.O. lay-shaft driven directly through a flexible coupling, with the lay-shaft mounted on heavy duty bearings which are capable of taking the excessive load caused by belt tensioning and transmission of power.

6) Planning for power take-off drives

When planning the type and layout of the power take-off train the following points have to be taken into consideration:-

- The engine speed and required winch driving shaft speed (or drum speed) must be known so that the correct reduction gears and pulleys may be calculated and fitted;
- The power required from the engine and the arrangement of the P.T.O. and whether a short P.T.O. lay-shaft is required;
- The required type of drive, whether totally by flat belt, V-belt or chain drive or by a mixture of these types. This decision is dependent upon the availability of local materials, or which type best suits the application and cost;
- It is preferable to design a system where there is as little machinery as possible turning when the P.T.O. is not in use. Avoid putting the only clutch after the belt drive to the overhead lay shaft’ it is better to have two clutches, one to drive the lay-shaft, and one to drive the deck machinery;
- The position of the winch or deck machinery. This subject is dealt with under “Winch Positions” later;
- Plan to locate shafting and belt drives so that they leave as much usable space on board as possible, while at the same time allowing accessibility for servicing and repair;
- When the engine has to be aligned to the propeller shaft the movement required will affect the alignment of the P.T.O. belt and overhead lay-shaft. If is important that belt pulleys on the engine P.T.O. and the lay-shaft are in line and parallel. If not, a flat belt will not stay on the pulleys, and Vee-belts, chains and sprockets will wear quickly;
- Provide adjustment for all belts. This may be done by a jockey clutch pulley, by a belt tension pulley only, or by slotted holes to allow movement;
- There should be a universal or flexible joint between the overhead lay-shaft arrangement and winch. This should take up any slight misalignment between the two units in use. An alternative to a flexible joint is a belt or chain drive from the lay-shaft to the winch/deck machinery;
- Provide adjustment for all belts. This may be done by a jockey clutch pulley, by a belt tension pulley only, or by slotted holes to allow movement;
- The controlling lever for the drive clutch to the winch/deck machinery must be close to the other controls, so that the operator can disengage the drive quickly in an emergency;
- All shafting, belts and pulleys should be fitted with guards to prevent the crew from becoming entangled in the machinery when it is running.

a) drive to winch by overhead lay-shaft, clutch control in wheelhouse

b) Direct drive to winch, clutch and controls on deck

c) Clutch forward of engine, clutch control on deck

3. Flat belt drive

A flat belt may be used to transmit power from an engine to a hauling device. It is a very practical and inexpensive method of drive but does have some limitations in use.

The flat belt drive installation is most suitable in the following situations:-
- When the distance between the engine and hauler pulleys is greater than about 1.20 m (4 feet)
- When the reduction between the engine and hauler pulleys is small, about 1:2. Large reductions may be made only if the distance between pulley centers is long, more than 2.40 m (8 feet);
- When a degree of belt slip is acceptable;
- When a jockey pulley clutch is required.

Advantages:-

Fig. 2.6 Various PTO/Winch drives
- Not suitable when there is a short distance between engine and hauler pulleys, as it is then difficult to get sufficient tension on the drive belt to obtain a good grip. The weakest point of the belt when being tensioned is usually at the join;
- A flat belt drive cannot be used in an oily situation;
- Alignment must be very accurate otherwise the belt will run off the pulleys.

![Fig. 2.7 Jockey pulley – flat belt drive](image)

**Installation of flat belts**

When designing an installation it is most important to ensure correct alignment of pulleys, which must be parallel. Provision must be made for adjustment of belts. If a jockey clutch is being used, the same jockey pulley arrangement is used both for the clutching operation and belt tension. All belts must be fitted with belt guards to prevent clothing and hands being caught in revolving shafts.

The choice of the correct size pulleys and belt width must be made to transmit the power available at the correct speed for the hauling operation. **Fig. 2.7** shows a typical flat belt drive installation.

**Adjustment of flat belts**

In any installation using driving belts there must be provision made for adjustment as all belts stretch in service. Initially, a flat belt may stretch considerably in which case it is practical to shorten it by cutting off the surplus and fitting a new belt fastener to one end. Once the belt has ‘run-in’ it will require adjustment less frequently.

In the case of a flat belt drive to a small unit, for example, a generator, which can be moved slightly without affecting other installed equipment, belt adjustment may be achieved by slotted holes in the base of the unit permitting some movement to tighten the belt. See **Fig. 2.11**.
In a typical winch installation however, this in normally not practical, as the engine is in a fixed position relative to the propeller shaft, and the winch is securely mounted on deck. Belt adjustment is most often achieved by the use of a jockey or idler pulley acting on the SLACK side of the belt.

The jockey pulley is more effective if fitted as close as practical to the driving pulley to assist the belt to ‘wrap around’ it and so providing better grip. The jockey is fitted to the outside of the belt as shown in Fig. 2.12 (b).

If the belt is adjusted by fitting the jockey on the inside of the belt the drive is not so positive. The jockey pulley lifts the belt away from the engine driving pulley and reduces the ‘area of contact’ of the belt with the pulley which leads to belt slip. See Fig. 2.12 (a) and (c).

4. V-belt drive

The V-belt drive system has been developed to provide an alternative to the flat belt system and may be used to advantage in the following types of installation:-
- Where the distance between pulley centers is shorter than is possible with a flat belt drive;
- Where a greater reduction ratio between the engine and hauler is required. This is possible provided there is sufficient distance between centers. If there is a very large difference in pulley sizes on a short belt run the belt will not grip
the small pulley sufficiently, as there is not enough belt in contact with the pulley;
- Where greater power is required and can be transmitted by using matched sets of multiple belts;

V-belts are more tolerant of slight misalignment although this can cause excessive belt wear.

Disadvantages:
- V-belts are more difficult to replace if either pulley is fitted between components which have to be removed before the belts can be slipped over the shaft. This can be a long operation which may result in fishing gear in fishing gear or catch being lost if a breakdown occurs whilst hauling;
- Belts in a multiple belt drive have to be replaced in complete sets if one becomes broken or damaged, which can prove to be expensive;
- A jockey pulley clutch cannot be used on V-belts except in the case of a single belt drive off a small engine.

Installation

Correct alignment is very important with V-belts. Shafts must be parallel and pulleys exactly in line otherwise the shanks of the belt will wear quickly. Provision must be made for adjustment to prevent the belt slipping and overheating. A jockey pulley is the usual method of belt adjustment in a hauler installation, but V-belts cannot be fitted with a jockey pulley clutch. Some other clutching device must be fitted, either at the engine or at the hauler itself. All belts and pulleys must be guarded as with flat belts. The careful choice of V-belt pulleys, numbers of V-belts in the drive and the section of belts to be used is advised to ensure the available power and speeds may be transmitted without belt slip. Fig. 2.13 shows a typical drive arrangement, but with out jockey pulley adjustment.

Vee belts are made in various sizes and widths, and numbered according ot the cross-section of the belt as shown in Fig. 2.14.
Recent changes in the measurement of V-belts have been introduced. Imperial sizes in inches are no longer made, all new classification being in metric sizes. As there are still many Imperial sizes in stocks around the world, they have been included for purposes of recognition.

Belts are measured by inside length and usually have the cross-section size and length stamped or printed on them, for example:-

A 3490 (millimeters) or A 136 (inches)

The above example shows that the belt has an A cross-section, and that it is 3490 mm long (136\") when measured from the inside.

The method for working out the length of a V-belt is as follows:-

a) take a length of soft wire or string (that does not stretch);
b) lead it over the pulleys (in the grooves) and mark the wire or string as shown below;
c) Measure the total length of the wire or string to the mark.

Note: Any adjustment pulleys or slotted adjustment bolts must be loosened, so that the SHORTEST belt length is being measured.
Standard lengths of V-belts:

When planning the layout of an installation, it is possible, by being aware of the standard belt sizes, to place the components to suit the length of standard belts.

This improves the spare parts availability for the future. A list of standard length V-belts is given in Table 1.

<table>
<thead>
<tr>
<th>Table 1 Standard length of v-belt</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Adjustment of V-belt drives</strong></td>
</tr>
</tbody>
</table>

Similar principles apply concerning the adjustment of V-belts as with flat belts. Adjustment may be achieved by moving one unit, usually the driven unit, by slotted holes in the base.

On a winch drive installation adjustment by jockey pulley is more common by fitting the jockey to act on the INSIDE of the belt. However, it is quite acceptable to adjust the belt by a jockey on the OUTSIDE of the belt, but by using a different type of pulley.
Inside adjustment: A V-pulley is fitted as near to the LARGER pulley as practical, acting on the inside of the belts. See Fig. 2.18 (a).

a) Inside adjustment Jockey as near as practical to larger pulley

b) Outside adjustment Jockey within 1/3 of total distance between centers, near to engine pulley

Fig. 2.18 Methods of rotation

Outside adjustment: A Flat pulley is fitted within 1/3 of the total distance between pulley centers from the DRIVING (engine) pulley acting on the outside or back of the belts. Excessive pressure on the back of the belt causes fatigue and overheating leading to early wear. This is because the belt has to flex first in one direction around the V-pulleys and then in the other direction around the flat pulley. This is called DEFLECTION. See Fig. 2.18 (a) and (b).

Adjustment of V-belts in a multiple belt drive is quite difficult, as it is necessary to ensure that there is even tension acting on all belts. Uneven tension is caused by the jockey pulley being out-of-parallel with the belts and pulley and results in the belts on one side being stretched to much. Eventually the stretched belt will not drive and the complete set of belts must be changed. Similar problems occur if there is major misalignment of any of the shafts or pulleys, as the belts ‘scuff’ and overheat.

It is important to note that with multiple belts the correct number of belts should always be used. In the event that one-belt breaks the strain placed on the remaining belts in the drive will cause them to fail very early. Never add one new belt to a set of used, and therefore partly worn, belts.

5. Timing belt drives

A more recent concept in driving machinery, the timing belt drive design uses a flat rubber-type belt with the underside having flat square ‘teeth’ or notches. The pulleys over which the belt runs also have notches into which the belt fits. It is a very positive like chain, but being slightly elastic it is very smooth and quiet and needs not lubrication. Another good feature is that once the belt has been adjusted on installation, no other adjustment is necessary. Depending on shaft
speed of the machinery concerned, power drives up to 100 horsepower are possible with the correct timing belt.

6. Chain drive

In some applications it may be considered preferable to drive deck machinery by chain rather than by flat or V-belts.

The main advantage is that a chain drive is very positive, and there is no slip between chain and chain sprockets. Positive drive is achieved without the tension, which is necessary on a flat or V-belt drive installation to ensure that no slip takes place.

Chain drive is particularly suitable for slow speed power transmission and between engine and hauler or winch when close together. It is not suitable for very long lengths of drive, or for very high-speed applications, because of the difficulty of guarding the chain to prevent accident to the fishermen.

Corrosion by salt water is a major factor to consider when planning an installation using chain in an exposed position. Ideally, all chain drives should run in an oil bath for correct lubrication and protection, but often this is not practical.

Chains in exposed conditions should be heavily grease for lubrication and protection against corrosion, and inspected frequently to ensure that rollers and links have not seized up with rust. A chain drive lying idle for long periods in exposed conditions will rust very badly and often break when next used. To prevent this, the chain can be removed beforehand, and left soaking in oil while the boat is not being used.

Installation of chain drive

Chain drives can be used easily in practically any position, with the exception of a vertical drive, which can be more difficult to install.

Alignment is again very important, all shafts and sprockets should be parallel and in line. The slack side of the chain can be very dangerous to the crew if it is allowed to become to slack when running fast. Fitting idler sprockets on the slack side of the chain, either on the inside or outside as preferred may tension the chain.

A clutch of the jockey type cannot be used with chain, therefore some other clutching device must be used either at the engine or hauler.

As with flat and V-belt drives all chains should be guarded. Any chain drive fitted with an oil bath will not need further guarding. The choice of correct size of chain and sprockets must be very carefully made to ensure that the drive can be transmitted. Fig. 2.22 shows a simple chain drive installation.
Adjustment of chain

Principle of adjustment is very similar to those of V-belts. Adjustment may be achieved by moving the driven unit by slotted holes in the base, as shown in Fig. 2.25 (a). An alternative method by the use of idler sprockets is more normally used, idler sprockets being fitted to the slack side of the chain. Typical positions for various designs of idler are shown in Fig. 2.25 (b). A chain idler may be fitted with a spring to maintain an even pressure on the chain and reduce chain whip.

Chain tension should be fairly tight when installed, with only a small amount of slack. New chains will loosen slightly and once the initial stretch has occurred there will be much less adjustment required. Chain tension is checked as shown in Fig. 2.26 and 2.27.
Adjustment guide

An approximate guide to the amount of chain slack allowed is given in Table 6. The method of measuring slack is shown in Fig. 2.27.

### Table 6 slack standard of chain

<table>
<thead>
<tr>
<th>Distance between centers (mm)</th>
<th>508</th>
<th>762</th>
<th>1016</th>
<th>1270</th>
<th>1524</th>
<th>1778</th>
<th>2032</th>
<th>2285</th>
<th>2540</th>
<th>3175</th>
<th>3302</th>
</tr>
</thead>
<tbody>
<tr>
<td>Chain slack (mm)</td>
<td>12.7</td>
<td>15.8</td>
<td>22.2</td>
<td>25.8</td>
<td>31.7</td>
<td>38.1</td>
<td>41.2</td>
<td>47.6</td>
<td>50.7</td>
<td>63.5</td>
<td>76.2</td>
</tr>
<tr>
<td>At center (in)</td>
<td>1/2</td>
<td>5/8</td>
<td>7/8</td>
<td>1</td>
<td>1 1/4</td>
<td>1 1/2</td>
<td>1 5/8</td>
<td>1 7/8</td>
<td>2</td>
<td>2 1/2</td>
<td>3</td>
</tr>
</tbody>
</table>

7. Overhead layshaft or counter shaft

The layshaft (sometime called countershaft) is the shaft used to transmit power from the engine power-take-off to the winch mounted on the deck of a fishing vessel.

When the P.T.O. is taken from the front of the engine it is often a long way from the position where the winch is to be mounted. The simplest way to transmit the power is to fit the layshaft overhead, usually fastened underneath the deck to the deck-beams. The lay-shaft should be installed so that the line of the engine and the lay-shaft are parallel both in plan view, and cross-sectional view. Belt pulleys must also be exactly parallel and in line with each other, to prevent belt wear, etc., as discussed earlier. (See Fig. 2.28)
Lay shaft diameter

The lay-shaft is normally made of solid shafting and the diameter must be large enough to transmit the power available at the engine. The method of determining the diameter of the shaft required for a given horse power and torque (turning action) is very involved: an approximate guide (but guide only) is shown in Table 7.

Table 7 Table of shaft size showing Hp transmission by shafting.

<table>
<thead>
<tr>
<th>Mild steel shafting</th>
<th>Speed of layshaft (in revolutions per minute)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Diameter in mm</td>
<td>100 rpm</td>
</tr>
<tr>
<td>40 1 ½ approx.</td>
<td>4</td>
</tr>
<tr>
<td>45 1 ¾ approx.</td>
<td>6</td>
</tr>
<tr>
<td>50 2</td>
<td>9</td>
</tr>
</tbody>
</table>

In the above table the figures given indicate the HP which can be transmitted by a power take-off shaft of mild steel driven through a flexible coupling.

For example: given an installation using a winch, which requires a maximum of 60 HP, with a layshaft speed of 1000 rpm. From table 7, the layshaft diameter to transmit the torque generated by this HP and rpm would be 42 mm.

8. Flexible coupling

A flexible coupling should always be fitted between the engine and PTO lay shaft, also between the dog clutch and winch, to take up any slight misalignment, which may occur in use. If helps prevent strain on the winch or axle driving bearings and oil seal. Running with any misalignment will cause oil seals to leak, bearings to fail, or possibly the fracture of the shaft itself.

There are several types of flexible couplings made using the principle of a rubber block, bonded to and sandwiched between two metal discs.
Flexible disc coupling

On small engine and hauler installations the flexible coupling may be made up out of a disc of thick rubber or canvas/rubber material as used on flat belts. Holes are cut in the disc in opposite pairs at 90° to each other. (See Fig. 2.34)

Vehicle propeller shaft coupling

A simple flexible coupling may be made from a vehicle propeller shaft of the type shown in Fig. 2.35. A typical propeller shaft consists of a tube, with a universal yoke joint at each end, at right angles to each other.

Cutting the tube in two places, removing the central cutout portion, and welding together the end sections with the yoke joints complete makes a coupling.

It is important to ensure that the tubes are welded together so that the yokes are at right angles to each other, as they were before the propeller shaft was cut. This allows the joints to work with the maximum amount of movement without straining the bearings, and also to balance the unit to cut down vibration.
When the tube is being welded, avoid welding too much at one time otherwise the joints will get too hot and damage the needle roller bearing in the yokes. Re-lubricate the bearing after welding.

9. Clutch type

Power taken from the engine by means of a power take-off to drive a winch or other deck machinery must be controllable, so that the drive can be disconnected or separated from the engine. In practice it is necessary to provide some form of clutch to do this, and most simple methods are as follows:

- Fast and loose pulleys
- Jockey pulley
- Dog clutch
- Friction clutch

- Flat belt drive only
- Flat belt drive, and some V-belt drives
- Flat or V-belt drive, or direct couple
- Flat or V-belt drive, or direct couple

Fast and loose pulleys

This system for flat belt drive only, uses an extra wide pulley on the driving unit, and a pair of pulleys on the same shaft on the driven unit. (See Fig. 2.36). One of the pair of pulleys is keyed to the shaft to drive it, whilst the other pulley is free to rotate on the shaft to drive it, whilst the other pulley is free to rotate on the shaft and therefore does not drive it. To engage or disengage the drive, the belt is moved across to the ‘fast’ pulley (the one keyed to the shaft) or to the ‘loose’ pulley (the one free to rotate). This normally is done by means of a fork or yoke over the belt at a point just before the belt goes over the fast and loose pulleys.
Jockey pulley clutch

A jockey pulley is used to tighten the flat belt between the driving and driven pulleys to provide a positive drive. By slackening the jockey pulley the belt is too loose to transmit any power between the engine pulley and the driven machinery, so providing a clutch.

