INDUSTRIAL FLUID POWER

VOLUME 3 – THIRD EDITION

Advanced Text on

HYDRAULICS, AIR, & VACUUM

for Industrial and Mobile Applications

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INDUSTRIAL FLUID POWER

Volume 3 - Third Edition

Covering Rotary Output of Fluid Power Systems

Prepared by CHARLES S. HEDGES
Assisted by the Technical Staff of Womack Educational Publications

Edited by ROBERT C. WOMACK
Member, Board of Directors, Fluid Power Educational Foundation
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Member, President's Council, National Fluid Power Association

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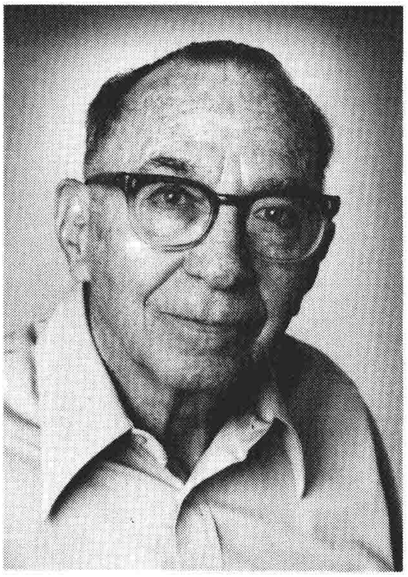

ABOUT THE AUTHOR

Mr. Charles S. Hedges holds an engineering degree from the University of Kansas. He has worked in industry in electrical and mechanical design, and now since 1953 in his association with Womack Machine Supply Company has written the Womack textbooks on fluid power, has worked with customers on a variety of fluid power applications, has prepared and published many design data sheets and technical bulletins, has taught fluid power classes in the larger cities of Texas, Oklahoma, and Louisiana, and has helped train new salesmen in fluid power applications and circuitry.

In this book, Volume 3 of the Industrial Fluid Power series, the topics covered are related to rotary motion (except pumps, which are covered in Volume 1). Subjects covered are hydraulic motors, air motors, rotary actuators, hydraulic (or hydrostatic) transmissions, rotary flow dividers, bootstrapping applications, power steering, and fluid powered vibrators. We suggest the other Womack books on fluid power be available for reference. See book listings inside rear cover.

This new edition of Volume 3 retains all material in the previous edition, with a new chapter on vibrators. An Appendix has been added with design information most appropriate to subjects covered in the book. Additional design data will be found in the Appendices of other Womack books.

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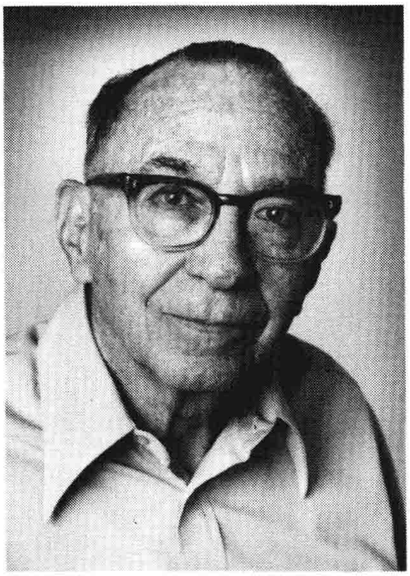

