



Applications Manual

Distributed by:



800-257-1155 • FAX 440-238-5266
Email: csdept@kraftfluid.com

Section 4

Transmission Circuit

Recommendations

Table of Contents

- 4.1 Transmission Circuit Components 3
 - 4.1.1 Introduction 3
 - 4.1.2 Charge Pump 4
 - 4.1.3 Controls 5
 - 4.1.4 Reservoir 6
 - 4.1.5 Conduit Sizing 7
 - 4.1.6 Case Drain Requirements 8
 - 4.1.7 Heat Exchangers 8
 - 4.1.8 Filters 9
 - 4.1.9 Loop Flushing Shuttle 12
 - 4.1.10 Pressure Controls 13
- 4.2 Components for Special Functions 15
 - 4.2.1 Flow Dividers / Combiners 15
 - 4.2.2 Flow Limiters 15
 - 4.2.3 Counterbalance Valves 16
 - 4.2.4 Accumulators 16
 - 4.2.5 Cylinders 17
- 4.3 Remote Variable Pressure Limiter 18
 - 4.3.1 Introduction 18
 - 4.3.2 Circuit 18
 - 4.3.3 Component and Circuit Performance 20
 - 4.3.4 Applications 22
- 4.4 Wheel Assist 23
 - 4.4.1 Description 23
- 4.5 Ganged Pumps 25
 - 4.5.1 Introduction 25
 - 4.5.2 Circuit Recommendations 25
- 4.6 Case Drain Routing Variations 27
- 4.7 One Charge Supply / Two Pumps 28
- 4.8 Charge From Implement Return Flow 29
- 4.9 Multiple Motors 30
- 4.10 Flow Dividers and Divider / Combiners 31
 - 4.10.1 Introduction 31
 - 4.10.2 Heat Generation 31
 - 4.10.3 Design Flow Range 31
 - 4.10.4 Loop Flushing 32
 - 4.10.5 Motor Life Reduction 32
 - 4.10.6 Differential Flow for Steering 33
 - 4.10.7 Motor Speed Synchronization (Non-Propel) 33
 - 4.10.8 Recommendations for Trouble-Free Circuit 33

4.1 Transmission Circuit Components

4.1.1 Introduction

Transmission performance and component life are functions of the hydraulic circuit design. Circuit design affects the ability of the various elements to control system pressure, fluid temperature and other factors relating to the quality of the fluid. The circuit elements will interact with each other to produce either desirable or undesirable results, depending on their arrangement in the circuit.

Circuit elements must be present to remove undesirable heat and contamination and to control the passage of cool, clean fluid into the main power loop. Other circuit elements are present to control pressure and, sometimes, to control flow.

A typical closed circuit schematic is shown in Figure 4-1. The closed circuit piston pump is normally factory equipped with a charge pump, charge pressure relief valve, and a control. It may also be equipped with high pressure protection devices such as relief valves (shown) or pressure limiters. A loop flushing shuttle valve can be integral to the pump or motor, or may be installed externally. Other components such as the reservoir, heat exchanger, and filter are added by the user.

The purpose of this article is to advise Sauer-Sundstrand users on generally accepted circuit arrangements, but it does not, and cannot, describe all possible circuits which may be appropriate for various applications. Transmission circuits are unique designs and must be developed and tested to ensure required performance and life.