The jockey pulley is considered to be most effective if it is made to act on the outside of the belt, against the slack side. It should be positioned close to the engine-driving pulley to assist in making the belt “wrap” around it to provide a more positive grip.

The system is suitable for a flat belt drives system and should not be used with multiple V-belts. However, low HP applications with a single V-belt are possible.

The jockey pulley may be tensioned by a spring to add against the belt when in the ‘drive’ or clutch-in position. This helps to maintain an even pressure on the belt even if it stretches slightly. An operating handle or lever is connected to the jockey pulley so that the operator can control the drive from a position CLOSE to the controls of the winch or deck machinery being driven.
It is most important that the clutch lever can be reached quickly by the operator, to avoid accidents. The operating lever should be designed so that when the winch or deck gear drive is engaged there are no pins or locking devices to be removed before 'clutching out'.

For reasons of safety, all pins or locking devices should be fitted to keep machinery in the 'no-drive' or 'clutch-out' position only, not in the 'clutch-in' position.

**Dock Clutch**

A dog clutch is a mechanical unit fitted between the engine and the power take-off to allow the drive to the winch, etc., to be disconnected or connected as required.

In normal use the dog clutch can be engaged only when the engine is stopped. Any attempt to engage the drive while the engine is running will result in damage either to the dog clutch or the driven equipment. If can be disengaged whilst the engine is running. A typical dog clutch arrangement is shown in Fig. 2.38.
Friction clutch

The friction clutch is a similar unit to a dog clutch, except that the engagement between the driving and driven units is by a friction disc or discs instead of the dog clutch. The friction disc drive is very much smoother, and the unit is designed to be clutched in or out of drive whilst the engine is running, and does so smoothly without shock to the driven gear.
Hydraulic Power Source

1. Basic knowledge

1.1. What is oil hydraulics?

Oil hydraulics is a power (energy) – transmitting assembly controlling power and motion by means of hydrodynamics. A power source (driver) consists of a motor and an engine. Power is transmitted to an actuator (oil hydraulic motor for rotational motion or cylinder for rectilinear reciprocating motion) attached to the portion intended to driven.

A transmission medium is oil hydraulic operating oil (hereafter may be referred to simply as oil or fluid) which is sucked up or discharged by a pump mounted to the driver.

Valves (Pressure control valve, directional control valve, and flow control valve) control power and motion.

1.2. What is oil for oil hydraulics?

It means oil hydraulic operating oil, which is available at any gas station. For commercially available brands, physical properties, viscosity characteristics, etc., refer to Article 5-1 (P. 112). Price of this oil is 30 to 50 % higher than that of automobile gasoline.

1.3 Are there any another methods except oil hydraulics?

In addition to the oil hydraulic method, electricity, machine, pneumatic and water hydraulics are available in order to transmit power (energy).

After fully examining merits and demerits of each method, the most suitable method will be adopted. Among others, the oil hydraulic method will continue to be widely used because it is best suited for transmission of large power.
They are often used in the best combination, such as hydraulic method plus electrical method.

(Table 1) Features oil hydraulics

Comparison of power transmission system

1. Range of power transmission
   Electrical > Fluid > Mechanical
2. Simplicity and accuracy of control (including remote control)
   Electrical > Fluid > Mechanical
3. Amplitude of transmitted power
   Fluid > Mechanical > Electrical
4. Safety
   No particular difference among these three methods.
5. Manufacturing cost
   Small horse power: Fluid, Mechanical > Electrical
   Large horse power: Fluid, Mechanical > Electrical
6. Resistance to corrosion
   Oil hydraulics > Pneumatic > Water hydraulics
7. Accuracy of control, amplitude of power, safety
   Oil hydraulics > Water hydraulics > Pneumatic
8. Price of medium, availability
   Pneumatic > Water hydraulics > Oil hydraulics

1.4 Merits of oil hydraulics

1. Large power can be transmitted in spite of its compact size.
2. The amplitude of output and its velocity can be easily controlled without stage.
3. Automatic and remote control are possible.
4. Start-up from full-load is possible.
5. Countermeasures against overload can be easily taken.
6. The location of input and output units can be freely changed.
7. Small inertia of moving elements allows quick start-up and stop.
8. Accumulator facilitates accumulation of power.

1.5 Demerit of oil hydraulics

1. Speed of an output unit is prone to change due to fluctuation of oil temperature (fluctuation of oil viscosity).
2. Low mechanical efficiency of power transmission of the oil hydraulic equipment leads to large power loss.
3. Piping works such as bending of pipes, welding, pickling and flushing are troublesome.
4. The oil hydraulic equipment is prone to cause noise and vibration.
5. Maintenance of oil is troublesome. It is necessary to check service life of oil, prevent foreign matters from entering, and clean filters.
6. Oil is flammable and therefore dangerous. Application of Fire Services and introduction of frame resistant fluid should be considered.
7. Care must be taken to prevent oil leakage from pipe joints, packing, etc.
1.6. What is transmitted power (energy) in hydrodynamics?

In hydrodynamics, power L1 is expressed by the following equation.

\[ L_1 = \frac{P \times Q}{612} \quad \text{kW} = \frac{P \times Q}{450} \quad \text{PS} \]

Where, P=Pressure (kgf/cm²) Proportional to power (amplitude of the output)
Q=Flow (l/min) Proportional to motion (velocity of the output)

For reference, power in mechanics is expressed as follows.

1) Power L2 in linear motion:

\[ L_2 = \frac{F \times V}{102} \quad \text{kW} = \frac{F \times V}{75} \quad \text{PS} \]

Where, F=Weight of object (kgf)
V=Velocity of object (m/sec)

2) Power L3 in rotational motion:

\[ L_3 = \frac{N \times T}{975} \quad \text{kW} = \frac{N \times T}{716} \quad \text{PS} \]

Where, N=Rotational speed (rpm)
T=Torque (kgf-m)

**1.7. Official units and common names expressing pressure, flow, etc. (Table 4)**

**Table 4** Official units and common names

1. Power, weight kgf, Ton : kilogram – force, ton
2. Liquid volume l, cc
3. Pressure kgf/cm² : Kilogram-force per square centimeter
4. Flow l/min : Liter per minute
5. Torque kgf-m : Kilogram-force meter
6. Rotational speed rpm : Revolutions per minute
7. Velocity m/min, cm/sec : Meter per minute, centimeter per second
8. Horse power, power Hp, PS, KW : Horse power, kilowatt
9. Displacement cc/rev : Cubic centimeter per revolution
    (Note) Mass Power, weight Kg (kilogram)
    Torque Kgf (kilogram-force)

1.8. What is pressure?

Pressure is the force per unit area generated on fluid in opposition to the load exerted on the actuator...kg/cm². “Press” means a pressed of compressed state of an oil hydraulic fluid discharged from a pump by the loaded actuator. When no load is imposed on the actuator, it is driven at no-load at a pressure = 0 kgf/cm².
The greater load is the higher generated pressure, therefore larger energy will be required according to the equations in Article 1-6. Pressure, which is equivalent to voltage in electricity, is invisible and therefore measured with a pressure gauge.

1.9. What is flow?

Flow is the volume of fluid flowed per unit time (generally per minute)… l/min Pump generates a flow by sucking up fluid from oil tank and discharging it to the outlet. The discharge can be increased either using a larger pump or increasing the number of revolutions of a pump, but required power is larger according to the equations in 1-6. The larger flow into the actuator gives the higher output speed. However, stateless reduction of speed is possible by controlling the flow by using a flow control valve. Flow, which is equivalent to electric current in electricity, is invisible and measured with a flow-meter.

1.10. Unit conversions of pressure, flow, power, etc.

Although JIS and ISO, inch use m, kgf and l, pound and gallon are still used in U.S.A. and U.K.

1.11. Basic form of oil hydraulic equipment.

Figure 3 shows a basic shape of the oil hydraulic equipment. When a certain amount of force is exerted on a piston pump, pressure \( P = F/A \) is generated, which is obtained by dividing the amount of force by sectional area of the pump.

Although pressure increases with an increase of force exerted on the piston, it increases only up to the point where the pressure counterbalances the load.

If the load is constant as shown in Figure 3, pressure does not increase above this point. In other words, resistance (load) opposing the flow of fluid causes pressure, and when required pressure is generated, the load can be moved.

The speed at which the load moves depend only on the volume of fluid supplied to the cylinder. Figure 3 shows that, when the piston is pressed down fast, a larger volume of fluid is supplied to the cylinder in unit time and therefore the load rises faster.

The system in Figure 3 shows a basic form. The actual equipment should be provided with the following controls.

1) Directional control of cylinder movement.
2) Speed control of cylinder movement.
3) Restrictive control of a maximum load of cylinder.

In addition, instead of a manual piston pump, a continuous drive rotary pump should be used. In order to give a clear explanation, Figure 4 shows a simple oil hydraulic circuit diagram.
Pump (1) is driven by a motor or engine (internal combustion engine), sucks up fluid from tank (2), and discharges it to the piping leading to a cylinder provided with various control valves or oil hydraulic motor. If a flow is free from a resistance, fluid does nothing but flows. However, as shown in Figure 4, if the cylinder has a load, the pressure rises until it counterbalances the resistance that is until the cylinder begins to move. The maximum pressure must be restricted in order to prevent the oil hydraulic equipment from being damaged, which means to protect the system against an excessive load. The upper limit of working pressure can be set with relief valve (safety valve) (3).

The relief valve is designed so that a spring pushed a steel ball against a valve seat by mechanical force, and the pressure in the piping acts on the surface of the steel ball.

When force (Force=Pressure * receiving area of valve seat) generated on the steel ball by the equation mentioned above, i.e. F=P*A, becomes larger than spring force, the valve opens.

The flow discharged from the pump returns from the relief valve to the tank, and the pressure does not rise above the set pressure.

In Figure 4, a passage is formed from port P to port A (connection port) in directional control valve (5) and fluid flows into the cylinder. On the other hand, Figure 6, since spool (6) in the valve is pushed to the left. The passage is formed from port P to port B so that flow into the cylinder is opposite and pulled in piston rod (4.2).
At this time, the fluid in a room opposite to the cylinder passes from port A to port T of directional control valve (5) and is pushed back to the tank.

In Figure 5, the spool (6) in the directional controlling valve (5) is located in the middle between the position in Figure 4 and that in Figure 6. Since ports A and B are not connected to ports P and T, the cylinder stops. The rate of fluid flowing into the cylinder or discharged form the cylinder is changed in order to control the speed at which the load moves. In Figure 7 the flow is regulated with throttle valve (7). The amount of fluid (flow) flowing into the cylinder per unit time is decreased by reducing the opening area of the valve passage in throttle valve (7) so that the cylinder can be moved slowly. At this time, redundant fluid among excessive fluid discharged by the pump returns to the tank from relief valve (3).

The state of pressure is as shown below in the circuit in Figure 7. That is, a pressure between pump and the throttle valve (7) reach the maximum pressure set with relief valve (3), and a pressure between throttle valve (7) and cylinder is determined by the size of the load.
The larger the load is, the higher the required pressure becomes. Therefore, adjust the pressure of relief valve (3) to a high setting.

<table>
<thead>
<tr>
<th>Relief valve (Safety valve) (3):</th>
</tr>
</thead>
<tbody>
<tr>
<td>Setting the upper limit of working pressure</td>
</tr>
<tr>
<td>- Controlling the amplitude of the output</td>
</tr>
<tr>
<td>- Preventing damage</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Directional control valve (Safety valve) (5):</th>
</tr>
</thead>
<tbody>
<tr>
<td>Switching the direction of flow</td>
</tr>
<tr>
<td>- Controlling the direction of the output</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Throttle valve (Safety valve) (7):</th>
</tr>
</thead>
<tbody>
<tr>
<td>Changing the passing flow</td>
</tr>
<tr>
<td>- Controlling the speed of the output</td>
</tr>
</tbody>
</table>

1-12. Fundamental illustration of oil hydraulic circuit

In reality, an oil hydraulic circuit diagram is not represented by such figures 4 through 7.

Each unit of the oil hydraulic equipment is represented with simplified oil hydraulic graphic symbols are internationally standardized by ISO (International Standards Organization), and in Japan they are standardized by JIS (Japan Industrial standards) B-0125.

As of now, there are slight differences between ISO and JIS. Figure 8 is an actual oil hydraulic circuit diagram represented figure 7.

Oil hydraulic element equipment

<table>
<thead>
<tr>
<th>1. Power source</th>
<th>Oil hydraulic pump</th>
</tr>
</thead>
<tbody>
<tr>
<td>2. Actuator</td>
<td>Oil hydraulic motor, cylinder</td>
</tr>
<tr>
<td>3. Valve</td>
<td>Pressure control valve</td>
</tr>
<tr>
<td></td>
<td>Directional control valve</td>
</tr>
<tr>
<td></td>
<td>Flow control valve</td>
</tr>
<tr>
<td>4. Accessory</td>
<td>Pressure gauge, filter, oil tank, cooler, Heater, oil thermometer, accumulator, etc.</td>
</tr>
<tr>
<td>5. Medium</td>
<td>Oil hydraulic operating fluid</td>
</tr>
<tr>
<td>6. Piping material</td>
<td>Rubber hose, steel pipe, etc.</td>
</tr>
</tbody>
</table>

1-14. Three factors controlling the output

<table>
<thead>
<tr>
<th>1. Amplitude of output</th>
<th>It is controlled with a pressure control valve and by the size of an actuator (size and displacement).</th>
</tr>
</thead>
<tbody>
<tr>
<td>2. Direction of output</td>
<td>It is controlled with a directional control valve, variable displacement pump, and variable displacement motor.</td>
</tr>
<tr>
<td>3. Speed of output</td>
<td>It is controlled with a flow control valve, variable displacement pump, and variable displacement motor.</td>
</tr>
</tbody>
</table>
2. Oil hydraulic pump and Oil hydraulic motor

First of all, among the oil hydraulic element equipment (Table 5 on P.11), let us begin with the oil hydraulic pump and motor. Since the oil hydraulic motor has a similar structure, they are often explained at the same time.

The oil hydraulic pump and motor play a different roles as described below. The pump whose shaft is rotated by a driver sucks in fluid and discharges it from a discharge opening, while the oil hydraulic motor receives fluid from the pump and its shaft rotates. (See figure 2 on P.1)

2.1. Types and features (Table 7)

(Table 7) Types & features of gear pump/motor

<table>
<thead>
<tr>
<th>Gear type</th>
<th>Pressure</th>
<th>Flow</th>
<th>Rev. number</th>
<th>Price</th>
<th>Noise</th>
<th>Variable delivery</th>
</tr>
</thead>
<tbody>
<tr>
<td>Gear type</td>
<td>Medium</td>
<td>Small to medium</td>
<td>High</td>
<td>Low</td>
<td>Medium to high</td>
<td>Impossible</td>
</tr>
<tr>
<td>Vane type</td>
<td>Low to medium</td>
<td>Large</td>
<td>Medium</td>
<td>Medium</td>
<td>Low</td>
<td>Possible</td>
</tr>
<tr>
<td>Piston (Plunger type)</td>
<td>High</td>
<td>Medium to large</td>
<td>Low to medium</td>
<td>High</td>
<td>Medium to high</td>
<td>Possible</td>
</tr>
</tbody>
</table>

Note. Low pressure = 70 kg/cm² or less,
Medium pressure = 140 to 175 kg/cm² or less
High pressure = 210 to 350 kg/cm² or less

1) Gear pump/motor
It is relatively inexpensive because of a small number of components and can be driven at a high-speed revolution.

2) Vane pump/motor
It has little pulsation and is characterized by a low noise. Although a variable displacement type is available, generally a low-pressure type is popular on the market.
A pressure-balance type has a long service life in its bearing section.

3) Piston pump/motor
It can be used under a high pressure, and a variable displacement type is also available but relatively expensive.
2.2. Structure and working principle of oil hydraulic pump.

The pump creates a flow of liquid, and usually sucks in fluid from the tank and discharges it from its discharge opening. The discharged fluid reaches the actuator (cylinder, oil hydraulic motor, etc.) through various control valves. After operating the actuator, it returns to the tank. A load imposed on the actuator stops a flow of fluid. At this point, pressure is generated to counterbalance the load. Pressure in the oil hydraulic circuit is not generated by the pump but generated with reference to the actuator’s load. As a result, pressure is exerted on the pump and a load is imposed on the pump-driving source (electric motor or engine).

2.2.1. Gear pump

The gear pump is a fixed displacement type and the following two types are available.

1) Internal gear pump (Figure 12)

![Fig. 12 Gear pump](image)

When a set of internal gears 2 and 3 rotates clockwise in main body 1 as shown in the figure, the pump sucks in fluid from below and discharges it from above.

Both gears rotate in the same direction. The fluid flows into the pump by a difference between negative pressure generated in the section where the mesh of the gearwheels separates and atmospheric pressure exerted on the surface of fluid in the tank.

The phenomenon is known as “the pump sucks”.

The fluid fills up a space where the mesh separates and is led to the discharge opening, passing through crescent-shaped component 4.

The fluid equivalent to the volume of the space where the mesh separated before is discharged form the discharge opening so that gears can intermesh again.

The intermeshing gears prevent the fluid from flowing backwards from the discharge opening to the suction opening.
2) External gear pump (Figure 13)

This type of pump has a set of intermeshed external gears. When gear 2 is rotated in the direction shown with an arrow (clockwise), gear 3 rotates in the opposite direction (counterclockwise).

The principle of suction and discharge is the same as in the case of an internal gear pump. This suction/discharge phenomenon occurs in the un-meshing and re-meshes section. As shown in Figure 13, a small amount of fluid remains in the mesh section. This small closed chamber undergoes such a change that it returns to the original state after the size of the chamber (volume of the chamber) becomes small from the time when it starts to close to the time when it opens.

For this reason, a very high pressure is generated in this small chamber, which causes noise and exerts bad influence upon the service life of bearings.