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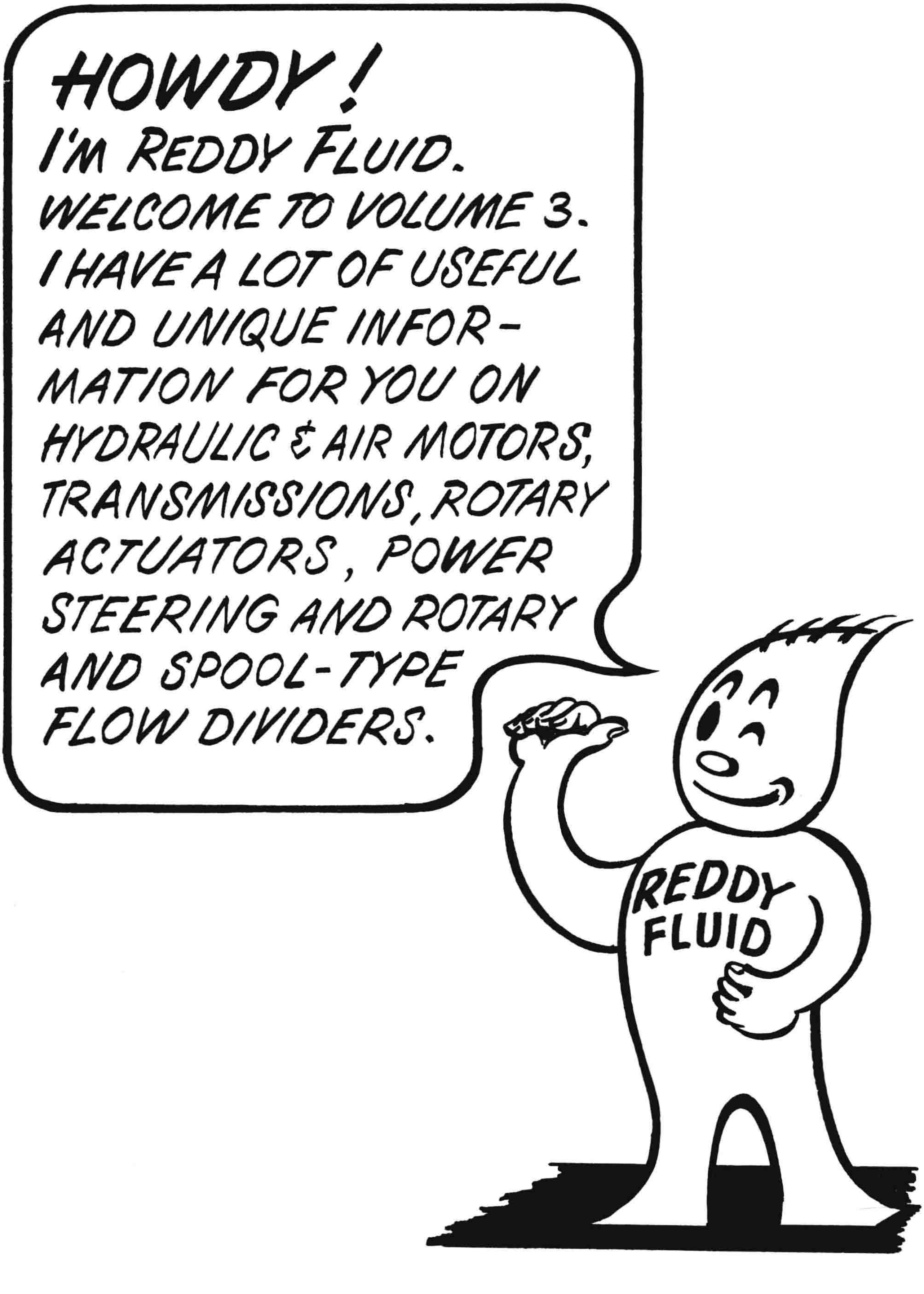

HOWDY! I'M REDDY FLUID. WELCOME TO VOLUME 3. I HAVE A LOT OF USEFUL AND UNIQUE INFORMATION FOR YOU ON HYDRAULIC & AIR MOTORS, TRANSMISSIONS, ROTARY ACTUATORS, POWER STEERING AND ROTARY AND SPOOL-TYPE FLOW DIVIDERS.
REDDY FLUID

Contents by Chapter

SUMMARY – A 4-line capsule summary of some of the more important topics in each chapter. A comprehensive index starts on Page 246. A section of design information and data tables starts on Page 229. Other design data can be found in the Appendices of other Womack books.

Introduction to Third Edition

by Mr. James I. Morgan, CAE. President, National Fluid Power Association

The modernization of the world's machinery has been made possible largely through the simultaneously expansion of the use of fluid power systems – systems that provide simple and reliable power and control for the machine's elements.

In the area of rotary power transmission, the subject of this text, fluid power systems easily provide drives with constant horsepower, constant torque and/or constant speed. Oscillating drives or those requiring "stall" for indefinite periods of time are also accomplished easily with fluid power. Achieving some of these drives with other means of transmission such as electrical or mechanical, may be difficult and sometimes almost impractical.

Although fluid power systems have been used industrially for more than a half century, the rapid expansion of applications of these systems to all forms of machinery has created an almost insatiable need for educational texts for self-study as well as for classroom use. This text, along with the other Womack fluid power texts, has filled a most timely need. The Womack method of presentation translates a complex engineering subject into a form easily understood by all those with mechanical aptitude and have the opportunity to work with fluid power operated machinery.

This text, and the others in the Womack fluid power series, together with other Womack efforts furthering fluid power education, were achievements on behalf of the fluid power industry which led to the conferment of the industry's highest honor, the National Fluid Power Association's Anniversary Award to the editor of the Womack publications, Mr. Robert C. Womack.