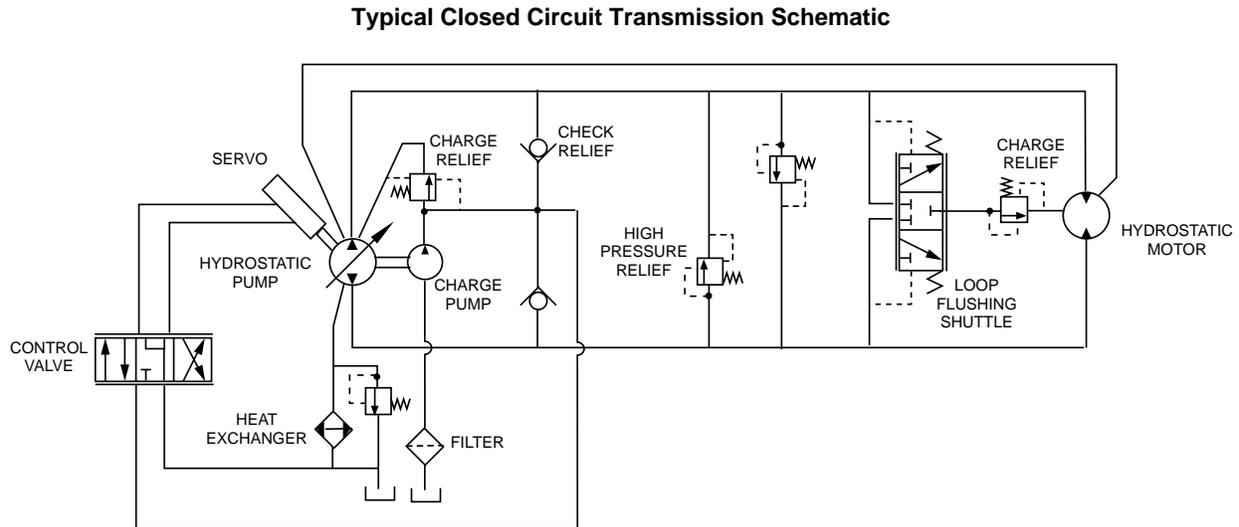


Figure 4-1

4.1.2 Charge Pump

A charge pump is absolutely necessary in a closed circuit hydrostatic circuit. This article lists the various functions of the charge pump, while Section 1.4 of this applications manual describes charge pump sizing considerations.

The charge pump functions associated with its volume flow rate are:

- replenish loop fluid lost through volumetric inefficiency of the pump and motor(s).
- replenish loop fluid lost through the loop flushing valve.
- provide make-up fluid in the loop for load-induced bulk modulus effects.
- provide flow to activate the servo control piston (on units with servo controls).
- provide constant flow to the Electric Displacement Control (EDC) for proper operation (on units with EDC's).
- provide a flow source for auxiliary circuit functions such as releasing parking brakes or shifting motor displacement.

The charge pump functions associated with its charge pressure are:

- maintain correct low loop pressure to ensure sufficient "hold down" forces on the rotating groups.
- provide sufficient pressure to activate the servo control system (on units with servo controls).
- control loop flushing flow by the differential pressure between the charge relief valve and the loop flushing relief valve.
- provide a pressure source for auxiliary circuit functions such as parking brake release.

All of these charge flow and charge pressure requirements must be accommodated in each mode of operation or the system will experience poor performance or component damage. Sauer-Sundstrand highly recommends that each application be scrutinized for sufficient charge flow and charge pressure to ensure proper function in all modes of operation.

Most Sauer-Sundstrand hydrostatic pumps are offered with integral charge pumps sized for typical installation requirements. Additional or alternate charge flow sources can be utilized. A gear pump mounted in tandem with the hydrostatic pump is often a clean installation.

4.1.3 Controls

The output flow of axial-piston hydrostatic pumps is controlled by varying the swashplate angle per some operator command. The two basic types of displacement controls are direct displacement control and servo control.

Direct displacement controls simply use a mechanical lever attached to the swashplate upon which the operator exerts a force, possibly through a linkage assembly. The force required to engage a direct displacement control is a function of the system pressure that the swashplate is working against. There is no neutral deadband or swashplate feedback in a direct displacement control. Because the required control force is a function of system pressure, direct displacement controls are generally only suitable to light duty products.

Servo controls are an interface between the operator and the swashplate. The operator input is a low force device which ports charge pressure to the servo piston which moves the swashplate. Thus, the operator input is not dependent on system pressure. Servo controls have a neutral deadband. Standard servo controls also have a mechanical feedback mechanism which ensures that the swashplate position stays where commanded. Several Sauer-Sundstrand products are available with “non-feedback” servo controls which are appropriate for certain applications. The three types of operator inputs for servo controls are manual, electric, and hydraulic.