In order to prevent this, bearing block 6 in Figure 13 is provided with a clearance groove (section A) so that the pressure in this small closed chamber does not become so high. Furthermore, note that the clearance between bearing block and gear side face is related to the following important matter.

- If the clearance is too much: Low friction and large leak.
- If the clearance is too little: High friction and small leak.

If the size of this clearance is fixed, the leak increases with an increase in friction and the volumetric efficiency (a value indicating the decrement of discharge as to the increase of leak from each clearance accompanying the increase of the working pressure) further decreases as the working pressure rises.

The pump shown in Figure 13 adopts a structure known as a shroud-movable type, and the bearing block, which has a structure in which a shroud and a bearing are incorporated. It moves in the direction where the sides of gear 5 are pushed by means of a discharge pressure led by seal material 7 and decreases the clearance.
2.2.2. Vane pump

1) Fixed displacement type

As shown in Figure 14, main components of a vane pump are a main body, cam ring 1, rotor 2, and vane 3. The inner diameter portion of the cam ring has an eccentric circle in both directions. The rotor rotates at the center of the cam ring. Two vanes (double vane type) inserted in the radically cut rotor grooves jump out in the radial direction by means of both centrifugal force and discharge pressure applied to the root of the vane, and slide in contact with the inner diameter portion of the cam ring.

In Figure 14, when the rotor rotates in the direction shown with an arrow (counterclockwise), a negative pressure is generated in chamber 4 because it becomes gradually wider on the suction side and fluid is sucked in from the tank.

When the rotor further rotates, chamber 4 reaches the discharge side and in turn the fluid is discharged because it becomes gradually small.

This suction and discharge stroke occurs twice while the rotor is making a rotation.

Since two suction chambers and two discharge chambers are located symmetrically, the drive shaft and bearing can be easily designed because the load imposed on the rotor is offset.

This type is called a pressure balance bane pump.

![Fig. 14 Vane pump](image)

The pressure on the discharge side idled to chamber 5 at the root of vane 3 shown in Figure 14, which optimizes contact force between the tip of the vane and the inner diameter portion of the cam ring so that an oil leak from this part is prevented. (See Figure 15.)

![Fig. 15 Needle of vane pump](image)
However, since this contact force becomes too strong on the suction side, chamber 6 is connected to the tank. The pump of this example can be used under a pressure as 175 kgf/cm².

2) Variable displacement type 1 (Figure 16)

The principle of the suction and discharge of this pump is the same as that of the above-mentioned fixed displacement vane pump. However, as shown in Figure 16, the inner diameter portion of cam ring 1 is not eccentric but circular so that it is called an unbalanced vane pump. Although spring 2 is adjustable with adjusting screws 6 by giving cam ring 1 a maximum eccentricity, the maximum amount of eccentricity, that is, maximum displacement volume can be regulated with adjusting screw 5. The vertical position of the cam ring can also be adjusted with adjusting screw 4.

![Figure 16 Variable displacement vane pump](image)

The pressure on the discharge side of the pump acts on the upper half of the cam ring inner-diameter portion as shown in Figure 17. As a result, horizontal force opposing to the spring is generated by component force.

If a discharge pressure becomes high, the eccentricity becomes small. When the preset pressure is reached, the discharge flow is almost zero.
At this time, since the preset pressure is maintained and only the minimum required flow to a leak is discharged from the pump, a power loss and a rise in oil temperature are minimized. Figure 18 shows a P-Q chart (relationship diagram between pressure on the discharge side P and discharge flow Q) of the pump.

The gradient of the horizontal line in this P-Q chart is generated by the spring constant, and this gradient differs depending on four types of spring (there are four types by pressure adjustment range) available for this pump.

Air vent valve 7 in Figure 16 is standard. When there is no air at the start-up, the valve is open because the passing resistance in this portion is small. When fluid comes, the passing resistance becomes high, and the steel balls move downward in opposition to the spring so that the passage is automatically closed. Thus, air bleed is completed.

3) Variable displacement type 2 (Figure 19)

This pump is of a pressure adjusting type like the above mentioned variable displacement vane pump. There are, however, some differences. In the first place, this type of pump has two vanes (double vane type) incorporated in one groove. The second place, this pump is of a flow control type.

A preset force at the approximated rate 1 to 2 presses to cam ring 4. Although spring 3 in piston 2 is weak, it always pushed cam ring 4 to the left so that a smooth pump actuation is possible. The preset pressure cam be changed with spring 5 in pressure regulating valve 6, and this spring pushes spool 7. When the preset pressure is reached, spool 7 moves downward in opposition to spring 5 and the right chamber (chamber having spring 3) of piston 2 is connected to the tank.

As a result, cam ring is moved to the right by means of discharge pressure applied to the left chamber of smaller piston 1, and the flow is reduced to the pump discharge flow required by the actuator.
The cam ring is move required by the actuator. It moved not directly by the force of the spring, but by the oil hydraulics through pressure control valve 6. Therefore, a better pressure override characteristic (this refers to a pressure rising state from the pressure where the discharge flow becomes zero. The slower this pressure rise is, the better the pressure override characteristic is.) can be obtained than that in the P-Q chart shown in Figure 18.

![Fig. 19 Variable vane pump](image)

Since the cam ring is moved by the hydraulic force. A function to control the maximum discharge flow can be added by providing the pipe on the discharge side outside the pump with throttle valve 1 and manipulating it, as shown in the hydraulic circuit in Figure 20.

The pressure difference (P1-P2) before and after throttle valve 1 pushes a spool 2. When a flow exceeding the preset flow is discharged, this pressure difference becomes bigger and the internal connection in regulating valve 3 is switched so that the chamber on the back of larger piston 4 is connected to the tank. The cam ring is pushed by smaller piston 5 in such a way that the eccentricity is smaller, and the pump discharge is decreased. Therefore, the preset flow is maintained. The spool in regulating valves 3, throttle valve 1, and spring 2 is the same component elements as those of the flow control valve with pressure compensation described later. Therefore, the pump discharge flow corresponding to that pressure difference can be obtained by manipulating throttle valve 1 to change the pressure difference.

Furthermore, the constant output control (P*Q constant control) is possible by replacing the regulating valve in the vane pump with this shape with other one in order to prevent the overload of a pump driver (electric motor, engine, etc.).
2.2.3 Axial piston pump/motor

A pump with a piston is located axially (parallel to the axis) to the drive shaft is called an axial type and is popular on the market. Like the vane pump, a fixed displacement type and a variable displacement type are available. This pump is classified into a swash plate type (Figure 23) and a bent axis type (Figure 24) depending on a slight difference in structure. The bent axis type shown in Figure 24 is characterized by the fact that its mechanical transmission efficiency. In other word the mechanical efficiency of the pump/motor is slightly better than a swash plate type because fluid acts directly on the drive shaft through the piston, but there is no great difference in volumetric efficiency and weight.

The discharge flow is proportionate to the driving speed of rotation and the amount of piston stroke (S) in both pumps. Also, in case of oil hydraulic motor, the axial speed of rotation is proportionate to the supply flow and inversely proportionate to the amount of piston stroke. The output torque is in proportion to the pressure difference between the high pressure side (inlet) and the low pressure side (outlet).

1) Fixed displacement type of swash plate type

The one shown in Figure 25 has 9 pistons 5 arranged parallel to drive shaft 3, which perform reciprocating motion in cylinder block 4. Cylinder block 4 is connected to drive shaft 3 by spine.

The left end of piston 5 is connected to piston shoe 8 to which a universal joint can be attached, and this piston shoe is pressed to swash plate 6 with an inclination of 15 degrees by the force of spring 10 by means of parts 11, 12, and 9.

The one with the shape shown in this Figure 25 is of a fixed displacement type because the inclination of swash plate 6 cannot be changed. Fluid is supplied to the piston and discharge switching is controlled by port plate 7 fixed to casing 1.

That is, fluid is sucked from the side where a control holes of the port plate and connected to the tank by piston 5. As moves in the outgoing direction (leftward in the figure) from cylinder block 4, and the piston which moves in the opposite direction pushes out fluid to the discharge opening through another control hole of the port plate.

The pump is designed to high-pressure fluid on the discharge opening side passes through each hole of piston 5 and piston shoe 8. So that, lubrication the
universal joints of these two components as well as the sliding part between piston shoe 8 and swash plate 6 and also reduces contact force of this sliding part.

![Fig. 25 Axial piston pump](image)

2) Variable displacement type 1 of swash plate type

The one shown in Figure 26 is designed so that the inclination (hereafter called angle of inclination) of swash plate 2 can be changed to the drive shaft within the range of 0 to 15 degrees by regulator 5. Consequently, the displacement volume changes freely.

If the angle of inclination of this swash plate is zero, the swash plate is perpendicular to the piston. Since the piston stroke is zero, the displacement volume is also zero.

Adjusting screw 6 can restrict the maximum discharge flow.

![Fig. 26 Axial piston pump](image)
2) Variable displacement type (2) of swash plate type
(Figure 27 and 28)

The pump shown in Figure 27 is a combination of auxiliary pump 2, regulator 3, two relief-valves 5, and relief valve 4 for setting boost pressure (pressure for replenishing oil for the main circuit) in one.

This type of pump differs from those described so far in the following point. Piston 7 is not parallel to the drive shaft but is arranged slightly in a tapered form. If the speed of rotation becomes high, the force maintaining contact between piston shoe and swash plate 1 increase. In addition, contact face 8 between cylinder block and port plate is spiral. This contact face sticks fast without coming apart even in a high-pressure use so that there is no leak (oil leak) from this portion, resulting in improvement in volumetric efficiency.

Also, this pump is designed for a closed circuit. The fluid is returned from the actuator back directly to this pump instead of returning to the tank and a leak from the actuator or the like is made up by auxiliary pump 2.

![Figure 27 Variable piston pump](image)

**Figure 27 Variable piston pump**

A and B: Main circuit
G: Discharge opening or auxiliary pump
L1: Opening for replenishing fluid into the casing
L2: Drain connection port (connection port to return leaked fluid into the tank)
MA: Port for measuring pressure in main circuit A
MB: Port for measuring pressure in main circuits B
R: Fluid removal port in the casing
S: Inlet port of auxiliary pump
X1 and X2: Oil pressure inlet for controlling angle of inclination
Y1 and Y2: Connection port for remote control

![Fig. 28 Main circuit](image)
Two relief valves (with a check valve) 5 restrict the maximum pressure of main circuits A and B and prevent overload of the driver. Office 6 is used to adjust the speed when the angle of inclination is changed.

4) **Fixed displacement type of bent axis type**

The one shown in Figure 29 is an incorporation of drive shaft 2, cam plate 3, cylinder block 4, piston 5, connecting rod 6 for connecting piston and cam plate, port plate 7 and center pin 8 into housing 1.

Connecting rod 6 and center pin 8 have a universal joint to cam plate 3, and connecting rod 6 has a universal joint to piston 5. The former and the latter are jointed by means of pressure bar and “caulking” by rolling, respectively. Cam plate 3 is perpendicular to drive shaft 2, and seven pistons 5 are inclined 25 degrees to the drive shaft.

When the drive shaft rotates, cylinder block 4 also rotates. Piston 5 reciprocates in the cylinder block 4 also rotates. Piston 5 reciprocates in the cylinder block. It sucks fluid from a control hole of port plate 7 in a half-turn from the bottom dead center to the top dead center and discharges it from another control hole of the port plate.

Since the angle of inclination of the force applied to cylinder block 4 in the direction becomes small by means of absorbed discharge pressure is by center pin 8. Rolling of bearings installed on the periphery of the cylinder block, which are found in ordinary pump is not necessary.

![Fig. 29 Fixed displacement type of bent axis type](image-url)
5) **Variable displacement type 1 of bent axis type**

The one shown in Figure 30 is of a type in which the angle of inclination can be changed within the range of 0 to 26.5 degrees. The minimum discharge flow is secured with adjusting screw 5 and the maximum discharge flow is restricted with adjusting screw 4.

![Fig. 30 Variable displacement type 1 of bent axis type](image)

6) **Variable displacement type 2 of bent axis type**

The one shown in Figure 31 is of a type in which cylinder block 4, piston 5, port plate 7, and housing 9 move together. The angle of inclination changes to the drive shaft within the range of +/- 25 degrees. If the angle of inclination is zero, the displacement volume is also zero so that the discharge quantity is zero. With an increase of the angle of inclination, the displacement volume changes accordingly. Also, the suction opening and discharge opening are reversed depending on whether the angle of inclination is plus or minus. This pump is mainly used for a closed circuit.

![Fig. 31 Variable displacement type 2 of bent axis type](image)
2.3. How to control variable displacement pump and motor

There are various methods available to change the angle of inclination and displacement volume. These methods can be classified as shown below according to purpose.

1) Constant output control

This control method, which is generally called constant horse power control, constant input control, and constant P-Q control, is the most popular. The discharge flow Q is controlled by pressure on the discharge side of the pump P, and its relationship is expressed as $P \times Q = \text{constant}$.

![Figure 32 Relief valve diagram]

In other words, the pump is controlled so that the pump output is constant, and the output can be utilized to the full without overloading the drivers such as an electric motor and engine.

This is used for a winch, running gears, etc. with a big load fluctuation.

2) Constant pressures hold control

Discharge flow Q equivalent to the amount required by the actuator is discharged by pressure on the pump discharge side P so that this pressure P remains constant. As shown in the P-Q chart is Figure 18 (P.24), when the actuator is at rest, the pump discharge flow is almost zero discharge (called full cut off) so that power for driving the pump is minimized, resulting in energy-saving. The output of the actuator can be made constant. This is suitable for applying and holding pressure for a long time by a cylinder, such as rubber vulcanizing press, plywood press, etc.

3) External plot pressure control

Pump discharge flow rate Q is controlled on the command pressure from the external oil hydraulic source. This has no related to the pressure on the discharge side P. The command pressure is at the highest 40kg/cm², and discharge flow Q can be changed by changing the pressure.
4) **Manual control**

Discharge flow Q is controlled by turning an adjusting screw or round handle by hand.

5) **Electric control**

As contrasted with the manual control type mentioned above, discharge flow Q is controlled by turning it with a small electric motor with a reduction gear (reverse rotation is possible).

6) **Electromagnetic proportional control**

Pump discharge flow Q is controlled in proportion to the size of external command current value. The components on the pump side receiving a command current are an Electro-hydraulic servo valve and an proportional DC magnet.

This control method is faster in response time than the electric control mentioned above.

7) **Mechanical control**

Discharge flow Q is controlled by rotating (within approx. 120 degrees), or pushing and pulling a control lever mounted on the pump.

8) **Constant flow hold control**

A ship has an engine as drives a pump, all construction of equipment, vehicle and the speed of rotation is always changes.

In this control method, pump discharge flow Q is automatically made constant although the speed of rotation changed. The horsepower required for the pump maintains constant even at a high speed of rotation and the actuator is not accelerated.

This method is applied to an oil hydraulic source for driving a generator, steering wheel, movement the ship, a concrete mixer, etc.

**Precautions Handling**

1. **Difference between rated pressure and maximum pressure**

The rated pressure is the upper limit of pressure where a continuous use is possible. If a continuous use is done exceeding this rated pressure, the performance described later (volumetric efficiency and mechanical efficiency) decreases and sudden damage is easily caused by generation of heat. On the other hand, the maximum pressure means that the rated pressure can be exceeded by 1.2 to 1.5 times in a short time. However, unless as exact explanation of this “a short time” is given (for example, 5% or less of the duty cycle time), the upper limit of the preset pressure may be generated temporarily until a relief valve opens.
2. **Inlet pressure limit of pump**

The inlet pressure is described as –0.2 to 5 kg/cm². However, the pressure on the minus side, as is called a negative pressure, is lower than the atmospheric pressure, which is the limit against cavitations. On the other hand, the pressure on the plus side, if it is 1 kg/cm² or less, is determined based on the long-time withstand pressure of built-in oil seal. If this pressure is 1 kg/cm² or more, the pump has such a structure that a suction opening and oil seal is not directly connected, and the plus side pressure is determined according to the strength of other components.

When the pump sucks in fluid, if suction conditions are bad, the pump cannot normally suck it in, causing a cavity in fluid. This is called cavitations. As a result, the pump causes abnormal noise. The causes of cavitations are as follows:

1) A filter (strainer) on the suction side is clogged, a filter is too small, or a filtering accuracy is too high (the mesh is too fine).

2) An inlet pipe is too long, it has too many bends, or it is too thin.

3) A wrong fluid is chosen, or viscosity of fluid becomes high due to a decrease in oil temperature.

4) The pump is installed at a position to high from the oil level in the oil tank (installation position).

5) The pump is driven at the specified maximum speed of rotation or higher.

6) An air breather (Entrance/exit of air at the top of the tank. Provided with an air filter) is too small, or a built-in air filter is clogged.

3. **Range of working speed of rotation**

The limit of the minimum speed of rotation is determined, considering the following matters.

1) The decrease in volumetric efficiency exceeding the limit must not lead to damage of internal parts caused by abnormal generation of heat.

2) A poor total efficiency must not cause any problem as a matter of practicality.

3) In case of an oil hydraulic motor, non-uniform rotation easily occurs on the output shaft, but its limit must not be exceeded.

The limit of the maximum speed of rotation is determined, considering the following matters.

1) In case of a pump, no cavitations must occur.
2) Abnormal generation of heat in bearings and sliding parts must not lead to damage of internal parts.

3) A short service life of bearings must not cause such a situation that general use condition is not satisfied.

4) Serious noise must not cause any problem as a matter of practicality.

The optimum speed of rotation is related to working pressure, and the best total efficiency must be chosen from the performance table.

4. What is GD² value?

This is a value (kgf-m²) to express a flywheel effect of parts rotating in an oil hydraulic motor. The GD² value of an oil hydraulic motor is installed. We can know an accelerating / decelerating torque required to start or stop the entire equipment, time required for the equipment to reach the specified speed of rotation, or time required for the equipment to stop.