We are confident that after studying this text the reader will have a much fuller understanding of rotary fluid power systems and components – and a greater appreciation for the educational value of this and the other Womack texts.

1

Fluid Motor Types

The term "fluid motor" is normally used to identify those devices which deliver continuous rotary power output when supplied with flowing fluid under pressure. Most fluid motors are designed to operate on hydraulic fluids; a limited number of manufacturers offer motors for air or vacuum operation.

The term "motor" is used by some manufacturers in place of the more common term "cylinder" to denote a power device with straight-line reciprocating movement. Technically this is correct usage but may lead to a misunderstanding of the kind of actuator to be used. In this book the term "motor" will always denote a continuously rotating device. Cylinders are described along with their directional and speed control circuits in Volumes 1 and 2 of this textbook series.

Fluid motors are built in a wide range of shaft speeds, from 10 RPM (or less) to 5000 RPM (or more). Hydraulic motors have the same general appearance, size, and weight as hydraulic pumps of the same type and horsepower capacity, and operate on the same mechanical principles applied in reverse. In fact some devices could be used either as a pump or motor.

Rotary actuators are available from a limited number of suppliers either for compressed air or hydraulics. They are termed "oscillating motors". They produce an oscillating angular movement of less than one turn at low speed and high torque.

FLUID MOTORS COMPARED WITH FLUID POWER CYLINDERS

Whether to use a cylinder or a fluid motor as the output actuator depends to a great extent on the type of motion – straight-line or rotary – which can more readily or efficiently be applied to the type of load to be moved. On applications where either one could be used, cost is also a factor, a motor being more expensive than a cylinder operating at the same horsepower. Usually the advantage of one or the other will be immediately apparent.

In Volumes 1 and 2 of this textbook series, many diagrams were shown for cylinder operation. Some of these circuits could be applied to motor operation. However, fluid motors have certain characteristics which might prevent them from being substituted in cylinder circuits. The more important differences in action of cylinders and motors are these:

1. Slow Speed Operation. On the one hand, a cylinder which has leaktight piston seals will give consistent performance at very slow speeds against a wide range of load resistance. On the other hand, a hydraulic motor has metal-to-metal internal sealing surfaces between the motoring elements and these cannot be made leaktight. This prevents a motor from maintaining a consistent speed under changes in load. And especially at low speeds the internal leakage is a greater percentage of the total oil flow and definitely reduces the low speed limit. Limitations on low speed operation are explained in detail in later chapters.

2. Speed Range. A cylinder which has leaktight piston seals can be operated over a wide range of speed, and under changing load, with reasonably consistent performance. But the practical range over which a hydraulic motor can be operated successfully depends primarily on its rate of internal leakage. Motors built to ordinary tolerances may have a high leakage rate and must be limited to a moderate range of speeds. Those motors built to very close tolerances have a lower internal leakage and can be successfully operated over a wider range of speeds. The effects of internal leakage are covered in later chapters.

3. Hydraulic Shock. When stopped suddenly by blocking the oil flow, hydraulic motors tend to generate high pressure spikes in the hydraulic system – much higher than normally produced in a cylinder system. This is because the momentum energy contained in a rapidly rotating load is usually much greater than in a load moving in a straight line at relatively low speed at the same horsepower. This requires a more careful consideration as to how this momentum energy can be dissipated safely.

4. Load Holding. On any fluid power application, whether cylinder or motor, the system should not be designed with the intention of supporting a weight load or a reactionary load in a stationary position for an extended period by locking trapped oil inside the cylinder or motor. Internal leakage plus leakage in associated valving will allow the load to drift. However, for limited periods of time, a hydraulic cylinder may be able to support a weight or reactionary load if it has leaktight piston seals and is operated through virtually leaktight valving. But the greater internal leakage in a hydraulic motor makes it impractical to support a load without significant drift. A mechanical lock or brake may be required.