4.1.4 Reservoir

In addition to providing the charge pump inlet with a constant source of fluid, the reservoir also helps remove air from the system and cool the fluid. The reservoir also provides make-up fluid for volume changes associated with fluid expansion or contraction, possible cylinder flow, and minor leakage.

The reservoir should be designed to accommodate fluid volume changes for all modes of operation and to promote de-aeration of the fluid as it passes through the tank. The total reservoir volume (in gallons) should be between 1/2 and 1-1/2 times maximum charge pump flow (in GPM) for closed circuit applications. The fluid volume should be approximately 80% of the total reservoir capacity. A large reservoir promotes de-aeration better than a small one. However, excessively large reservoirs may contribute to difficulties with fluid temperature and contamination control.

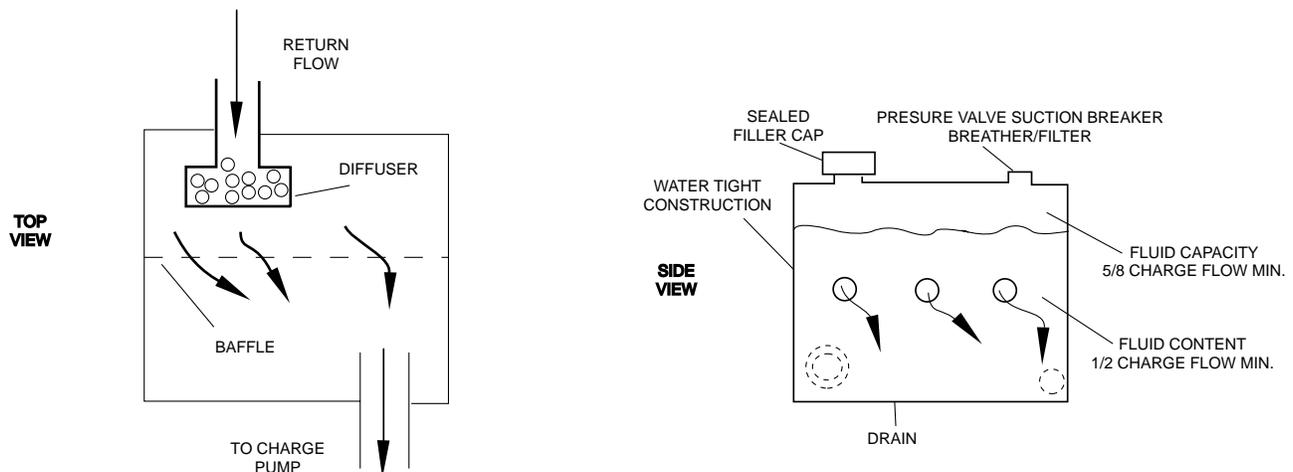
The reservoir may be pressurized or air breathing. Pressurizing with a low pressure relief valve helps to keep out dust and dirt as well as to prevent air leaks in the charge pump suction line. However, the amount of entrained air in the fluid increases with pressurized reservoirs, and the pressure should be limited to 10 psig. A suction breaker relief valve is required for pressurized reservoirs. In either case, an air breather filter with adequate protection from environmental contamination must be used.

Water contamination is harmful to transmission components, and reservoirs must be designed to prevent the entrance of water into the reservoir. Since many machines must withstand high pressure washing, covers and other openings must be designed to always seal tightly, especially after maintenance operations are performed.

The charge pump suction port must be near the bottom of the reservoir, but not so near that settled contaminants are sucked into the charge pump. The return port should be located as far from the inlet port as possible to provide maximum dwell time of the fluid in the reservoir. The returning fluid must always discharge below the fluid level, preferably through a low velocity diffuser to prevent aeration. Reservoirs should have a baffle(s) between the inlet and outlet ports to ensure adequate dwell time and to prevent surging and aeration of fluid.