5. What is torque constant?

This is an output torque of an oil hydraulic motor generated when an effective differential pressures (A difference in pressure between inlet pressure P₁ and outlet pressure P₂. It is also described as P₁-P₂ = ΔP.) of 1 kgf/cm² is applied to an oil hydraulic motor. Torque constant x Effective differential pressure = Output torque. Since mechanical efficiency is not taken into consideration for this output torque, the actual output torque decreases by the amount equivalent to mechanical efficiency.

6. What is P-q value?

Since the input power concentrates on one drive shaft in the multiple pump, this section may be wrenched off unless it is strong enough.

The P-q value is (P x q) and is obtained from the following equation. (Refer to “Basic calculation equation: on P.52.)

\[ P1 \times q1 + P2 \times q2 + P3 \times q3 + \ldots = T \times 2\pi = P – q \text{ value} \]

Where,

- \( P1 \) = Pressure of the first pump
- \( q1 \) = Displacement of the first pump
- \( P2 \) = Pressure of the second pump
- \( q2 \) = Displacement of the second pump
- \( T \) = Allowable torque of the drive shaft

7. Rotation direction

If the driver and pump rotated clockwise seen from the drive shaft, it is a right-handed rotation. If the driver rotates clockwise, a reverse rotation, that is counterclockwise rotation, must be chosen for the pump.
If the pump is rotated reversibly, the suction opening and discharge opening are reversed. For this reason, pay attentions to the rotation direction clearly are two types of pump, right-handed and left-handed rotation. Also, a bi-directional type is available for the oil hydraulic motor. Therefore, there are total three different types. Although there is little difference in appearance, the internal parts are slightly different from each other. Note that they cannot be replaced with each other.

8. Procedure of drain piping

Unless internal leaks oil in the gear motor, vane pump/motor, piston pump/motor except the gear pump is let escape into the tank, the built-in oil seal is damaged. The piping for this purpose is called the drain piping.

Only the gear pump let this internal leak oil escape to the suction opening in the pump itself so that the external drain piping is not necessary.

Internal leak oil is indispensable because it serves for both lubrication and cooling of pump/motor bearings, but too much internal leak oil causes a decrease in transmission power. From the above-mentioned, the drain piping must be done according to the following procedure.

1) Use the highest drainpipe opening and fill the main unit with internal leak oil. In case there is only one drain pipe mouth and it comes to the bottom, arrange the drain pipe higher than the top position of the main unit once, then connect it to the tank so that oil in the main unit does not escape.

2) The long-time withstand pressure of a built-in oil seal is described in the catalog specification which is generally about 1 kg/cm². For this reason, it is necessary to hold down the passing resistance of the drain pipe to this withstand pressure or lower, and attention must be paid to the thickness and length of the pipe, and number of bends. Furthermore, connect the drainpipe to the tank singly. Do not join it to a return pipe through which a large amount of oil passes.

3) Before performing a trial run or replacing a pump/an oil hydraulic motor, pour fluid from the drain port, arrange the drain pipe, then put the pump/oil hydraulic motor into operation.

9. How to see the performance table and efficiency

Judge whether the performance is food or poor by seeing the following three types of efficiency.
1) Volumetric efficiency

Generally, efficiency is expressed with $\eta$ and a subscript indicates what efficiency it is. Volumetric efficiency is described as $\eta_v$ and is obtained from the following equation, which shows a condition of the internal leak of the pump/motor.

\[
\begin{align*}
\text{Volumetric efficiency of oil hydraulic motor} & \quad \text{Outlet flow at a very low pressure without internal leak} \\
& = \frac{\text{Outlet flow at a measured pressure + External drain flow rate}}{\text{Discharge flow rate at the measured pressure with internal leak}} \times 100 \% \\

\text{Volumetric efficiency of pump} & \quad \frac{\text{Inlet flow rate at suction side + External drain flow rate}}{\text{Discharge flow rate at the measured pressure with internal leak}} \times 100 \%
\end{align*}
\]

(Note) Make an arrangement so that the speed of rotation is always constant or make compensation by means of a calculation so that it is always constant.

If the volumetric efficiency of the pump is 90%, in reality it discharges only 90 l/min even if it sucks in 100 l/min.

The remaining 10 l/min is returned to the tank with the drainpipe as an internal leak. If the volumetric efficiency of the oil hydraulic motor is 90%, only 90 l/min is used for rotating the oil hydraulic motor even if 100 l/min is flowed in so that the speed of rotation decreases by 10%.

The higher the pressure is, the more the internal leak is. The higher the oil temperature is, the more the internal leak is. As a result, the volumetric efficiency decreases. Also, the internal leak flow does not change so much if the working pressure and oil temperatures are determined. Therefore, if the pump/motor rotates at a low speed, the “discharge (outlet) flow at a very low pressure without internal leak” becomes small, and a proportion with the internal leak flow becomes high. As a result, the internal leak flow becomes high. As a result the volumetric efficiency decreases.

2) Mechanical efficiency (also called torque efficiency)

It is described as $\eta_t$. Transmission torque decreases due to a rolling resistance or sliding resistance of pump/motor bearings. The mechanical efficiency represents this decrease rate. The transmission efficiency of a reduction gear or the like corresponds to this.

Torque on the axial side $T_1$ is measured with a torque meter, and torque on the oil pressure side of a pump/motor $T_2$ is obtained from the following equation. The proportion of these values is mechanical efficiency.

\[
T_2 = \frac{(P_1 - P_2) \times q}{200\pi} \times 100 \%
\]

Where, 
- $P_1$ = Inlet pressure (kgf/cm$^2$)
- $P_2$ = Outlet pressure (kgf/cm$^2$)
- $q$ = Displacement (cc/rev)
3) Total efficiency

It is expressed only with $\eta$, but a subscript such as $\eta_p$ and $\eta_m$ are added to distinguish between pump total efficiency and oil hydraulic motor total efficiency.

This total efficiency $\eta$ has the following relationship with volumetric efficiency $\eta_v$, mechanical efficiency $\eta_t$.

$$\eta = \eta_v \times \eta_t$$

This means that if two efficiencies are known, the other efficiency can be calculated. The performance table of a combination of either $\eta_v$ and $\eta$ of $\eta_t$ is sufficient as a performance table.

Since the total efficiency represents an overall efficiency of the pump/motor, it affects the transmission efficiency of the entire oil hydraulic equipment, which is the power (energy) transmission equipment.

That is, the input from the driver decreases by the amount equivalent to the total efficiency of the pump/motor, which becomes an output of the oil hydraulic equipment. Therefore, a model having a good total efficiency must be chosen and the equipment must be used taking the use conditions into consideration.

4) How to see performance table

![Performance Curve of Hydraulic Pump/Motor](image)

Fig. 37 performance curve of hydraulic pump/motor

10. Service life

The service life of a pump/motor is determined by the following two items.

1) Service life of bearing (incase of a roller bearing)

The calculated service life is obtained from the basic load carrying capacity of a bearing, speed of rotation under use conditions, and bearing load (proportionate to working pressure). In case a working pressure exceeds 200 kgf/cm² in a piston pump/motor having a rated pressure of 320 to 350 kgf/cm². Note that a service
life as per calculated cannot be obtained due to an influence of dust in oil unless a filter of 40 to 25 μm or less is installed on the tank return pipe. In addition to a filter of 100 μm (150 meshes) is generally installed on the suction pipe.

2) Use limit due to decrease in volumetric efficiency

When dust of abnormal sizes entered pump/motor from the inlet. A front shroud in the gear pump/motor and a cylinder block face of the sliding face between a port plate and a cylinder block in the piston pump/motor may be damaged by the dust because they are made of low hardness material (copper alloy). Therefore, a leak from this portion increases, resulting in a low volumetric efficiency.

It generally is said that internal parts are damaged due to abnormal generation of heat in a continuous use at a volumetric efficiency of 70% or less. There is no problem although temperature of the pump/motor main unit is higher by 20 °C than oil temperature. However, if it exceeds this temperature, overhaul and inspection are required.

11. Alignment of shaft and procedure of horizontal pulling

Generally, in case of a pump/motor, a chain coupling (often used if the transmission power is 100 kW or less) or gear coupling (often used if the transmission power is 100 kW or more) is used for connection in a shaft coupling to the shaft of the counterpart.

At this time, unless the center of the shaft is aligned with each other, abnormal force is generated, which may damage a shaft coupling or bearing in the pump/motor in a short time. Make an alignment with accuracy shown in the figure below.

In the same meaning most models cannot be used by horizontally pulling the pump/motor shaft. Incase it is driven by horizontal pulling, do so according to the figure below.
Fig. 38 Alignment of shaft and pulley

12. Starting torque of oil hydraulic motor

When the oil hydraulic motor is started, a starting torque is required to overcome a static friction resistance. After it is started, it can be driven at a torque lower than the starting torque due to a dynamic friction resistance. For this reason, when a full load is imposed at the start of the oil hydraulic motor, it cannot be started unless the starting torque is taken into consideration. The starting torque is calculated from the starting torque efficiency.

13. Using oil hydraulic motor at a low speed of rotation and reduction gear

When the oil hydraulic motor is used at a low speed of rotation, its efficiency decreases and non-uniform rotation (called stick slip) occurs. For this reason, install a reduction gear, if necessary. Types and features of reduction gears commercially available and generally used are described below.

Table 8

<table>
<thead>
<tr>
<th>Type</th>
<th>Features</th>
<th>Reduction ratio</th>
<th>Transmission efficiency</th>
<th>Price</th>
</tr>
</thead>
<tbody>
<tr>
<td>Spur gear</td>
<td>(Single)</td>
<td>1/3 ~ 1/5</td>
<td>90 ~ 92%</td>
<td>Low</td>
</tr>
<tr>
<td>Spur gear</td>
<td>(Double)</td>
<td>1/9 ~ 1/25</td>
<td>85%</td>
<td>Medium</td>
</tr>
<tr>
<td>Planetary</td>
<td></td>
<td>1/6 ~ 1/30</td>
<td>96%</td>
<td>High</td>
</tr>
</tbody>
</table>

Note: Usually, a planetary reduction gear is used because a double spurs gear type. Reduction gear is poor in transmission efficiency.
14. Countermeasures against negative load of oil hydraulic motor
(Measures against runaway)

When a negative loads is imposed on an oil hydraulic motor. (in case an oil hydraulic motor is rotated by self-weight of load) a counterbalance valve (back pressure valve) must be installed on the tank return side as shown in the figure below.

Thus, a runaway rotation of an oil hydraulic motor can be prevented.

15. Basic calculation equation (related to the pump)

1) Actual discharge

\[ Q = \frac{q_P \times N_P \times \eta_v}{1000} \text{ l/min} \]

Where,  
- \( q_P \): Displacement (cc/rev)  
- \( N_P \): Number of revolution (rpm)  
- \( \eta_v \): Volumetric efficiency  
- \( \eta_P \): Pump total efficiency

16. Basic calculation equation (related to the oil hydraulic motor)

1) Required flow

\[ Q_o = \frac{q_o \times N_o}{1000 \eta_v} \text{ l/min} \]

2) Output torque

\[ T_o = \frac{(P_1 - P_2) \times q_o \times \eta_t}{200\pi} \text{ (kgf-m)} \]

3) Output

\[ L_o = \frac{N_o \times T_o}{975} \text{ (kW)} \]
\[
\text{Output Torque (PS)} = \frac{N_o \times T_o}{716}
\]

Where, 
- \(q_o\) : Displacement (cc/rev)
- \(N_o\) : Number of revolutions (rpm)
- \(\eta_v\) : Volumetric efficiency
- \(\eta_t\) : Torque efficiency, mechanical efficiency (To be replaced with a starting torque efficiency at a full load start.)

### 2.4. Structure and working principle of oil hydraulic motor

The oil hydraulic motor receives fluid discharged from the pump and rotates the drive shaft. In other words, this oil hydraulic motor converts liquid energy from the pump into mechanical rotational energy.

#### 2.4.1. Gear motor

The gear motor has almost the same structure as that of the gear pump mentioned above. It differs in the following points.

1. Since it is rotated in both directions (clockwise and counterclockwise), the pressure on each side of the connection port becomes high. Therefore, the internal oil leak into the oil seal is returned to the tank with a pipe called an external drainpipe.

2. Both connection ports are of the same size. (In case of the pump, the suction opening has generally a larger bore than the discharge opening in order to prevent a phenomenon in which a cavity occurs due to an inadequacy or deterioration in suction conditions called cavitations.)

![Diagram of gear motor](image)

**Fig. 33** shows the pressure exerted on the tooth flank of a gear and the output torque occurrence condition.
2.4.2. Axial piston motor

The axial piston motor has almost the same structure as that of the piston pump mentioned above. It differs in the following points.

1) Both connection ports are of the same size.
2) The size and number (two in case of the pump, and four in case of the motor) of small notch grooves provided on the port plate to control noise are different.

Figure 34 shows the pressure exerted on the piston and the output torque occurrence condition. In this figure, the one in which the angle of inclination cannot be changed is of a fixed displacement type. For a structural drawing, refer to Figure 29 (P.33), the same figure as that of the pump.

The motor in Figure 35 is a variable displacement motor. Since it can be used at the angle of inclination from 25 to 7 degrees, the change gear ratio of the oil hydraulic motor is approximately 3.5 times (25÷7=3.5) although the pump is of a fixed displacement type.

It can be used at a high speed of rotation when the load is light (low torque) and at a low speed of rotation when the load is heavy (high torque).

The angle of inclination of convex-shaped port plate (1) is controlled by adjusting piston (2) moved oil-hydraulic, manually, etc.
2.4.3. Radial piston motor (star motor)

If the oil hydraulic motors described so far are used at 800 to 1,500 rpm, it is an effective use. On the other hand, the radial motor (star motor) shown in Figure 36 is used at a low speed of rotation of 10 to 200 rpm and a large type is available so that a large output torque can be obtained. The radial motor is used for a low-speed rotation driver such as an axle of a bogie and construction vehicle, winch drum, truck mixer agitator, etc.

2.4.4. Cylinder and rotary motor

The cylinder is the oil hydraulic equipment to convert fluid energy sent by the pump into mechanical energy, and performs rectilinear motion such as pulling and pushing.

Output (F) of the cylinder is obtained from a product of load pressure (P) and pressure receiving area of cylinder (A).

\[ F = P \times A \]

This output (F) is constant from the starting point of a stroke to the end point. Velocity of cylinder motion (V) is determined by inflow of fluid within a certain unit time and pressure receiving area of the cylinder (A).
The rotary motor is the same as the cylinder and oil hydraulic motor. It is the oil hydraulic equipment to convert fluid energy sent by the pump into mechanical energy, but it is an actuator whose output axis rotates only the limited amount of rotation such as $2/3$ ($120^\circ$) to rotations.

3.1. Types and features

1) Single acting cylinder (ram type)
   Fluid flowed in from port A moves the piston to the right, but external force ($\leftarrow$) is required to return the piston.

2) Single acting cylinder (with a spring)
   The piston pushed out by fluid is returned to the original position by the force of a spring.

3) Double acting cylinder (single rod type)
   If fluid is supplied from port A, the cylinder is pushed, and if fluid is supplied from port B, it is pulled. In this case, there are two types of pressure receiving area (A) to be used in calculating cylinder output (F).
   1) Pressure receiving area on the head : Entire area on the left of the piston
   2) Pressure receiving area on the rod : Area of annular portion on the right of piston
   This means that pushing force is greater than pulling force on condition that the pressure is the same and that pushing speed is lower than pulling speed on condition that the inflow rate is the same.
4) Double acting cylinder (double rod type)
Since the piston rod of the same diameter is provided on both sides, the pressure receiving area is the same on both sides, and if the pressure and flow are the same, the output and speed are the same in both directions. Also, the pressure does not increase on the tank return side even in a meter-out control circuit.

![Diagram of double acting cylinder](image)

5) Telescope type cylinder
This is a cylinder used when the installing space is narrow but a large stroke is required. When fluid is supplied from port A, two pistons are pushed out one after another. The piston with a wider pressure receiving area starts to move first. For this reason, although the load and supply flow are the same, the generated pressure and speed differ depending on which piston is moving.

![Diagram of telescope type cylinder](image)

**Rotary motor**
This type of cylinder, which is most widely used, can be roughly classified into the following two groups.

1) Tie-rod type cylinder
As shown in Figure 40, that this tie-rod type cylinder refers to a model in which head cover 1 and rod cover 3 are fixed to cylinder tube 2 with four tie rod bolts shown in the photograph below.

![Diagram of tie-rod type cylinder](image)
The cylinder is composed of piston 4, bush 6, and fixing flange 7. It is provided with piston packing 10 to prevent an oil leak between head 8 and rod 9 and with rod packing to prevent oil leak to the outside between piston rod 5 and bush 6. The minimum and maximum speeds of the cylinder are up to the material of the packing and face roughness of a portion in contact with the packing.

The minimum speed is determined by the limit where a constant speed cannot be obtained anymore which is called a stick slip phenomenon or chatter. It is generally 8 mm/sec.

The maximum speed is determined by durability of packing, and is generally 200 to 400 mm/sec. The commercially available one as a JIS standard product has the following specifications.

Rated pressure: 35, 70, 140, 210 kg/cm²
Inner diameter of cylinder: 32 to 250 mm

2) Screw type structural cylinder, weld type structural cylinder
In this type of cylinder, a cylinder tube is mounted to a head cover or rod cover by means of welding or screw as shown in the following photograph and Figure 41. The one shown in Figure 41 is a cylinder designed for a high pressure of 350 kg/cm² and is sturdy.
Spool 2 mounted on the piston moves right and left by the force of fluid. The rack mounted on spool 2 rotates pinion gear 3 and rotates the output shaft within a certain speed of rotation. The torque of the output shaft is determined in proportion to a working pressure, and the rotational speed is determined by a supplied flow. The piston stroke is limited by adjusting screw 4, and the number of revolutions of the output shaft is determined.
This is used to open/close a large-size door or a marine hatches cover, which are usually used at rotation of 360 degrees or less. The product shown in the photograph is used at a rocking angle of 180 degrees, maximum torque of 2,600 kgf·m, and rated pressure of 160 kgf/cm².