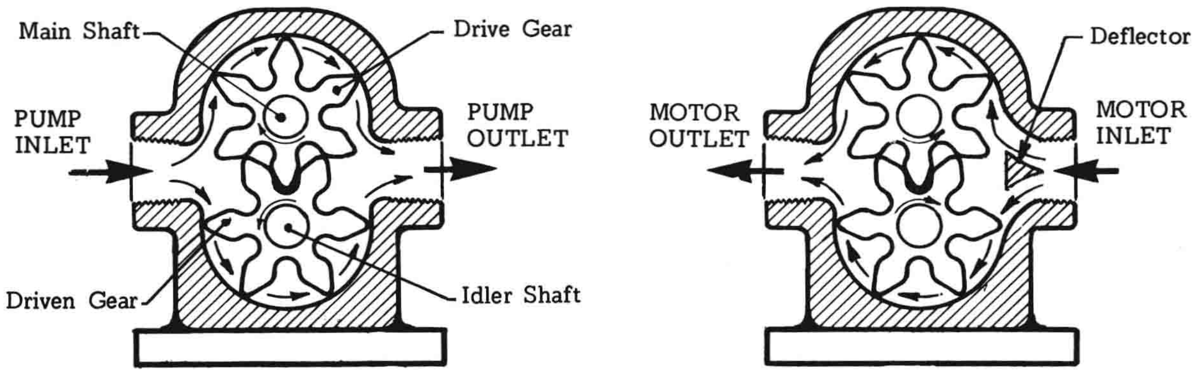
5. Efficiency. Losses caused by mechanical friction and fluid flow can be held to about 5% in a well-constructed cylinder. But in a motor or pump these losses will amount to 10 to 25% of the input power. But in spite of these higher losses, advantages to be gained by using a motor, will sometimes be worth the loss of efficiency. All power losses in a cylinder or motor end up as heat in the oil, so the addition of a heat exchanger is more likely to be required in a motor system.

HYDRAULIC MOTORS COMPARED TO HYDRAULIC PUMPS

Hydraulic pumps and motors are rotary devices for converting mechanical energy to fluid power energy or vice versa. Pumps accept mechanical power input and convert to an equivalent amount (except for internal leakage losses) of fluid power in the form of fluid flow under pressure. Motors work the other way. They accept fluid power and convert into an equivalent amount (except for internal leakage losses) of mechanical power output.

The following illustrations will show that most of the principles used in building pumps are usable in reverse for building motors. One important difference is that pumps are usually built for a specific direction of rotation, clockwise (CW) or counter clockwise (CCW), and internal draining of the leakage is arranged accordingly. But most motors are required to reverse rotation or must handle high pressure on both ports at the same time even when designed for a single direction of rotation. This usually requires additional arrangements for disposing of internal leakage, and may rule out the use of a pump as a motor on some applications.

As a supplement to better understanding the principles of various kinds of motors described in this chapter, the student may want to compare the description of similar pumps described in Volume 1.



(A). Hydraulic Pump *(B). Hydraulic Motor*

FIGURE 1-1. Comparison of Gear-Type Hydraulic Pump With Similar Type Hydraulic Motor

GEAR PRINCIPLE

Figure 1-1. In the gear-type pump pictured above, one gear (usually the top one) is rotated by a power source, usually an electric motor or an engine. This gear meshes with and rotates the lower gear. Shaft rotation in the direction marked on the drawing (CW) produces a vacuum at the pump inlet which draws fluid into the pump case. The oil is carried around the outside of both gears, being confined in tooth spaces, and is discharged from the outlet port when the gears mesh. Power is imparted to the fluid by force and movement of the gear teeth.

If the same device were to be used as a hydraulic motor, the high pressure inlet oil should be connected to the port which is designed to operate at high pressure. This is the port which is the outlet for operation as a pump. Since the oil flow is now reversed, the gears would rotate in the opposite direction (CCW). So as a motor the same unit should be used in reverse rotation with the pump outlet becoming the motor inlet.

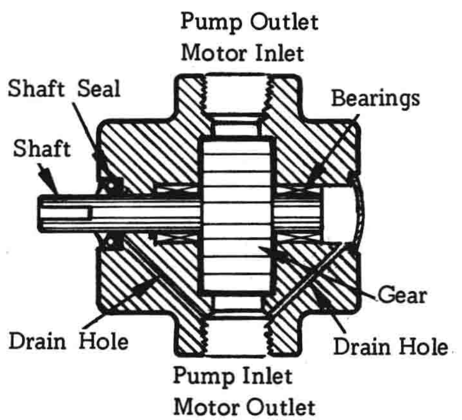


FIGURE 1-2. By-Pass Drain Passages.

Torque is produced by oil pressure exerted against the face area of the gear teeth. Movement of the fluid in rotating the gear teeth also produces shaft rotation. In the particular pump/motor model illustrated, the efficiency as a motor can be improved by adding a deflector to reduce turbulence in the incoming oil stream and to shield the gear teeth from velocity impact of the inlet stream which would produce an unwanted counter torque.