A filler port should be provided which minimizes the potential for contamination entering the system during servicing or operation. A drain port in the reservoir is recommended to permit a complete change of hydraulic fluid without disconnecting other hydraulic fittings.

When designing a reservoir to serve multiple hydraulic systems, stagger the elevation of the fluid outlet ports. If the fluid level becomes low during operation, the system with the highest outlet port will malfunction first and provide a warning to the operator.



**Figure 4-2
Reservoir Design**

4.1.5 Conduit Sizing

Conduit sizing should be based on acceptable fluid velocity and pressure drop through the conduit. Typical guidelines are:

	<u>Fluid Velocity (ft/sec)</u>
Main system lines:	20-30
Case drain lines:	10-15
Suction lines:	4-6

Consult hydraulic hose and tube suppliers for recommendations on calculating pressure drop and for applicable pressure ratings.

Undersized main system lines will result in heat generation caused by high fluid velocity. Oversized main system lines can contribute to charge pressure droop by increasing the amount of fluid being compressed when system pressure rises.

Lines should be sized taking into account extremes in temperature, fluid viscosity and required flow. Charge pressure may have to be increased in order to accommodate cold fluid, long lines, or restrictions.

Undissolved air in hydraulic fluid can drastically reduce unit life. The main cause of aeration of the fluid is air leaks, particularly on the inlet side of the charge pump. Flow restrictions such as inadequate line size, elbow fittings, and sudden changes in cross-sectional area can also contribute to aeration.

Inlet lines should be designed with a minimum number of joints. Recommended joint types are flared fitting or O-ring seals. Pipe threads are not recommended. Also, shut-off valves in suction lines are not recommended unless qualified for vacuum service. Suction line length is critical to avoid aeration and should be made as short as possible.

4.1.6 Case Drain Requirements

All Sauer-Sundstrand closed circuit pumps and motors require case drain lines to relieve internal leakage. Also, flow passing over charge relief valves and loop flushing valves must be removed through the case drain line.

The main criteria for case drain line routing are:

- back pressure on the unit cases
- fluid velocity / line sizing considerations
- cooling requirements.

Case drain lines can be routed in series through multiple hydraulic unit cases. For circuits with loop flushing, the loop flushing manifold would be the initiation point of case drain flow and the pump would be the unit immediately before the heat exchanger (see Figure 4-1). This allows flow through all unit cases for lubrication, cooling, and contamination removal. The pressure drop through the heat exchanger is additive to the unit case pressure. If this approach results in excessive unit case pressures, alternate routing must be considered. Article 4.6 of this application manual provides variations for case drain circuits.

For hydraulic units with lip seals, case pressure limitations are seal structural limitations. Exceeding these values, even for short pulses, could result in premature wear out or blow-out of the seal.

4.1.7 Heat Exchangers

Heat exchangers should be sized to keep the fluid within recommended temperature limits. This is normally done by selecting the worst continuous operating condition, and sizing for all the transmission loss being cooled in the hottest ambient environment, at the continuous temperature rating. For many machines, this occurs at the highest transmission output speed obtainable. A rule of thumb often used is to select a heat exchanger with a capacity equal to 1/3 the horsepower of the machine. **Testing to verify that temperature limits are maintained is highly recommended.**

For units with metal face seals, continuous case pressure capability is based on minimizing leaks from the shaft seal, and on preventing gasket blow-out during abnormal operations. This type of shaft seal is not a “leak-proof” device. There is an oil film across the seal at all times which results in normal seepage of about one pint per 1000 hours of operation at published case pressure limits. An increase in case pressure will increase this seepage.

For systems with integral charge pumps, the charge pressure relief valve is referenced to case pressure, and the relevant charge pressure is actually the differential above case pressure. When case pressures are elevated, particularly during transient peaks, the differential charge pressure must be maintained. **Failure to maintain charge pressure differential with respect to case pressure may result in damage to the hydraulic unit.**

Systems with remote charge pumps should be tested to ensure that charge pressure differential and low loop pressure are always maintained. If a remote charge relief valve is not referenced to hydraulic unit case pressure, it is possible that the charge relief valve will not follow case pressure transients due to flow surges. The charge relief setting should be verified to be sufficiently high to account for expected peaks in case pressure.