**Precautions in handling cylinder**

1. **Model of rod**

   Among the JIS conforming articles, two types of piston rod, standard type (also called type C) and strong type (also called type B) which is slightly thicker, are available on the market? The strong type has the following features, compared to the standard type.
   1) The buckling strength increases because of a thick rod.
   2) The pulling speed is easily increased because the pressure receiving area on the rod side decreases.
   3) The output (pulling side) decreases because the pressure receiving area on the rod side decreases.

2. **Restriction on maximum stroke**

   Considering the bucking strength of the rod, the rod can be prolonged in case of a low load or in case of a high load by pulling. However, note that manufacturer as shown in Table below restricts the maximum stroke of a standard article.

<table>
<thead>
<tr>
<th>Inner diameter of cylinder (mm)</th>
<th>40.50</th>
<th>63.80</th>
<th>100~250</th>
</tr>
</thead>
<tbody>
<tr>
<td>Maximum stroke (mm)</td>
<td>1,200</td>
<td>1,600</td>
<td>2,000</td>
</tr>
</tbody>
</table>

3. **Position of port, cushioning mechanism, and air bleeder**

   Sufficient workspace is necessary to perform the piping of ports, adjustment of cushioning mechanism, and air bleeding. Note that the model indication of JIS conforming articles has a specification of these positions (4 directional positions of top bottom left and right).

4. **Minimum working pressure and performance (efficiency)**

   In the JIS conforming cylinders, the upper limit of pressure at which the cylinder is actuated by a pressure from the head is specified by the shape of a packing and the nominal pressure, and it is approximately 2 kgf/cm² for general product available on the market. This is the minimum working pressure, and the cylinder should be actuated at this pressure or lower.

   The static friction resistance of a piston packing and rod packing corresponds to this pressure. Since it changes to the dynamic friction resistance after start-up, the minimum working pressure decreases to 0.2 to 0.3 kgf/cm².

   However, the mechanical efficiency of the cylinder is considered to be 100% as a matter of practicality because these pressures are very low compared to the working pressure. Volumetric efficiency $\eta_v$ is 100% because there is no leak from a piston packing or rod packing.
5. **Upper and lower limit of speed**
The minimum and maximum speeds are specified for the JIS conforming cylinders. For example, 8 to 400 mm/sec for the inner diameter of the cylinder of 32 to 63 mm. 8 to 300 mm/sec for that of 80 of to 125. 8 to 200 mm/sec for that of 140 to 250 mm. If both limits are exceeded, a cylinder is made to order because the shape, material, etc. of a packing must be changed. The maximum speed has such a problem that cylinder speed is “not uniform” (called stick slip) and what is known as “chatter” occurs.

6. **Air bleeding**
Air in the cylinder should be bleeder with an air vent valve provided on both rod and head sides at a trial run after installation. Otherwise, “inconsistencies” in cylinder motion may occur in proportion to the remaining quantity of air.

7. **Protective measures for piston rod**
Rust and flaws on the rod may damage a rod packing and cause an external oil leak. Take the following points into consideration as countermeasures against rust.
1) The rod end which is pulled in should be used for a longer time, and it should be exposed to air for a time as short as possible.
2) Material should be changed to stainless or the rod should be plated to take countermeasures against water, seawater, etc. To prevent flaws, the following measures are taken.
   1) Application of bellows (Figure 52)

![Figure 52 Protection of piston rod](image)

   2) If there is some problem in strength of bellows, make a protective cover with an iron plate or the like.
**Valve**

A valve is installed at an intermediate position between an actuator and a pump to control the magnitude of output, the direction of output, or the speed of output of the actuator or to control so that the pump makes no-load drive.

**Type of valve**

1) Classification according to the purposes of control

- Pressure control valve
  - Relief valve (safety valve) [direct operated type and balanced piston type]
  - Reducing valve [dito]
  - Sequence valve
  - Sequence valve [dito]
  - Counterbalance valve [dito]
  - Unloading valve [direction operated type and balanced piston type]
  - Unloading relief valve
  - Brake valve

- Directional control valve
  - Check valve
  - Check valve
  - Pilot operated check valve
  - Prefill valve
  - Shuttle valve
  - Directional control valve
  - Hand operated valve
  - Solenoid valve
  - Pilot operated valve with solenoid control
  - Pilot operated directional valve

- Flow control valve
  - Throttle check valve
  - Flow control valve
  - Deceleration valve
  - Flow dividing valve

2) Valves classified according to the shape
   - Stacked valve
   - Multiple control valve

3) Valves classified by the method of control
   - Remote control valve (direct operated relief valve), solenoid proportional valve, and serve valve.

4) Element valves to combine multifunctional combination valves logic valve (cartridge type two way valves)

5) Valves included in accessory
   - Stop valve (butterfly valve, gate valve), gage cock, pressure switch.

**Pressure control valve**

1) **Relief valve (Safety valve)**

Valve perceives the inlet pressure (A port). When pressure rise over a set pressure, the needle of valve moved up to decrease the hydraulic oil in high pressure back to tank (B port) as preventing the exceeding set-pressure.
In other words, this valve limits the upper limit of the pressure. When the pressure at the entrance becomes below the set pressure, the valve is closed.

Where this valve is used to control the output of an actuator, this valve is called a relief valve. Where this valve is used to prevent oil hydraulic units from damaging and is not opened except emergency, this valve is called a safety valve. The name of the valve changes depending upon its application.

When the valve is opened, the hydraulic oil (flow Q L/min) drops its pressure from a set pressure (P kg/cm²) to 0 kg/cm². At this time, the pressure energy, which has been lost changes to thermal energy, which raise the oil temperature.

\[
\text{Calorific value} = \frac{Q \times P}{612} \times 860 \quad \text{Kcal/hr}
\]

When the actuator has stopped, the delivery flow from the pump passes through this valve, and the output of the drive machine is all used for heating (in the case of a fixed delivery pump). The opening of the valve seat part changes depending upon the flow which passes there. As the flow is increased, the pressure at the A-port becomes higher somewhat.

Over-ride pressure characteristics represent this condition. The valves where the pressure at the A-port is as close to the cracking pressure as possible when the passing flow is increased may be called good valves.

1) Direct operated type relief valve

Fig. 58 indicates the basic diagram of this valve. A needle valve 5 is pressed against a valve seat 6 by a spring 3. The spring chamber is led to a tank (B port). Further, a pressure is introduced in the valve seat part from the A port to push up the needle valve 5 against the spring 3. Since the repulsion of the spring 3 can be adjusted freely by an adjustment screw 4, the pressure from the A port can be set at any point observing a pressure gauge.
An actual construction drawing is shown in Fig. 59. A cushion (damper) 7 is attached to the needle valve 5 to absorb the vibrations of the needle valve when it is opened. This valve is a direct operated type relief valve of a cartridge manifold block, etc. The bottom surface of the spring receiving part of the needle valve is actuated so that the valve seat part can be farther opened receiving the impulse force of the hydraulic oil when the valve is opened. Thus, the override pressure characteristics can be held better.

2) Balanced piston type relief valve
This valve has the following features when compared with the direct operated type valves:

1) Better override pressure characteristics.
2) Small and lightweight for large capacity valves.
3) Good stability for occurrence of chattering, tramp, etc. of valve.
For these reasons, this balanced piston type relief valve is mostly used except the cases where impact loading is applied or where used at a small flow below 10L/min.

Fig. 60 Balance piston type of relief valve

Fig. 60 shows the basic diagram of this valve, and the reference numeral 1 designates the direct operated relief valve described above. The pressure of the A-port acts on the bottom side of a main valve 2 and on the upper side of the main valve 2 and the needle valve part of the direct operated type relief valve through an orifice 3.

When the needle valve is not opened, the pressure balance is held. As shown in the figure, the main valve 2 is located at the starting position by the spring 4, and the A-port is separated from the B port by the valve seat part 5.

When the pressure in the A-port has increased and the direct operated type relief valve is opened, the hydraulic oil flows to the tank (B port) through the orifice 3, the needle valve, and the central hole of the main valve 2.

At this time, a pressure drop occurs in the orifice 3, the pressure balance between the upper surface and the bottom surface of main valve 2 is broken. The main valve 2 moves upward against the spring 4 to open the valve seat part 5, and the
hydraulic oil escapes to the tank through the A-port and the B-port so that the pressure in the A port do not exceed the set pressure.

As the pressure in the A-port is lowering, the needle valve part is closed, and the main valve 2 is pushed down by the force of the spring 4 to close the valve seat part 5.

![Fig. 61 structure of balance type of relief valve](image)

**Fig. 61**, the flange type, shows the actual internal construction of the basic diagram (**Fig. 60**).

![Fig. 62, the gasket type, shows a balanced piston type relief valves although the internal construction is different a little.](image)

**Fig. 62**, the gasket type, shows a balanced piston type relief valves although the internal construction is different a little.

The pressure in the A port in **Fig. 62** acts on the upper part (the chamber where the spring 4 exists) of the main valve through orifices (3.1), (3.2), the valve seat part of the needle valve (1), and the orifice (3.3).

The orifice (3.3) acts as a cushion to prevent the main valve from vibrating. The filter 6 prevents clogging of the orifice (3.2) due to large dust. In this valve, the hydraulic oil flowing after the needle valve has opened is connected with the B port (the internal drain system). Where a back-pressure is expected to occur in the B port, it is possible to switch to the external drain system.
The valve shown in Fig. 63 is a relief valve having an unload solenoid valve 3 fitted. The solenoid valve is connected immediately before the valve seat part of the needle valve. Even when the needle valve part is closed, the solenoid valve is excited. If the hydraulic oil flows into the solenoid valve, the upper part of the main valve becomes atmospheric pressure. As a result, the pressure in the A port becomes a low pressure to resist the spring of main valve having force of about 3 kgf/cm², and the hydraulic oil flows from the A-port to the B-port to become an unload condition.

The attached solenoid valve 3 includes a type, which opens to connect with the B port when excited, and a type, which is closed when excited. This valve is used where a pump is started under no load or where the loss of power is minimized while the machine is stopped. The valve shown in Fig. 64 is used for the same purpose as Fig. 63. The solenoid valve 3 is connected with the bent connection port (X port) of a flange type relief valve through external piping.

Fig. 63 Relief valve with unload solenoid valve

Fig. 64 Started under no load of relief valve with solenoid
In Fig. 64, a direct operated type relief valve 2 is further connected with the vent connection port. Therefore, even if the needle valve part of the main relief valve 1 is not opened, the main valve part of the main relief valve 1 is opened by the opening of the direct operated type relief valve 2.

This valve is used where the set pressure of a main relief valve 1 is required to change frequently but the main relief valve is located at a place somewhat apart (within about 6m) to feel inconvenience. By installing the direct operated type relief valve 2 closely, the inconvenience can be improved.

The direct operated type relief valve used in such a way is also called a remote control valve for the purpose of application. By making flow 1.5 to 2 L/min of hydraulic oil through the direct operated type relief valve 2 and the solenoid valve 3, the main relief valve 1 is full opened. Therefore, the smallest size is sufficient.

If it is desirable that the set pressure of the main relief valve 1 is 10 kgf/cm2 higher than the maximum working pressure of the direct operated type relief valve 2 for the safety in malfunction of the direct operated type relief valve 2.

3) Precautions on handling
a. Designation of valve size
The designation of valve size and the type indication method are determined depending upon the connection piping size.

<table>
<thead>
<tr>
<th>Nominal piping size = Nominal valve size</th>
<th>inch Designation</th>
<th>¼ in</th>
<th>3/8 in</th>
<th>1/2 in</th>
<th>⅛ in</th>
<th>⅛ in</th>
<th>1/4 in</th>
<th>1 in</th>
<th>1/2 in</th>
<th>2 in</th>
<th>⅔ in</th>
<th>2 in</th>
<th>⅓ in</th>
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<tbody>
<tr>
<td>B</td>
<td>B</td>
<td>1/4</td>
<td>3/8</td>
<td>1/2</td>
<td>⅛</td>
<td>⅛</td>
<td>1/4</td>
<td>1</td>
<td>1/2</td>
<td>2</td>
<td>⅔</td>
<td>2</td>
<td>⅔</td>
<td>3 ⅔</td>
<td>3</td>
<td>1/2</td>
</tr>
<tr>
<td>B</td>
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<tr>
<td>B</td>
<td>⅛</td>
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</tr>
<tr>
<td>5mm/6mm</td>
<td>5A / 6A</td>
<td>5A</td>
<td>5A</td>
<td>5A</td>
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<td>5A</td>
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<tr>
<td>5A / 6A</td>
<td>10</td>
<td>10</td>
<td>15</td>
<td>20</td>
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<td>100</td>
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<tr>
<td>mm Designation</td>
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<td>A</td>
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<td>A</td>
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<td>A</td>
<td>A</td>
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</tr>
<tr>
<td>Indication of the type of valve size</td>
<td>inch Designation</td>
<td>02</td>
<td>03</td>
<td>04</td>
<td>06</td>
<td>08</td>
<td>10</td>
<td>12</td>
<td>16</td>
<td>-</td>
<td>24</td>
<td>-</td>
<td>32</td>
<td>40</td>
<td>48</td>
<td></td>
</tr>
<tr>
<td>mm designation</td>
<td>5/6</td>
<td>10</td>
<td>15</td>
<td>20</td>
<td>25</td>
<td>32</td>
<td>40</td>
<td>50</td>
<td>65</td>
<td>80</td>
<td>90</td>
<td>100</td>
<td>125</td>
<td>150</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

The designation of valve size is like the piping size.
Example inch designation --- ½ in valve, ⅜” valve, ½ B valve
Example mm designation --- 15 mm valve, 15A valve

The indication of the type of valve size becomes as follows:
Example inch designation --- 04 (example of ½ in valve)
Example mm designation --- 15 (example of 15 mm valve)

The ½ in valve is expressed as 04 for the indication of type. This is because the denominator (8) is eliminated from ½ = 4/8 and 0 (zero) is added to the numerator (4).
In future, the designation will be united to the mm designation. At present, the mm designation is used half-and-half as can seen in our catalogue.

b. Allowable back pressure
Of course, the over-ride pressure characteristic becomes worse by the back pressure generated. From the viewpoint of the valve safety, the back pressure must be low as much as possible.
In the catalogue, the values of allowable back pressure are indicated from the strength, so care should be taken.

c. Minimum set pressure
In the case of the balanced piston type, an oil pressure to overcome the force of the spring at the main valve is necessary, so the minimum set pressure can not become zero. In the test run adjustment, care should be taken so that the cylinder does not move due to this pressure (residual pressure).

d. Maximum set pressure
In the case of the balanced piston type, the spring of the direct operated type relief valve, which has been incorporated, includes 2 to 4 types, so the maximum pressure which can be set is different according to the type of springs.

Existence of one spring having the strongest spring force may be sufficient, but the fine adjustments when low pressures are set becomes difficult, and the over-ride pressure characteristics becomes worse.

e. Flow rate
The rated of flow shown in the catalogue includes two types of expression, so the size of valve should be determined taking full care about.
There two types; to determine according to the piping size to be connected, and to determine from the critical flow which can ensure the functions of the valve. The farmer is determined from the flow speed in the pipe (3 to 4 m/sec), and the latter is an experimental value which is obtained only from the primary functions of the valve without considering the over-ride pressure characteristics and the piping resistance of the whole oil hydraulic circuit.
Of course, the latter is expressed as a larger flow, and the connection piping should be increased using reducers. Care should be taken that both of them are described in our catalogue.

f. Connection method
There are three types of connection methods as shown in Fig. 65. In the flange type, the valve can be suspended on the way of piping, and the valve body can be removed by removing the bolts in the flange part.
The gasket type is convenient to install on a manifold block. When installed separately, it can be mounted on a sub-plate.
The thread type can be suspended on the way of piping. The valve body has tapered screws, so the number of parts is small. However, with respect to the thread type, it should be noted that the tapered thread part requires a seal tape, that the tapered screw should not be tightened too much (from the viewpoint of strength), and that it is difficult to remove from the piping.
g. High vent type
As shown in Fig. 63 and 64, on a relief valve having an unloading solenoid valve attached, it takes generally about 2 seconds to change from an unloading condition to an on-load condition (valve is closed). When it is desired to become faster the on-load condition, it is sufficient to increase the force of the spring present in the main valve part. Generally, one more spring is added. The time for changing the condition is shortened to about 0.5 seconds. However, the minimum set pressure mentioned previously becomes higher since the force of the spring has been increased, and the pressure loss is also increased.

h. Over-ride pressure characteristics
As the passing flow (called relief flow) is increasing from the cracking pressure condition (valve opening start pressure), the pressure will gradually increase. The override characteristics indicate this relationship. Fig. 66 shows the difference in the characteristics between the direct operated type relief valve and the balanced piston type relief valve. The characteristics are also called “previous leak characteristics”. As can be seen from Fig. 66, the valve is opened somewhat earlier (at somewhat lower pressure) when viewed from the set pressure side, and the oil is leaked to the tank. When the pressure becomes close to the set pressure, the previous leak flow occurs, and the available flow, which is sent to the actuator is decreased, so the movement of the actuator is decelerated. For this reason, the balanced piston type relief valves, which have small previous leaks are used widely.
i. Wear of the needle valve

As shown in Fig. 67, the flow of oil strikes the needle valve to change the direction of the flow. Therefore, this area of the needle valve is worn earlier since the flow speed is higher for higher setting pressures.

Fig. 67 Further, due to the fine dust in the oil, the wear advances earlier. When the wear has advanced, the leaks this area is increased, and the set pressure is reduced. It is necessary to make a scheduled inspection of this area at least once every six months.
j. Chattering
The phenomenon that the needle valve strikes continuously the valve seat to generate a noise of “pee” is call chattering. If this phenomenon has occurred, on machine tools, streaks will be produced on the working faces or the tools may be broken. Further, the valve seat side is also struck to be deformed, requiring sometimes replacing the parts. As can be seen in Fig. 67, the needle valve is separated from the valve seat, being supported only by the spring. At this time, the needle valve and the spring generate resonance due to the pressure vibration (called pulsation of pressure caused by the pump) on the hydraulic oil source side. This condition is chattering. Chattering is apt to occur in the direct operated type relief valves. It occurs frequently in the high-pressure and high-temperature oil.

k. Surge pressure
If the relief valve does not open faster against a rapid pressure rise, a surge pressure is produced exceeding the setting pressure. In Fig. 68, the condition where a general surge pressure is produced is shown. In the balanced piston type, first, the needle valve part opens, and then, the main valve part opens, so the valve opening time is slow, and the surge pressure is high. For this reason, for the machines such as construction machines, vehicles, etc. where impulse loads are generated, the direct operated type relief valve is widely used.