Internal By-Pass. Figure 1-2. In any gear device a small part of the high pressure oil will slip through clearances between the gear sides and the housing, and will collect in the seal and bearing pockets. These pockets, if not vented, would soon be exposed to the same high pressure appearing on the outlet port. To prevent this accumulation from dislodging the shaft seal, small holes can be drilled through the body connecting the pockets with the port which is the inlet as a pump or the outlet as a motor. A pump or motor drained in this way can be used in only one direction of rotation.

Reversible Rotation . . .

On most hydraulic motor applications it is necessary to have a motor which can accept high pressure on either port. The three methods of construction most often used to keep from dislodging the shaft seal are:

1. External Drain. Figure 1-3. The seal and bearing cavities can be internally joined and vented to tank through a drain port located on the housing. This is a preferred mode of construction for gear, vane, and gerotor-type motors for bi-rotational service.

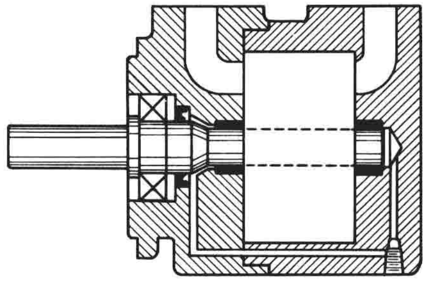


FIGURE 1-3. Externally Drained Motor.

2. Internal Check Valves. Figure 1-4. Hydraulic motors can be built with internal check valves which allow accumulated leakage oil in bearing pockets to flow to either main port, the one which is at the lower pressure. These check valves prevent backflow of high pressure oil into the pockets. The external drain is eliminated and these motors can be used on reversible applications where high pressure is never on both ports at the same time.

The check valve method of draining leakage is not acceptable on applications where high pressure may appear on both ports, and this happens when restrictions such as a counterbalance valve is placed in the outlet line to prevent overrunning by the load.

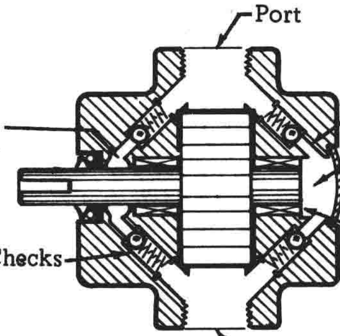


FIGURE 1-4. Internal Check Valves.

3. High Pressure Shaft Seal. Figure 1-5. Motors can be built with a face-type shaft seal which will not be dislodged by having full pressure on both main ports at the same time. This is a mechanical seal in which two lapped surfaces run against each other to seal the shaft. The rotating surface is usually bronze or carbon graphite, and the stationary surface is steel. The two surfaces are spring loaded toward each other, and are held in tight contact by internal hydraulic pressure.

Pump or motor operation is simplified by the use of high pressure face seals because no drain line is required and the motor can be used in any kind of circuit without danger of dislodging seals.

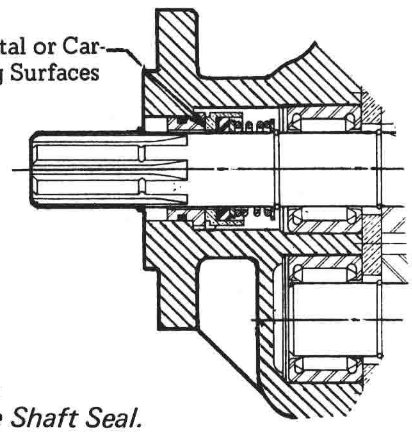


FIGURE 1-5. High Pressure Shaft Seal.

The major disadvantage of face-type high pressure seals is that they are not leaktight. Lubrication for the sealing surfaces is provided by the hydraulic oil, and it is inevitable that a small amount of "weepage" may wipe off to the outside. On mobile applications an occasional drop of external leakage is not objectionable in view of the other advantages, but is not acceptable on most industrial applications.

Any two lapped surfaces running against each other are sensitive to fine abrasives. Although the leakage oil which finds its way into seal pockets has been filtered, unless very fine filtration is used on the system, abrasive particles may lodge between the lapped surfaces and eventually cause the external leakage to increase to an objectionable level.

For these reasons, the high pressure face-type seal is acceptable only for mobile pumps.