Heat exchanger sizing may be affected by other circuit elements. A loop flushing shuttle generally decreases required capacity while flow restricting valves increase it. Frequent relief valve operation will increase the required heat exchanger capacity.

A bypass around the heat exchanger is usually required to limit back pressure during cold starts.

4.1.8 Filters

To prevent premature wear of the hydraulic components, it is imperative that only clean fluid enter the hydrostatic circuit. A filter capable of maintaining the fluid cleanliness to ISO 18/13 or better is recommended and may be located either on the inlet (suction filtration) or discharge (charge pressure filtration) side of the charge pump. All Sauer-Sundstrand pumps can be used with suction filtration, and several can be equipped with charge pressure filtration adapters.

The selection of a filter depends on a number of factors including the contamination ingress rate, the generation of contaminants in the system, the required fluid cleanliness, and the desired maintenance interval. Filters are selected to meet the above requirements using rating parameters of efficiency and capacity.

Filter efficiency may be measured with a “Beta” (β) ratio as defined by ISO 4572 as the ratio of the number of particles greater than a given size upstream of the filter to the number of particles greater than the same size downstream of the filter. The Beta ratio applies to a specific particle size measured in microns. For example, $\beta_{10} = 20$ means that for every 100 particles of 10 micron diameter entering the filter, only 5 particles of 10 micron diameter are found downstream of the filter.

For closed circuit systems with suction filtration and open circuit systems with return line filtration, a filter with $\beta_{35-45} = 75$ ($\beta_{10} = 2$) or better has been found to be satisfactory. For closed circuit systems which share a common fluid reservoir with open circuit systems, a considerably higher filter efficiency is recommended. This also applies to systems with gears or clutches using a common reservoir. For these systems a filter within the range of $\beta_{15-20} = 75$ ($\beta_{10} = 10$) or better is typically required.

The **filter capacity** required depends on the amount of contaminants ingressed into the system and retained in the filter and the desired maintenance interval. As a rough guide, a capacity in grams equal to twice the charge flow in GPM or 1/2 charge flow in l/min has been found to be satisfactory for many closed circuit systems.

Since every hydraulic system is unique, the filtration requirements and performance must be determined by test. It is essential that monitoring of prototypes and evaluation of components and performance throughout the test program be the final criteria for judging the adequacy of the filtration system.

On direct displacement non-servo controlled pumps, a pressure drop indicator on the filter is recommended to warn against damage caused by low charge pressure.

Suction Filtration

The suction filter is placed in the circuit between the reservoir and the inlet to the charge pump as shown in Figure 4-3. A non-bypass filter is recommended with a filter contamination monitor. A vacuum gauge, located nearest the charge pump inlet, is suggested. Inlet vacuums up to 6 in Hg (0.8 bar absolute) are acceptable during normal system operation. An indicator set at 10 in Hg (0.7 bar absolute) is recommended for a filter change. Vacuums on the order of 20 in Hg (0.3 bar absolute) are acceptable during cold start or intermittent conditions. Exceeding these values may result in charge pump cavitation, aerated fluid, and life reduction of the hydraulic components.

Charge Pressure Filtration

There are two basic types of charge pressure filtration; full flow and partial flow. In either case, the pressure filter is located downstream of the charge pump. Charge pressure filtration is often used in systems where the inlet vacuum recommendations would be exceeded if using a suction filter. Cold weather applications and systems with a long inlet lines are prime candidates for charge pressure filtration. Charge pressure filtration generally provides greater filtering efficiency than suction filtration.

Charge pressure filters must be rated for at least the maximum expected charge pressure per the application. Generally, filters rated at 35 bar (500 psi) minimum collapse pressure are acceptable. A 100µm to 125µm strainer located in the reservoir or in the charge pump inlet line is recommended to protect the charge pump from coarse contaminants.