![Surge pressure of relief valve](image)

<table>
<thead>
<tr>
<th></th>
<th>Direct operated type</th>
<th>Balanced piston type</th>
</tr>
</thead>
<tbody>
<tr>
<td>Time to be stable</td>
<td>0.02 sec.</td>
<td>0.10 sec.</td>
</tr>
<tr>
<td>Surge pressure (To the set pressure)</td>
<td>1~1.2 times</td>
<td>1.2~1.5 times</td>
</tr>
</tbody>
</table>

l. Minimum flow
The balanced piston type must not be used at a small flow (below 5 to 8% of the rated flow) which makes the set pressure unstable.
Reducing Valve

This valve is used to limit the pressure on the secondary side (the outlet side of valve).

When the pressure on the secondary side becomes a set pressure, it is held at the set pressure even if the pressure on the primary side (the entrance side of valve) becomes a higher pressure. Therefore, it becomes possible to use a part of the pressure on the oil hydraulic circuit after being reduced.

1) Direct operated type reducing valve

![Reducing valve diagram](image)

Fig. 69 Reducing valve

At the starting position, a spool 1 is pushed to the left side by a spring 3, and the primary side (B port) and the secondary side are opened. The hydraulic oil flows to an actuator through the B-port and the A-port. The secondary pressure of the A-port acts on the left end face of the spool 1 through a passage 4. When the set pressure which resists the spring 3 is reached, the spool 1 is moved to the right to close the passage passing from the B-port to the A-port.

As a result, the secondary pressure will not become a pressure above the set pressure.

The flow to be supplied to the secondary side is the flow which the actuator requires. If the actuator is stopped, this valve is full closed.

The valve shown in Fig. 69 has in common the function of a safety valve to open a passage from the A-port to the Y-port against. An abnormal high-pressure on the secondary circuit besides the function of a reducing valve. Reducing valves, which have in common no function of the safety valve, are widely used. If any conditions where the secondary circuit becomes a pressure exceeding. The set pressure can be considered. In the cases where an abnormal external force is applied to the actuator in the full closed condition, or where the secondary piping is heated from the outside to expand the hydraulic oil in the full closed condition, a safety valve (relief valve) must be installed separately.
To make the return oil from the actuator a free flow, a check valve 5 is incorporated.

The G-port is a port for installing a pressure gauge. If not necessary, apply a blind plug.

2) Balanced piston type reducing valve

The reducing valve of a gasket type shown in Fig. 70 and that of a flange type shown in Fig. 71 are both balanced piston type reducing valves, which are used in a greater flow compared with the direct operated type reducing valves.

Reference numeral 1 designates a direct operated type relief valve, which controls the opening and closing of the main piston part of a valve body 2. The main piston is pushed downward by a spring 8 so that the B-port is full opened to the A-port at the lowest end of the starting position.

The pressure in the A-port acts on the bottom of the main piston, and at the same time, it acts on a needle valve 7 and the top surface of the main piston through a passage 4 having orifices 5, 6.

When after the pressure in A-port has reached to a set pressure. The needle valve part is opened the hydraulic oil flows in the passage 4. A pressure drop occurs in the orifices 5, 6 and the main piston moves upward against the spring 8 to close the passage from the B-port. It does not follow the pressure in the B-port even if
it becomes a high pressure, and the pressure in the A-port does not become a pressure above the set pressure.

When the pressure in the B-port to the A-port. As a result, the pressure in the A-port does not follow the pressure in the B-port even if it becomes a high pressure, and the pressure in the A-port does not become a pressure above the set pressure.

When the pressure in the B-port is above the set pressure, the needle valve part is opened continuously, and a flow of about 1.5 to 2 l/min continues to flow from the drain port (Y) to the tank.

For this reason, an oil hydraulic circuit having many reducing valves installed requires an excess pump delivery flow by the flow corresponding to this.

Where a check valve is required to make the flow from the A-port to the B-port a free flow, an incorporated check valve is employed.

3. Precaution on handling
a. Determination of the set pressure
   The pressure on the secondary side to be set to a lower pressure is determined so that a difference above 10 kgf/cm² is produced against the pressure on primary side of a high pressure.

b. Surge pressure
   As can be seen from the internal structure, the primary side and the secondary side are opened at the starting position. For this reason, if the secondary pressure rises abruptly, the secondary pressure becomes a surge pressure exceeding the set pressure due to the delay of action.

   Generally, a surge pressure of about 20% is generated. As the difference between the primary pressure and the secondary pressure becomes larger, a higher surge pressure is generated. For this reason, the pressure gauge to be installed on the secondary side should be, on a conservative basis, a pressure gauge for high pressures being the same as that to be installed on the primary side.

c. Unsteadiness of set pressure
   Many of reducing valves are used in a condition near the full closed condition. Therefore, if the dust in the oil passes the reducing valves, the secondary pressure is apt to fluctuate. For this reason, by the use of a fine filter below 40 microns, the set pressure will be stabilized.

d. Wear of the needle valve
   In the case of the balanced piston type, the wear of the needle valve is the same as in the relief valves mentioned above.
   During the pressure is reduced, the needle valve continues to open. Therefore, it should be noted that the needle valve wears faster than in the case of relief valve.
e. Drain backpressure
   The drain pressure should be low as much as possible since the secondary pressures rises is the drain backpressure. Unlike the relief valves, the secondary side becomes a high pressure, so external drain is produced.

f. Remote control
   As in the case of relief valves, remote control becomes possible making use of the vent connection port (X-port).

**Sequence Valve**
A valve is available as a sequence valve, but its name changes to a relief valve, a sequence valve, a counterbalance valve or an unloading valve depending upon its use.

A sequence valve is classified into the valves mentioned above from the following factors: the introduction route of a pilot pressure to open or close the valve is connected in the valve of led from the outside of the valve; and when the drain oil leaking to the chamber where the adjustment spring exists is returned to a tank, the drain oil is connected to a port connected with the tank or, where such a port does not exist the drain oil is connected to the outside through separate piping.

The drain oil is connected to the outside through separate piping.

The table below shows the relation between the connection methods and the names of the valve.

**Table 11**

<table>
<thead>
<tr>
<th>Name of valve (Application)</th>
<th>Pilot connection</th>
<th>Drain connection</th>
<th>Dwg. Symbol</th>
</tr>
</thead>
<tbody>
<tr>
<td>Relief valve (For safety)</td>
<td>Internal</td>
<td>Internal</td>
<td><img src="image1" alt="Symbol" /></td>
</tr>
<tr>
<td>Sequence valve (For order)</td>
<td>Internal</td>
<td>External</td>
<td><img src="image2" alt="Symbol" /></td>
</tr>
<tr>
<td>Counter balance valve (Against back pressure)</td>
<td>Internal / External</td>
<td>Internal / External</td>
<td><img src="image3" alt="Symbol" /></td>
</tr>
<tr>
<td>Unload valve (Unload)</td>
<td>External</td>
<td>Internal</td>
<td><img src="image4" alt="Symbol" /></td>
</tr>
</tbody>
</table>

(Note 1) Showing external pilot and external drain.
The relief valve shown in the Table, whose over-ride pressure characteristics are worse, is not used since a special relief valve exists separately.

1) **Sequence valve**
The sequence valve is used to determine the sequence of the operations of actuators. This is used in a case where a first actuator acts and a second actuator begins to act when this sequence valve is opened after the pressure has risen at the end point of the action.

![Sequence valve diagram](image)

**Fig. 72** shows an example of use using 2 sequence valves. The sequence of the cylinders is as shown in the figure.

**Fig. 73** Sequence valve use to secure the pilot pressure for opening of a pilot operation check valve. (Resistance valve)

In case of a cylinder on the left side is used as clamp for a work, the clamp will not be loosen during the operation of cylinder on the right side.

In the example of use shown in **Fig. 73**, this valve is used to secure the pilot pressure for opening of a pilot operation check valve.

In this application, the sequence valve is also called a resistance valve for its purpose of use.

2) **Counterbalance valve (back pressure valve)**

![Counterbalance valve diagram](image)

**Fig. 74** Counterbalance valve
As shown in Fig. 74, a back pressure valve is mounted to a tank return line to prevent an actuator from being moved in an unlimited condition by external force (in the figure, a load drops at a high pressured by its own weight).

That is, this back pressure valve is used to generate a back pressure on the tank return line for preventing the actuator from running away due to the engine of external force.

In the cases of oil hydraulic motors, this valve is necessary in lifting and lowering of winch.

Fig. 74 indicates an internal pilot type. The purpose of the use can be achieved by either of them, but the setting pressure is different.

3) Unloading valve
When the external pilot pressure exceeds a set pressure pump becomes an unloading condition from a certain pressure. This circuit is generally called two-pressure control, or low-pressure great flow/high pressure small flow control.

This valve is used in pressing machines, etc. to feed at high speeds in an unloading condition and feed at low speeds in a high loading condition, resulting in a reduced capacity of motor.
4) Direct operated type sequence valve
In a gasket in Fig. 77, the pressure in the A-port is led to a small spool part through an orifice in a main spool 2. By the pressure acting on the cross section of the small spool, the main spool 2 is moved to the left against the force of a spring 3. When it pressure exceeds a set pressure, the passage from the A-port to the B-port is opened.

If the maximum setting pressure is 25 kg/cm² or less, the small spool is removed, and the pressure acts on the whole cross section of the main spool.

Of the setting pressure is 150 to 210 kg/cm², a small spool is used and 2 springs are employed.

The X-port is an introduction port for external pilot. When using this, the orifice in the main spool 2 is replaced with a blind plug.

Further, the Y-port is a connection port for external drain. When using this, it is necessary to close the hole connecting between the chamber where the spring exists and the B-port. A check valve 5 incorporated permits the free flow from the B-port to the A-port.

Fig. 77 Sequence valve

Fig. 78 shows a flange type, the operating principle of which is the same as that for the valve in Fig. 77. The small spool is under a lower cover outside of the main spool 2, and the bottom surface receives the pilot pressure. As shown in Fig. 79, the valve can be changed to an internal connection type or an external connection type depending upon the fitting method of the lower cover. Further, the oil in the chamber where the spring 3 exists can be assembled to an internal drain type or an external drain type depending upon the fitting method of the upper cover.
5) **Balanced piston type sequence valve**

In Fig. 80, the pressure in the A-port acts through a passage 3 so that a pilot spool 4 is pushed toward the right. At the same time, the pressure reaches the bottom surface of a main piston 2, the top surface of the main piston through an orifice 5 in the main piston, and the pilot spool 4 through an orifice 7.

When the pressure in the A-port has exceeded a set pressure, the pilot spool 4 moves to the right against a spring 6, and the hydraulic oil coming through the orifice 7 is returned to the tank through a passage 8.

At this time, by the action of the orifices 5, 7, a pressure drop occur. By the pressure difference between the top surface and the bottom surface of the main piston, the main piston is pushed upward, and a flow is generated from the A-port to the B-port.

To permit a free flow of back flow from the B-port to the A-port, valves incorporating a check valve are available.

In Fig. 80, the internal pilot connection (X-port is a blind plug) and the external drain connection (drain connection from Y-port) are drawn with a large scale, and the other connection systems are also shown below.
Fig. 80 Balanced piston type sequence valve

6) Precautions on handling

a. Over-ride pressure characteristics
The characteristics are not so good, so special care should be taken in application according to the table of performance described on the catalogue.

b. Previous leak of the sequence valve
If the primary pressure before opening of the sequence valve is close to the pressure to be used on the secondary side, the actuator on the secondary side often starts to move before opening of the sequence valve due to the leaks in the valve.

c. Setting pressure of the back pressure valve (external pilot type)
The circuit efficiency becomes better as the pressure is lower. But, the setting pressure is set to about 20 to 30 kg/cm² since an unstable condition is apt to occur.

d. Special back pressure valve without internal leaks
The valves shown above are of a spool type, so oil leaks occur from the clearances of the valve body and the actuator is moved little by little by the external force. Therefore, the movement of actuator must be stopped by attaching a mechanical brake to the actuator or by the use of a pilot operated check valve. However, it is the best to prevent the oil leaks in the backpressure valve by itself.
In a direct operated type back pressure valve (external pilot and internal drain type) shown in Fig. 81, the part 6 of a main spool 2 prevent the oil from leaking together with a sheet-shape valve seat. Further, this valve is provided with orifices for damming at the small spool and the main spool 2, so loads can be lowered with out hunting in truck cranes or winches having frequent changes in loading. Since the setting pressure is not changed, the spring 3 has no adjustment screw.
Fig. 81 Special back pressure valve without internal leaks

**Directional control valve**
This valve can be classified into a check valve to prevent back flow and a directional control valve to change the direction of the flow of hydraulic oil.

**Check valve**
Check valves include a check valve, which has only the function to prevent back flow, a pilot operated check valve, which permits back flow under a certain condition, and a pre-filled valve. First, the check valve will be described here.

This valve to make the back flow impossible against a free flow in one direction has a structure as shown in Fig. 82. Since a valve and a valve seat make contact with a certain angle, valve opening without leaks becomes possible. As the valve, as shown in Fig. 82, a check plunger 1 is used, and a steel ball is sometimes used. The cracking pressure (valve opening start pressure) is determined by the strength of a spring 2, generally being over a range of 0.35 to 8 kgf/cm².

If the purpose is only the prevention of back flow, it is better that the passing resistance is smaller. Therefore, a spring of 0.35 to 0.5 kgf/cm² is used.

Spring of 3 to 8 kgf/cm² may be considered as a relief valve without pressure adjustment. They are used as the by pass in emergency for the filter or oil cooler provided on tank return circuits, or as the resistance valve.
The precautions on use are as follows:
1) Prevention of back flow is done, but not perfectly. Some oil leaks exist. When compared with the spool type valves, the leaks are very small.

2) Change of cracking pressure
   The structure permits no adjustments. By inserting a washer in the spring seat, some change becomes possible.
   If the valve becomes an unstable condition and the unstable condition to repeat valve opening and valve closing occurs, it is better to remove the spring or to change the cracking pressure by means of the washer.

Pilot operated checked valve

Fig. 83 Flange type of check valve Fig. 84 Gasket type of check valve

Fig. 83 shows a flange type, and Fig. 84 shows a gasket type. Both valves have a function to prevent back flow by means of a check plunger 1 and 2 in the same way as a check valve. Besides this, a pilot piston 4 which is actuated by a pilot pressure at a pilot connection port (X-port) pushes the check plunger 1, 2 to
release the prevention of back flow. The pilot piston 4 pushes, first, the small check plunger 2 to open a valve seat part (A2 part) in the large check plunger 1, and gradual decompression can be obtained even if the B-port is held at a high pressure. After the pressure in the B-port has lowered to some degree, a valve seat part (A1 part) of the large check plunger is opened, and a great flow flows from the B-port to the A-port.

In Fig. 85, the pilot pressure (Pst) necessary for X-port and the pressure generated in B-port can be obtained by the following formulas:

\[
Pst = P_1 \times \frac{A_1}{A_3} + C
\]

\[
P_1 = P \times \frac{A_K}{A_R} + C
\]

Where,
- \(A_1\): pressure bearing area of the large valve seat A1 (cm\(^2\))
- \(A_3\): pressure bearing area of pilot piston 4 (cm\(^2\))
- \(C\): spring force \(\div A_3\)
- \(A_K\): piston area of cylinder (pressure bearing area on the head side) (cm\(^2\))
- \(A_R\): annular area of cylinder (pressure bearing area on rod) (cm\(^2\))
- \(P\): pressure on the head side of cylinder (kgf/cm\(^2\))
- \(F\): load on cylinder (kgf), and
- \(A_2\): pressure bearing area of the small valve seat A2 (cm\(^2\)).

The oil hydraulic circuit diagram in Fig. 85 shows the use of the valves shown in Fig. 83, 84. When the hydraulic oil flows from the B-port to the A-port, the port is connected with a tank, so no pressure is generated in the A-port. The valves shown in Fig. 87, 88 are of an external drain type. As shown in the oil hydraulic circuit diagram in Fig. 86, these valves are used where a pressure (backpressure) is generated in the A-port when the hydraulic oil flows from the B-port to the A-port. Since the valves in Fig. 87, 88 have a drain port Y to be led separately to a tank being separated from the A-port, the pressure in the A-port acts only on the A4 surface of the pilot piston. Therefore, the pilot pressure (Pst) necessary becomes the following;

\[
Pst = \frac{P \times A_K - P_2 (A_1-A_4)}{A_3} + C
\]
Prefill valve

This valve can be called a large pilot check valve. The purpose of use is limited; that is, this valve is used to suck a great flow of hydraulic oil from a tank so that a large cylinder of a large pressing machine, etc. can be moved quickly regardless of the flow supplied by a pump.

This valve has an internal structure as shown in Fig. 89, being installed between an upper tank connected with the head side of the cylinder and the pressing machine as shown in Fig. 90.

A check plunger 2 is normally closed by a spring 3. When an external load W or an auxiliary cylinder pulls down the piston rod of a cylinder, the head side (Ak surface side) of the cylinder, that is the B-port side of the valve becomes a negative pressure to open the valve.

Thus, the hydraulic oil is sucked from the A-port to the B-port. As the cylinder rod is pulled down more rapidly, the negative pressure in the B-port becomes higher, and the suction flow from the tank becomes more.
At this time, the lowering speed of the cylinder is not related so much to the supply flows from the pump, being determined by the capacity (size) of this valve.