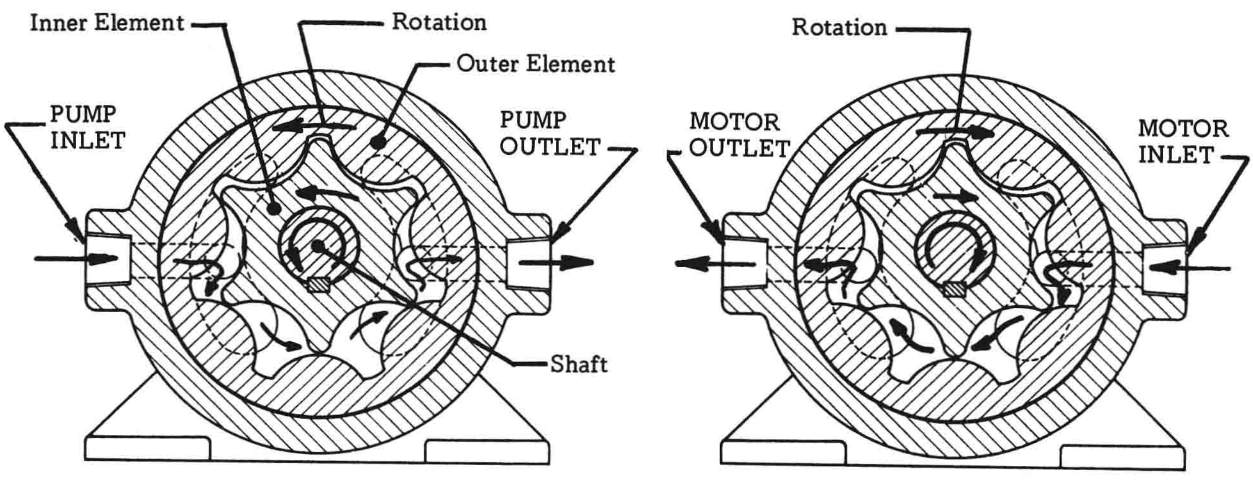
GEROTOR PRINCIPLE

Figure 1-6. A set of "gerotor" elements mesh together and rotate in a slip fit inside a housing, the inner element being keyed to the shaft. They are machined to close tolerances, so the tips of the teeth on the inner element follow the contours of the cavities in the outer element, keeping in tight sliding contact at all times. The inner and outer elements come into full mesh at top dead center, and are completely out of mesh at bottom dead center. Gerotor devices are built with one less tooth on the inner element, which causes the cavities to open and close as each tooth on the inner element progresses from one cavity to the next.

In the pump illustration, on the left, the opening of cavities between the two elements as they are rotated by the shaft, creates a vacuum during the first half revolution from top dead center, and this pulls oil into the inlet, through kidney-shaped openings in the side plate. As each tooth on the inner element passes bottom dead center it starts to move into the adjacent cavity, imparting pressure to the trapped oil, forcing it through another kidney-shaped opening and discharging it from the outlet port.

In the motor illustration, on the right, the action is reversed. Fluid pressure of the oil entering the inlet forces the cavities to become larger; that is, causes the gerotor elements to rotate in the direction which will allow the cavities to become larger to accept the incoming flow of oil.

Internal Leakage. A small amount of oil will slip, under pressure, down the sides of the elements and collect in the bearing and seal pockets. This leakage must be handled in the same way already



(A). Hydraulic Pump. *(B). Hydraulic Motor*

FIGURE 1-6. Comparison of Gerotor-Type Hydraulic Pump and Motor.

described for gear-type motors. To produce a bi-rotational pump or a reversible motor, either external draining, internal drain check valves, or high pressure shaft seals must be employed.

Direction of Rotation. A gerotor device, if designed with the simple internal drains as shown in Figure 1-2, can be operated in only one rotation. The port into which the leakage is directed must be the low pressure port, that is, the inlet as a pump or the outlet as a motor. As in a gear motor, rotation as a motor will be opposite to that as a pump.

We recommend the use of a reversible motor on all applications whether the rotation will be in one direction only or in both directions. Even on single rotation applications pressure spikes can be produced in the outlet while the motor is decelerating with a momentum load.

VANE PRINCIPLE

Figure 1-7. Sliding vane rotary devices work with good efficiency as motors as well as pumps. The illustrations show a balanced construction of a pump and a motor in which the cam ring is machined with two lobes, causing each vane to make two cycles on each revolution of the shaft. Mechanical side loads, caused by hydraulic pressure on the outer surface of the rotor, balance each other on opposite sides of the shaft. The motor or pump can be built with much smaller bearings than used for a gear or vane motor. The only side loading on bearings comes from external loading, if any. Life expectancy of bearings is increased and the motor can accept higher pressure to produce more torque by using the two-lobe construction. Torque produced by both lobes is combined on the shaft.