When the cylinder speed has lowered to a desired speed, a high pressure is generated on the head side of the cylinder by the supply flow from a pump connected with the P-port, and the check plunger 2 is closed.

The capacity of the pump can be determined from the cylinder speed at high pressure, and the pump becomes a small size. When the cylinder rod is pulled up after completion of the lowering step, a pilot piston 5 which is actuated by the pilot pressure to the X-port opens the check plunger 2, and the hydraulic oil on the head side of the cylinder is returned quickly to the tank.

The A-port is connected with the tank, so it is always of atmospheric pressure. This point is different from the pilot check valve mentioned above.

The A-port is connected with the tank, so it is always of atmospheric pressure. This point is differential pressure between A and B ports is basically 0.3 kgf/cm².

The size of the valve may be determined from this rated flow, or it may be determined after obtaining a piping size so that the suction speed becomes 0.8 m/sec or less.

If the capacity of the air bleeder is not sufficient, this valve will not act normally.

### 4.3.4. Types and Classification of Directional Control Valve (Table 12)

<table>
<thead>
<tr>
<th>Classification</th>
<th>Symbol</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>1. Number of ports (connection ports)</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td>2 port</td>
<td><img src="image1" alt="Symbol" /></td>
<td>Valve have 2 ports</td>
</tr>
<tr>
<td>3 port</td>
<td><img src="image2" alt="Symbol" /></td>
<td>Valve have 3 ports</td>
</tr>
<tr>
<td>4 port</td>
<td><img src="image3" alt="Symbol" /></td>
<td>Valve have 4 ports</td>
</tr>
<tr>
<td>Multi-port</td>
<td><img src="image4" alt="Symbol" /></td>
<td>Valve have 5 or more ports</td>
</tr>
<tr>
<td><strong>2. Number of positions to be switched</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td>2 position</td>
<td><img src="image5" alt="Symbol" /></td>
<td>Valve have 2 positions</td>
</tr>
<tr>
<td>3 position</td>
<td><img src="image6" alt="Symbol" /></td>
<td>Valve have 3 positions</td>
</tr>
<tr>
<td>Multi-position</td>
<td><img src="image7" alt="Symbol" /></td>
<td>Valve have 4 or more positions</td>
</tr>
<tr>
<td><strong>3. Shape of flow at central position</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td>All port block</td>
<td><img src="image8" alt="Symbol" /></td>
<td>Old name is closed center</td>
</tr>
<tr>
<td>PT connection</td>
<td><img src="image9" alt="Symbol" /></td>
<td>Called center bypass</td>
</tr>
</tbody>
</table>
### 4. Holding of switching position

<table>
<thead>
<tr>
<th>Switch Type</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>Spring offset</td>
<td>Without the switching operation force, returned to the home position by the force of spring. (2 position)</td>
</tr>
<tr>
<td>Spring center</td>
<td>Without the switching operation force, returned to the central position by the force spring. (3 position)</td>
</tr>
<tr>
<td>No spring</td>
<td>Without the switching operation force, the switching position becomes unstable.</td>
</tr>
<tr>
<td>Detent</td>
<td>Without the switching operation force, each position selected is held.</td>
</tr>
<tr>
<td>Pressure center</td>
<td>The central position is held by the pilot oil hydraulic force</td>
</tr>
</tbody>
</table>

### 5. Central system (operating system)

<table>
<thead>
<tr>
<th>System</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>Input system</td>
<td>Valve which is operated by input</td>
</tr>
<tr>
<td>Solenoid system</td>
<td>Valve which is operated by solenoid</td>
</tr>
<tr>
<td>External pilot system</td>
<td>Valve which is operated by pilot oil hydraulic force</td>
</tr>
<tr>
<td>Solenoid pilot system</td>
<td>Valve which is operated by solenoid and operated by controlled pilot oil hydraulic force</td>
</tr>
</tbody>
</table>
Hand Operated Valve

Fig. 91 Hand operated valve

The valve shown in Fig. 91 is a 4 port type (4 ports of P, T, A, and B), 3 position type (3 positions of \( \text{\(\uparrow\)} \), \( \text{\(\uparrow\)} \), \( \text{\(\uparrow\)} \),), all ports block type, spring center type, and manual type directional control valve. The symbols of each port represent the following;

- \( P \) = pressure port (pump port)
- \( T \) = tank port (may be called \( R \) = return port), and
- \( A, B \) = user port (actuator port).

In the figure, a lever stands vertically, the valve being at the central position \( \text{\(\uparrow\)} \). When the lever is moved manually to either the left or the right, a spool 4 in the body 1 is also moved to the left or the right so that each port is connected together or interrupted to make the flow of \( \text{\(\uparrow\)} \) or \( \text{\(\uparrow\)} \) occur and to cause the actuator to move.

The difference in the flow of \( \text{\(\uparrow\)} \) and \( \text{\(\uparrow\)} \) moves the actuator in the opposite directions (opposite rotation).

When the hand is released from the lever, the spool is returned to the central position to stop actuator.

Replacing the spool 4 with that of other shape can change the conditions of the flow between individual ports. The operation examples are shown in Fig. 92 to 94.

Fig. 92

Fig. 93
Fig. 94

Fig. 92 uses, as in the case of Fig. 91, an all ports block type directional control valve, Fig. 93 uses a PT connection type (center by pass type), and Fig. 94 uses a 4 port/2 position type directional control valve.

The precautions on using this valve are as follows:

1) **Spool symbols**
Each maker uses their own spool symbols, and there is no relationship between them.
For our products, care should be taken that the flange type and the gasket type use different spool symbols.

2) **Connection condition of each port on the way of switching**
After completion of the switching, as can be seen from Fig. 92 to 94, the connection condition of each port is determined by a mechanical stopping mechanism (stopper). At that time, the flows between individual ports are as shown in the figure. However, the flow between individual ports on the way of switching indicates a quite different flow not shown in the figure. The actuator sometimes shows an unexpected movement only for a short time.
In Fig. 95, the condition where the P-port is connected with the T-port is tried to be switched to the condition where the P-port is connected with the A-port by moving a spool to the right. Since the dimensions $X_1 < X_2$ in the figure, all ports are interrupted and separated from each other only for a short time on the way of switching.

At this time, a peak pressure is generated in the P-port, and its magnitude depends upon the switching time, the flow, and the set pressure of the relief valve.

However, in this valve, the actuator has a load, and even if any pressure occurs in the A-port, there is no risk that the actuator is moved independently by the load on the way of switching. Thus, the safety is ensured.
In Fig. 96, since the dimensions change to $X_1 > X_2$, all ports are connected to the T-port for only a short time before the P-port is separated from the T-port, and the switching can be made smoothly without occurrence of the peak pressure.

In Fig. 97, since the dimensions $X_1 = S_2$, cut off between the P-port and the T-port is made at the same time with the connection between the P-port and the A-port.

This valve is use principally in servo valves, since the fine movement of the spool correctly controls the direction of flow and flow rate.

Thus, over lap, under lap, and zero lap have been described. In the actual products, they must be selected understanding well their mechanism.

In our products, as shown in Fig. 98, the symbols on the way of switching are represented being enclosed with dotted lines.
3) Internal leak
In the example of Fig. 92, the valve is of all ports block in the center valve position. As can be seen from the internal structure, oil leaks through the gaps between the spool and the body. For this reason, if external force is applied to the actuator, the actuator will move from A to T or from B to T due to the oil leaks.

Further, when a directional control valve of all ports block is used in a parallel circuit, a cylinder, which is not used may begin to move. This is because the A-and B-ports becomes the same due to the leaks from P-->A and P-->B ports, and force is applied in the direction of expansion depending upon the difference in pressure bearing area.

To prevent this, as shown in Fig. 85 in Page 102, the use of a pilot operated check valve or the employment of a directional control valve of A, B, T connection type is necessary.

4) Passing resistance
The passing resistance for the flows between individual ports is described clearly in graphs in the catalogue. When using them, the sum of the passing resistance from P-port to A-port and the passing resistance from B-port to T-port is the increment of the load pressure of the pump. If the actuator is a cylinder, it must be noted that the flow of the tank return side is often larger than the inflow flow because of the difference in pressure bearing area.

Solenoid Valve
This valve is also called solenoid valve. Iron cores are moved by the excitation of an electromagnet incorporated, and a spool is moved by the force generated to switch the direction of flow between individual ports. By adding the characteristic s of electricity to the characteristics of oil hydraulics, it becomes possible to make remote control, automatic etc. Therefore, this valve is used widely.

According to the structure of electromagnet and the electric sources applied, there 4 types as follows:
- 1) Dry dc. solenoid valve
- 2) Dry ac. solenoid valve
- 3) Oil-immersed dc. solenoid valve
- 4) Oil-immersed dc. solenoid valve

In the dc. solenoid valve, even if the spool stops on the way of switching, the electromagnet will not generated heat, and the switching can be made smoothly. This valve is suited where the switching frequency is so often (the frequency of 250 times of switching per minute becomes possible, about twice the case of a.c.).

The feature of the ac. solenoid valve is that switching time is short (about 1.5 to 2 times the case of dc., 30 ms = 0.03 sec). If the spool does not move on the way of switching, the coil will be burnt (after lapse of 1 to 1.5 hours for the oil-immersed type, and 10 to 15 minutes for the dry type). The dry electromagnet has a simple structure, being low in cost. The feature of the oil-immersed electromagnet is a long switching life of 20,000,000 times, about 4 times that of the dry type. To compare both types more easily, the dry dc. electro-magnet 1 is shown on the left side of Fig. 99, and the dry dc. electro-magnet 2 is shown on
the right side of Fig. 99. In both types, the hydraulic oil from the T-port is adapted not to enter the magnet iron core part by a seal installed on a bush 3.

Further, since the 2 springs exercise no force on the spool, the spool stops on either side, which is pushed by the electromagnet to form a 2 position directional control valve.

In the next Fig. 100, the oil-immersed dc. electro-magnet 4 is shown on the left side, and the oil-immersed ac. electromagnet 5 is shown on the right side. The chamber of the movable iron core part is connected with the T-port.

Since the 2 springs 6 have force to return the spool toward the central position, it forms a 3-position directional control valve. Both Fig.99 and 100 have emergency manually operated pushbuttons 7, which are used also to confirm the switching function of the solenoid valve manually.

![Fig. 99 dry dc. electromagnet](image)

![Fig. 100 oil-immersed dc. electromagnet](image)

The precautions on the use of this valve are as follows;

1) Maximum flow for each spool
   The output of the magnet to push the spool for switching is about 4 to 5 kgf at the middle point of the stroke, being very weak force.
   The force for returning the spool to the central position by hydraulic force becomes larger depending upon the flow. Thus, the maximum flow is determined.
   The hydraulic force is different depending upon the shape of spool. Note the maximum flow for each spool stated on the catalogue.
   Further, the output of the magnet is different between a dc. magnet and an ac. magnet, so the maximum flow becomes different.

2) Power source
   Generally, according to ac. or dc., to the difference in voltage, or to the difference in frequency, the magnet must be changed.
In our products, for both WE6 type and WE10 type, 6 types are available as the standard:
- dc. magnet; for 12V, 24V
- ac. magnet (for both 50 Hz and 60 Hz)
- 110V, 200V, 220V

If the voltage is not used within a range of $\pm 10\%$ malfunction due to insufficient output or failure due to abnormal heat development in the coil will occur.

3) Waterproof and explosion-protection

**Pilot Operated Valve with Solenoid Control**
The solenoid valve mentioned above includes a product having maximum flow of about 100 l/min. If a larger valve than it is desired, the magnet becomes so large, and the price becomes very high. Therefore, the pilot operated valve with solenoid control is used, which combines a solenoid valve and a pilot operated directional valve.

![Fig. 102 Pilot operated valve with solenoid control](image)

In the valve shown in Fig. 102, the hydraulic oil introduced from the X-port, as introduction port for external pilot pressure, is led into a chamber 6 or 7 of a pilot operated directional valve 2 by the action of a solenoid valve 1. So that a main spool 3 is moved to switch the direction of a large flow.

The structure of the valve in Fig. 102 is called as follows:

- **External pilot type**
- **External drain type**
- **Spring center type**
- **All port block type**
- **With tarry valve type**
  - **External pilot type**
  - **External drain type**
  - **Spring center type**
  - **All port block type**
  - **With tarry valve type**
1) Pilot system
An oil hydraulic source is necessary to move the spool 3.

- Valve shown in Fig. 102 ---- External pilot system
- Valve shown in Fig. 104 ---- Internal pilot system

The difference between both systems is determined by separating the P-port from the X-port by changing the mounting condition of a seal pin 11 or by connecting them.

The inner pilot system requires no external oil hydraulic source, but care should be taken about the following:

- a) Even when the actuator moves under no load, the P-port must have a minimum pressure of 4.5 kgf/cm², which is necessary for switching. For this reason, it is necessary to incorporate a check valve shown in Fig. 103 (cracking pressure is about 4.5 kgf/cm²) in this valve. At this time, this portion raises the pump loading pressure.

![Diagram of internal pilot system]

- b) The upper limit of the pilot pressure is determined by the strength of the body side, so if the pressure in the P-port exceeds this pressure, it is necessary to incorporate a reducing valve (stacked type) between the solenoid valve 1 and the pilot operated directional valve 2.

- c) Some amount of oil is necessary to move the main spool 3, and the start of the actuator is delayed momentarily.

2) Drain system
Both Fig. 102 and 104 are of the external drain type by the Y-port, but they become the internal drain type by connecting with the T-port in the valve.

The internal drain type requires no external drain piping, but the following should be noted:

- a) Since the upper limit of the pressure in the T-port is determined by the strength of the body side, when using in a series circuit, a series circuit, some restriction is applied.

- b) The surge pressure in the T-port which is generated when switching this valve (a high pressure generated in a short time, being a transient pressure change returning to a specified pressure) may have some adverse effects on the switching of the main spool 3.

3) The spring center type and the pressure center type
The valve in Fig. 102 is of the spring center type. When the solenoid valve 1 is at the center valve position, the chambers 6 and 7 are connected with the Y port (a drain port connected with the tank), and the main spool 3 holds the center valve position or returns to the center valve position by the springs.

The valve shown in Fig. 104 is of the pressure center type. When the solenoid valve is at the center valve position, the pilot pressure is into the chambers 6, 7, and a centering bush 8. A centering rod 9 make the main spool 3 forcedly hold the center valve position or return to it by means of oil by hydraulic pressure.

When comparing the pressure center type with the spring center type, the following can be said:

a) Fast switching speed. (Take care that shocks are apt to occur.)

b) When the backpressure of drain port is high due to fat distance to the tank, the switching speed becomes faster, and the operation is made exactly.

c) In our products, a spring is provided at both end parts of the main spool 3. Even if the pilot pressure becomes zero due to pour interruption, etc., the main spool 3 can return to the center valve position. However, in the valves, which have no springs, care should be taken that the safety criterion for press, etc. becomes unacceptable.

d) Separate piping for the drain port L becomes necessary.

4) Types of spools
   As in the cases of the manual directional control valve and the solenoid valve, various types are available.

5) Tarry valve
   This valve, which is generally called a throttle check valve with check valve (double type), is installed between a solenoid valve 1 and a pilot operated directional valve 2 in Fig. 102.

By installing this valve, the pilot flow to the chamber 6 or 7 can be adjusted, so the moving speed of the main valve 3, that is, the switching speed can be controlled.
From this fact, the start of the actuator can be smoothly accelerated. Next, the operating procedure and the symbols will be described.

In the valve shown in Fig. 102, when the magnet on the right side of the solenoid valve 1, the spool of the solenoid valve is moved to the left side. The pilot pressure reaches the chamber 6, the main spool 3 is moved to the right against the spring and the flow rate of P→B port or A→T port is generated. Since the chamber 7 is connected with the drain port Y, the main spool 3 can be moved. The return to the center valve position was described above, so its description will be omitted.

The symbol of the valve in Fig. 102 is shown in Fig. 105, and the simplified symbol generally used is shown in Fig. 106. In the symbols, by the excitation of the magnet a or b, what a flow the pilot selector valve side is switched to is shown. The mark “0” shows the center valve position.

In the valve shown in Fig. 104, the operating procedure and the symbols are as follows:

When the right electromagnet of the solenoid valve is excited, the spool of the solenoid valve is moved to the left side, the chamber 6 receives the pilot pressure as it is, and the other chamber 7 is connected with the drain port Y so that its pressure becomes zero. The main spool 3 is pushed to the right side by the centering rod 9 to generate the flow of P→B port or A→T port. When this valve is used vertically, even if the pilot pressure becomes zero, the springs existing in the chambers 6, 7 ensure that the main spool is at the center valve position. When the electromagnet at the right side is demagnetized, the chambers 6, 7 become the pilot pressure. Since the pressure bearing area of the main spool 3 is larger than that of the centering rod 9, the main spool moves to the left. However, since the pressure bearing area of the sum of the centering rod 9 and the centering bush 8 is larger, the
main spool stops at the center valve position where the centering bush is stopped.

The port L is a drain port, which makes the pressure of the chamber between the main spool 3 and the centering bush 8 zero continuously.

The precautions on handling the pilot operated valve with solenoid control are as follows:

1) Switching speed
   The switching speed is the sum of the switching time of the solenoid valve 1 Fig. 102 and the switching time of the main spool 3. The volume required when the main spools 3 moves are clearly described as “pilot volume” in the catalogue. The pilot flow necessary is obtained by dividing the pilot volume by a time of switching to be desired.
   To make the switching speed faster, the pilot flow must be increased. However, it should be noted that the pilot pressure must also be increased since the passing resistance of the passages connecting with the chambers 6, 7 is also increased.

2) Pilot pressure source
   Besides the method of installing a check valve having a cracking pressure of 4.5 kgf/cm² on the inlet side of the P-port as shown in Fig. 103. There is a method of installing a check valve on the way of the piping on the tank return side from the T-port to secure the pilot pressure.
   In the latter case, the back pressure presses on the actuator becomes higher, so it should be noted that the actuator makes an abnormal action. In both cases, the pump loading pressure is increased by 4.5 kgf/cm², but the house power necessary for the pump becomes considerably larger since the pump discharge flow is large. It is necessary to compare the cost with the external pilot system where a small pump is installed separately.
**Pilot Operated Directional Valve**

The valve shown Fig. 109 is a 3 port / 4 position directional control valve. The spool 1 is moved to the right by an oil hydraulic pressure (or air pressure) exerting on a piston 2 on the left side which is led from the outside, being held in a detect 3.

![Fig. 109](image)

If the oil hydraulic pressure exerting on the piston 2 is removed, the switching condition will be held, but the oil hydraulic pressure acting on the piston from the right side will switch the valve. Since the switching speed is determined by the hydraulic supply flow led from the outside, it becomes possible to switch comparatively slowly. Further, it is possible to make remote control, which is one feature of this valve.

**Flow control valve**

This valve controls the passing flow in the valve step less. As a result, this valve is used to change the opening and closing speed of each valve, or to adjust the changing speed of the displacement of variable displacement pumps. The valve can be classified into a throttle check valve and a pressure compensated flow control valve.

**Throttle check valve**

Fig. 110 shows a high-pressure (300 kgf/cm²) PT screw type, both of which have a check valve 2.

![Fig. 110](image)
The flow is controlled by the size of the opening area of the throttling part 1, which can be adjusted from the outside. However, the flow from the B-port to the A-port becomes a free flow when a check valve has been provided.

The check valve incorporated in the valve shown in Fig. 111 uses a steel ball.

The flow passing the throttling part 1 can be represented by the equation shown below (this represents the passing flow of orifice and the case of choke is represented by another equation):

\[
Q = CA \sqrt{\frac{2g(P_A - P_B)}{\gamma}}
\]

\[
C = \frac{DH}{64i\nu} \sqrt{\frac{V}{\nu}}
\]

Where,

- \(A\) = opening area of the throttling part,
- \(\gamma\) = specific weight of hydraulic oil = 0.86 to 0.9 kg/L
- \(P_A, P_B\) = pressure \(P_A\) before the throttling part, pressure \(P_B\) after the throttling part,
- \(g\) = gravitational acceleration = 9.8 m/sec
- \(d_H\) = opening diameter of the throttling part (except a circle, obtained by the following formula)
  \[d_H = 4 \times \frac{A}{U}\]

Where,

- \(A\) = opening area of the throttling part,
- \(U\) = circumferential length of the opening part,
- \(i\) = length of the throttling part,
- \(\nu\) = kinematic viscosity, and
- \(V\) = Mean flow speed

B. The shape of throttling part determines the flow coefficient \(C\). Generally, being 0.6 to 0.9. When the flow coefficient \(C=0.7\) and the specific gravity of oil = 0.9 kg/L, the passing flow \(Q\) is show in Fig. 112.

C.
From the calculation formula mentioned valve, the flow of a throttle check valve increases as the pressure difference \((P_A - P_B)\) between after and before the throttling part increases. Further, as the length of the throttling part becomes shorter, the change in the kinematics viscosity \(\nu\), which is greatly affected by the temperature of oil, becomes un-remarked, and the change in setting flow becomes small. That is, if the temperature of oil rises and the viscosity becomes low, the flow coefficient \(C\) becomes larger, and the flow at the throttling part increases. In such a case, if the structure of the throttling part is formed of a thin orifice instead of a choke, the increment can be restricted.

Trying to calculation how the setting pressure changes depending upon the pressure difference between before and after the throttling part. As an example, the pressure \(P_A\) at the outlet changes from 0 kgf/cm\(^2\) to 100 kgf/cm\(^2\) depending upon the load of the actuator. At this time, the rate of flow change \(\beta\) becomes:

\[
\beta = \frac{C}{A} \sqrt{\frac{2 g (P_A - P'_B)}{\gamma}} = \sqrt{\frac{(P_A - P'_B)}{(P_A - P_B)}}
\]

\[
= \sqrt{\frac{(140 - 100)}{(140 - 0)}}
\]
That is, the flow becomes almost a half. If there is no problem about the rate of flow charge $\beta$ thus calculated, it is not necessary to use the expensive flow control valve (flow control valve) that will be mentioned next. Further, if a load is continuously applied to the actuator and the change of the load is small, and the use of a throttle check valve is sufficient.

**Flow control valve**

This is a valve incorporated with a pressure compensating mechanism to hold a constant setting pressure even if any pressure difference occurred between the entrance and outlet of a valve.

By the use of the pressure compensating mechanism, the rate of flow change can be controlled to $\pm 2$ to 4%. If more accuracy is required, an electro-hydraulic servo valve or a solenoid proportional flow control valve may be used.

To understand better the function of this valve, the oil hydraulic circuit diagram shown in Fig. 113 will be described first. The oil hydraulic circuit is used to adjust the rotating speed of an oil motor, being a flow regulating system by means of a meter-in circuit system.

The hydraulic oil discharged from the pump is divided into a flow, which the oil hydraulic motor requires (set by the flow control valve) and excess oil to escape from the relief valve.

This means that the relief valve is normally opened and the setting pressure of the relief valve is always the pressure $P_1$ at the entrance (A-port) of the flow control valve. Further, the pressure $P_3$ at the outlet (B-port) of the valve changes depending upon the magnitude of the load acting on the oil motor. In the valve, a throttling part 1 and a pressure compensating mechanism 2 are provided, and the pressure at the area connecting them is $P_2$.

The pressure compensating mechanism 2 consists of a compensator spool and a spring to reduce the pressure $P_1$ to the pressure $P_2$. The pressure $P_2$ acts on the upper surfaces $A_2$ of the compensator spool and the pressure $P_3$ acts on the lower surfaces $A_3$. The relation of the force acting on the compensator spool can be shown by the following equation:
\[ P_2 \times A_2 = P_3 \times A_3 + \text{spring force } F \]

If \( A_2 = A_3 \) and \( A_2, A_3 = A \), the equation can be converted to the following:

\[ P_2 - P_3 = \frac{F}{A} = \text{generally, 5 to 7 kg/cm}^2 \]

As can be seen from this equation, the pressure drop \((P_2-P_3)\) at the throttling part 1 is always corresponding to the spring force \( F \). Since the deflection of the spring is vary small, the spring force \( F \) may be considered as a constant valve. For example, if the load increases and the pressure \( P_3 \) rises, the compensator spool is pushed up to reduce the pressure reducing effect, and the pressure \( P_2 \) rises. As a result, both pressures rises become equal, and the compensator spool takes a balanced condition.

That is, even if the pressure \( P_3 \) changes to any degree due to the load change. The differential pressure \((P_2-P_3)\) before and after the throttling part is normally held constant. Therefore, the passing flow of the valve is kept at the setting flow.

To change the setting flow, the opening of the throttling part 1 may be changed. Fig. 114 shows the structure of an actual valve. The throttling part consists of a throttle pin 2 and a sleeve 3. The pressure compensating mechanism consists of a compensator spool 4 and a spring 5.

To make possible a free flow from the B-port to the A-port, a check valve 6 is attached. Therefore, the flow control operates effectively only for the flow from A-port to B-port.

The pressure \( P_2 \) before the throttling part acts on the end face of the compensator spool on the opposite side of the spring through a hole 7. The pressure \( P_3 \) on the outlet side after throttling part gives force to the compensating spool in the same direction as the spring force through a hole 8. A valve incorporated with a stroke limited 9 is available, which can control to the minimum a flow exceeding a setting flow instantaneous when the hydraulic oil begins to flow. The reason that the compensating spool 4 has moved to the stroke end by the force of the spring when the hydraulic oil does not flow. So that the area to be opened is full open, so the pressure \( P_2 \) exceeds temporarily the entrance pressure \( P_1 \) when the hydraulic oil abruptly begins to flow).
4.4.3. Flow control system

![Fig. 115 Meter-in system](image1) ![Fig. 116 Meter-out system](image2) ![Fig. 117 Bleed-off system](image3)

The features of the 3 systems are summarized in Table 13:

<table>
<thead>
<tr>
<th>Flow control system</th>
<th>Meter-in system</th>
<th>Meter-out system</th>
<th>Bleed-off system</th>
</tr>
</thead>
<tbody>
<tr>
<td>1. Speed control where actuator is moved by external force</td>
<td>Not permitted to use (runaway)</td>
<td>Permitted to use</td>
<td>Not permitted to use (runaway)</td>
</tr>
<tr>
<td>2. Speed control where load becomes abruptly absent</td>
<td>Not permitted to use (delay of follow may occur)</td>
<td>Permitted to use</td>
<td>Not permitted to use (delay of follow may occur)</td>
</tr>
<tr>
<td>3. Accuracy of speed control</td>
<td>Above 10 kg/cm²</td>
<td>Good</td>
<td>Good</td>
</tr>
<tr>
<td></td>
<td>Below 10 kg/cm²</td>
<td>Good (high pressure)</td>
<td>Good (high pressure)</td>
</tr>
<tr>
<td>4. Loading pressure of pump (When actuator is moving)</td>
<td>Relief valve setting pressure (Note 1)</td>
<td>Relief valve setting pressure (Note 1)</td>
<td>Pressure corresponding to load</td>
</tr>
</tbody>
</table>

(Note 1): The relief valve setting pressure is reached even when a load (external force) is absent.
(Note 2): When the actuator is a cylinder and the pressure control system is of the meter-out system. If the pressure bearing area on the head side of cylinder(A1), the pressure bearing area on the rod side of cylinder(A2), and the relief valve setting pressure(P1), the pressure PR on the rod side becomes.

1) At no load: \[ P_R = P_1 \times \frac{A_1}{A_2} \] ------- dangerous being pressurized
2) When positive load (F1 kgf) is applied and the rod is moving:

\[ P_R = \frac{P_1 \times A_1 - F_1}{A_2} \]

3) When a negative load (F2 kgf) is applied and the rod is moving:

\[ P_R = \frac{P_1 \times A_1 + F_2}{A_2} \]

4) At stroke end: PR = 0 kg/cm²

**Precaution on using Flow Control Valve**

1. Minimum working pressure difference
   This is the pressure (P2-P3) which resists the spring force of the spring 5 in Fig. 114. In a full opened condition where the minimum working pressure difference mentioned on the catalogue does not occur, care should be taken that the pressure compensating mechanism will not be actuated.

2. Minimum flow
   For the safety operation of the pressure compensating mechanism, it is better that the valve should not be used at a flow below 10% of the rated flow (the maximum flow).

3. Accuracy of flow control
   In our products, it is clearly stated on the catalogue that the pressure compensating accuracy and the oil temperature compensation accuracy are both ±2%. Consider that the sum of the both is ±4%. 
**Basic Design and Planning of hydraulic winch**

The processes of designing hydraulic winch are given by the hauling capacity (tensile strength) in ton and the number of hauling speed or winch velocity (m/min.). At first the diameter of winch is considered for this purpose the total length of rope should be determined to obtain above the radius. The total of rope wound up on to the winch at one layer can be calculated by the following formula.

\[ m = \frac{b}{d} \]

Where:

- \( m \) = The total number of rope wound up on the winch (turns)
- \( d \) = Diameter of rope (mm)
- \( b \) = Width of reel

The diameter of drum at N-layer is expressed by the following formula:

\[ D_{dn} = D_d + (2N - 1)d \]

The total length of wire rope for N-layer can be expressed as follows:

\[ L_{dn} = \frac{1}{1000} \pi m \left[ D_d + (2N - 1)d \right] \]

Therefore, the total length of rope in each layer from the first-layer to last-layer can be computes one by one.

Shaft torque of winch at the rating point is expressed by:

\[ T_d = 1000 r_d F \]

Where:

- \( D_d \) = Diameter of drum without rope. (mm).
- \( D_{dn} \) = Diameter of drum including with rope at the determine layer (mm).
- \( N \) = The number of rope wound up on the winch (turns).
- \( L_{dn} \) = The lengths of rope wound up on the winch at determine layer (m).
- \( r_d \) = The radius from the drum center to the rope center at determine layer (mm).
- \( F \) = Working force (Kg).
Calculation

1. \[ r = \left( \frac{1}{1000} \right) \times \frac{D}{2} \]

2. \[ T_d = rF \]

3. \[ N_d = \frac{V}{2\pi r} \]

4. \[ T_M = \frac{T_D}{R_w \times \eta_w} \]

5. \[ N_M = R_w N_0 \]

6. \[ \Delta P = \frac{Q_M}{\eta_{vp}} \]

7. \[ Q_M = \frac{Q_M N_M}{\eta_{vM}} \]

8. \[ P = \Delta P + \Delta P_c \]

9. \[ Q_p = \frac{Q_p}{\eta_{vp}} = \frac{Q_p}{N_p} \frac{N_p}{N_E R_E} \]

10. \[ L_H = \frac{Q_p P}{450 \eta_{mP} \eta_{mR}} \approx \frac{FV}{450 \eta} \]

Where:

- \( D \) = Diameter including ropes wound over at the rate point (mm)
- \( \eta_w \) = Mechanical efficiency of winch \( \approx 0.9 \)
- \( Q_m \) = Theoretical flow quantity of motor per one revolution (l/rev)
- \( \eta_{mM} \) = Mechanical efficiency of hydraulic motor \( \approx 0.8 – 0.9 \)
- \( \eta_{vM} \) = Volumetric efficiency of hydraulic motor \( \approx 0.85 – 0.95 \)
- \( \Delta P_c \) = Pressure loss of valves and pipes, etc. Kg/cm²
- \( \eta_{vp} \) = Volumetric efficiency of hydraulic pump \( \approx 0.85 – 0.95 \)
- \( \eta_{mP} \) = Mechanical efficiency of hydraulic pump \( \approx 0.9 – 0.95 \)
- \( Q_p \) = Theoretical flow quantity of pump per one revolution (l/rev)
- \( N_E \) = Engine speed which drive the hydraulic pump (rpm)
- \( R_E \) = Ratio of acceleration communicated from engine speed to hydraulic pump.
- \( \eta_{mR} \) = Mechanical efficiency of acceleration \( \approx 0.95 – 0.97 \)
- \( \eta_{total} \) = Total efficiency of hydraulic transmission
### Table of basic calculation formulas of hydraulic pump and motor

<table>
<thead>
<tr>
<th>Items</th>
<th>Unit</th>
<th>Symbol or calculation formula of</th>
</tr>
</thead>
<tbody>
<tr>
<td>Flow Quantity per rev.</td>
<td>l/rev.</td>
<td>Q</td>
</tr>
<tr>
<td>Effective oil pressure</td>
<td>Kg/cm²</td>
<td>ΔP</td>
</tr>
<tr>
<td>Revolution</td>
<td>rpm</td>
<td>N</td>
</tr>
<tr>
<td>Leakage</td>
<td>l/min</td>
<td>ΔQ</td>
</tr>
<tr>
<td>Loss of torque</td>
<td>Kg.m</td>
<td>ΔT</td>
</tr>
<tr>
<td>Theoretical flow quantity</td>
<td>l/min</td>
<td>Q&lt;sub&gt;th&lt;/sub&gt; = qN</td>
</tr>
<tr>
<td>Theoretical Hydraulic power</td>
<td>Hp</td>
<td>L&lt;sub&gt;in&lt;/sub&gt; = (Q&lt;sub&gt;th&lt;/sub&gt; x ΔP) / 450</td>
</tr>
<tr>
<td>Theoretical torque</td>
<td>Kg.m</td>
<td>T&lt;sub&gt;th&lt;/sub&gt; = (q x ΔP) / 0.2⇒</td>
</tr>
<tr>
<td>Actual flow quantity</td>
<td>l/min</td>
<td>Q&lt;sub&gt;p&lt;/sub&gt; = Q&lt;sub&gt;thp&lt;/sub&gt; - ΔQ&lt;sub&gt;p&lt;/sub&gt;</td>
</tr>
<tr>
<td>Actual torque</td>
<td>Kg.m</td>
<td>T&lt;sub&gt;p&lt;/sub&gt; = T&lt;sub&gt;thp&lt;/sub&gt; - ΔT&lt;sub&gt;p&lt;/sub&gt;</td>
</tr>
<tr>
<td>Volumetric efficiency</td>
<td></td>
<td>η&lt;sub&gt;v&lt;/sub&gt; = Q&lt;sub&gt;p&lt;/sub&gt; / Q&lt;sub&gt;thp&lt;/sub&gt;</td>
</tr>
<tr>
<td>Mechanical efficiency</td>
<td></td>
<td>η&lt;sub&gt;m&lt;/sub&gt; = T&lt;sub&gt;thp&lt;/sub&gt; / T&lt;sub&gt;p&lt;/sub&gt;</td>
</tr>
<tr>
<td>Total efficiency</td>
<td></td>
<td>η&lt;sub&gt;total&lt;/sub&gt; = η&lt;sub&gt;v&lt;/sub&gt; x η&lt;sub&gt;mp&lt;/sub&gt;</td>
</tr>
<tr>
<td></td>
<td></td>
<td>= Q&lt;sub&gt;p&lt;/sub&gt;ΔP&lt;sub&gt;p&lt;/sub&gt; / 0.2⇒N&lt;sub&gt;p&lt;/sub&gt;</td>
</tr>
</tbody>
</table>

#### Procedure of planning

1. Rated tension ( Kg) and winding speed V ( m/min)
2. Rate point r : Radius of drum at the point of application (m)
3. Drum shaft torque of winch T<sub>D</sub> (Kg.m)
4. Drum speed of winch N<sub>D</sub> (rpm)
5. Reduction gear ratio of hydraulic motor and winch drum (R<sub>w</sub>)
6. Hydraulic motor torque T<sub>M</sub> (Kg.m)
7. Hydraulic motor speed N (rpm)
8. Effective oil pressure of hydraulic motor ΔP (kg /cm²)
9. Necessary oil quantity of hydraulic motor Q<sub>m</sub> (l/min)
10. Delivery pressure of hydraulic pump P (Kg/cm²)
11. Driven speed of hydraulic pump N<sub>p</sub> (rpm)
12. Displacement of hydraulic pump Q<sub>p</sub> (l/min)
13. Input of hydraulic pump L<sub>H</sub> (Hp)
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