When this device is used as a motor, pressurized oil enters the inlet port and is routed through internal passages to both lobes in parallel. Torque is produced by leverage action of hydraulic pressure working against vane exposure, multiplied by the average radius to the shaft centerline. Shaft RPM is, of course, determined by the rate of inlet oil flow and by the displacement volume of the device. Like gear and gerotor-type devices, the port used as inlet for pump operation becomes the outlet port for motor operation, and shaft rotation is opposite.

Leakage oil accumulates in bearing and seal pockets, and like the gear and gerotor-type devices, must be handled by the same means.

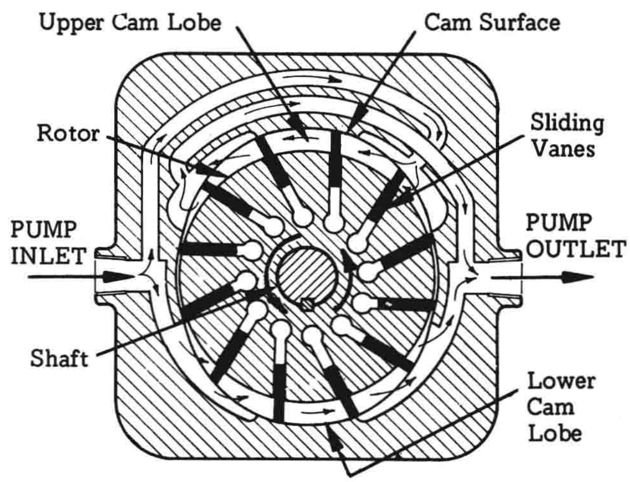


FIGURE 1-7(A). Vane-Type Hydraulic Pump.

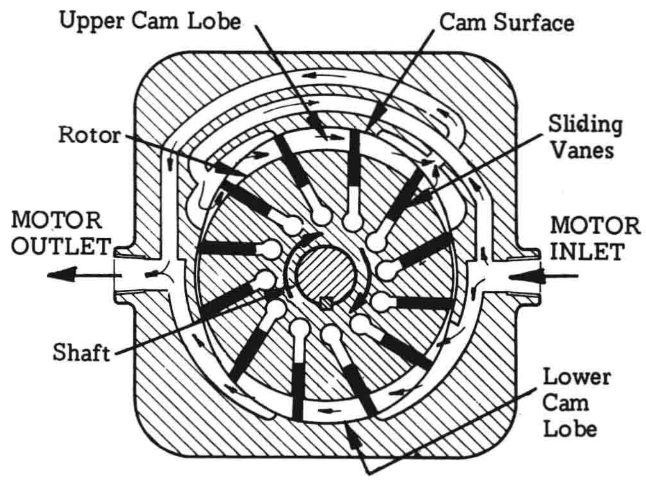


FIGURE 1-7(B). Vane-Type Hydraulic Motor.

When the device is used as a pump, the vanes will extend by centrifugal force when the shaft is rotated, although this force is often supplemented either by springs or hydraulic pressure behind each vane. When used as a motor, centrifugal force is not effective until a speed of 500 to 1000 RPM has been reached. To make the motor self-starting, a forcible means must be used to extend the vanes. Forcible means include springs under each vane or hydraulic pressure developed inside the motor by a "sequence plate" installed in series with the outlet. This plate introduces enough restriction to create a pressure sufficient to extend the vanes but it does add a power loss to the motor.

The vane principle is employed in the manufacture of low power air motors operating on air pressure up to 100 PSI.

PISTON MOTORS

Construction of piston motors is similar to that of piston pumps, sometimes with minor modifications. In fact, some piston pumps will operate as motors without any modification. An odd number of pistons is employed in piston pumps and motors, 3, 5, 7, 9, etc., to reduce the amplitude of ripple in the output. On motors this also gives a higher starting torque and permits the motor to operate more smoothly at very low speeds.

Leakage. There is internal leakage in all piston pumps and motors – oil which slips past the pistons and accumulates in the case. This leakage ***must*** be drained to the reservoir through a low resistance drain line. If not drained, pressure will build up inside the case and dislodge the shaft seal. During operation, pressure spikes could build up when external valving is shifted or when a load is suddenly applied. If drain line resistance is too high these spikes could damage the shaft seal.

Installation. When installing any kind of piston equipment the manufacturers instructions should be followed carefully because some brands may be restricted as to mounting position, drain port elevation, or pre-filling suction lines or case before start-up. The grade and viscosity of the oil may be critical on some models or brands.

Displacement. Piston motors are built with either fixed or variable displacement. In the case of variable displacement motors, decreasing the displacement will increase speed and décrease torque in the same proportion, while the horsepower output remains constant. Caution! Decreasing displacement too much may cause the motor to run at a destructive speed. The cam plate on a variable motor is never shifted across center to reverse the motor.

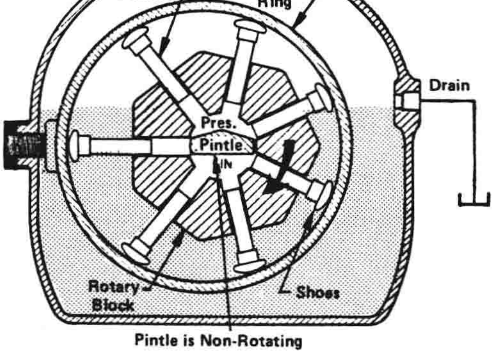


FIGURE 1-8. Radial Piston Motor.

Radial Piston Motor. Figure 1-8. The description of this device as a hydraulic pump, in Volume 1 will help the student understand its operation. Oil is ported to and from the pistons through longitudinal holes in the shaft which terminate in a twin rotary joint (not shown) which connect it to inlet or outlet ports in the motor case. The stationary pintle is positioned so it times the oil flow to and from the pistons, feeding inlet oil under pressure to those pistons which are "on stroke", that is, moving out, and collects discharge oil from those pistons which are exhausting. Hydraulic pressure forces some of the pistons against the reactor ring which is positioned eccentrically to the shaft centerline, causing the rotary block carrying the pistons to rotate. The greater the eccentricity of reactor ring to rotary block, the greater the motor torque, the slower the speed, and the higher the efficiency. On motors with variable displacement, the displacement should not be reduced so far that destructive overspeeding could occur. The reactor ring on a piston motor is never shifted across center to reverse motor rotation.

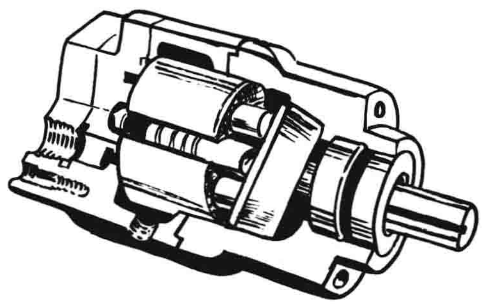


FIGURE 1-9. Axial Piston Motor.

Axial Piston Motor. Figure 1-9. The action of this device as a pump can be studied in Volume 1. As a motor, the pistons, under pressure, push against a circular ramp, usually called a "cam plate", to produce rotary motion on the shaft. The motor will produce equal torque and speed in either rotation, and can operate with high pressure on either or both ports at the same time. It can be built with fixed or variable displacement.

COMPARISON OF HYDRAULIC MOTOR CHARACTERISTICS

Some of the information in this chart was adapted from information published by Machine Design magazine. It was supplemented with information from other sources. Values in the chart are not necessarily the design limits of each type, but we believe are fairly representative of models now on the market. They should be taken as typical of products available from several sources. Some manu-

	Gear Type	Gerotor Type	Orbit Type	Radial Vane	Radial Piston Hi-Speed	Radial Piston Lo-Speed	Axial Piston Straight	Axial Piston Bent
Maximum Continuous Pressure, PSI	3000	2000	1500	2500	3000	3000	2500-5000	2500
Displacement, Cu.In./Revolution	20	5	57	12	28	415	39	44
Maximum Theoretical Torque, Inch Lbs.	6000	1500	10,000	4000	14,000	190,000	15,000	17,500
Starting Torque, In.Lbs., % of Theoretical	70%	70%	68%	70%	82%	80%	72%	80%
Running Torque In.Lbs. % of Theoretical	90%	87%	85%	90%	93%	93%	93%	93%
Continuous Speed Range, RPM	500-3000	500-5000	12-1000	500-4000	500-2000	2 - 600	500-4500	500-4500
Maximum Continuous Power, HP	200	100	90	140	250	270	360	250
Leakage, % of Theoretical Displacement	8%	10%	15%	2½%	3%	3½%	2½%	2½%
Weight-to-Power Ratio, Lbs. Weight per HP	0.2	0.2	1.5	0.3	5	4	0.7	1.2