Partial filter flow is achieved by incorporating the charge pressure relief valve ahead of the filter element as shown in Figure 4-3. Only the volume of fluid needed by the hydrostatic loop and servo control are filtered. A non-bypass filter is recommended. Insufficient flow through the filter will result in inadequate charge pressure and will be reflected in poor machine performance. The filter must be capable of a pressure drop equal to charge pressure without allowing the filter β_x ratio to become less than one. A loop flushing valve is recommended to increase the volume of “dirty” fluid being removed from the pressure loop and to increase the volume of make-up fluid passing through the filter.

Full filter flow is achieved by incorporating the charge pressure relief valve behind the filter element as shown in Figure 4-4. All of the charge pump flow is passed through the filter increasing the rate of contamination removal from the system.

With full flow filtration, a bypass valve is necessary to prevent filter damage and to avoid contaminants from being forced through the filter media by high pressure differentials across the filter. In the event of a high pressure drop associated with a blocked filter or cold start-up condition, fluid will bypass the filter. Operating the system with an open bypass should be avoided, and a visual or electrical contamination indicator is recommended. Proper filter maintenance is mandatory to prevent particle contaminants from entering the system.

Refer to Section 3, “Fluids and Filtration” of this applications manual for additional information.

Pressure Filter Circuit - Charge Relief Upstream of Filter

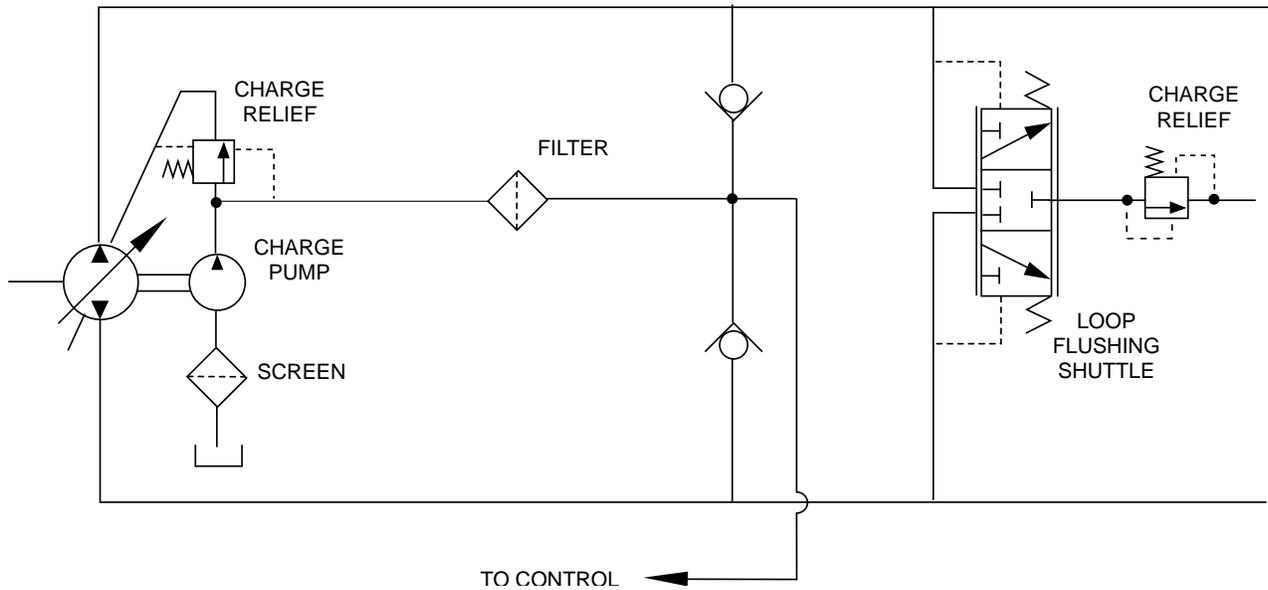


Figure 4-3

Pressure Filter Circuit - Charge Relief Downstream of Filter

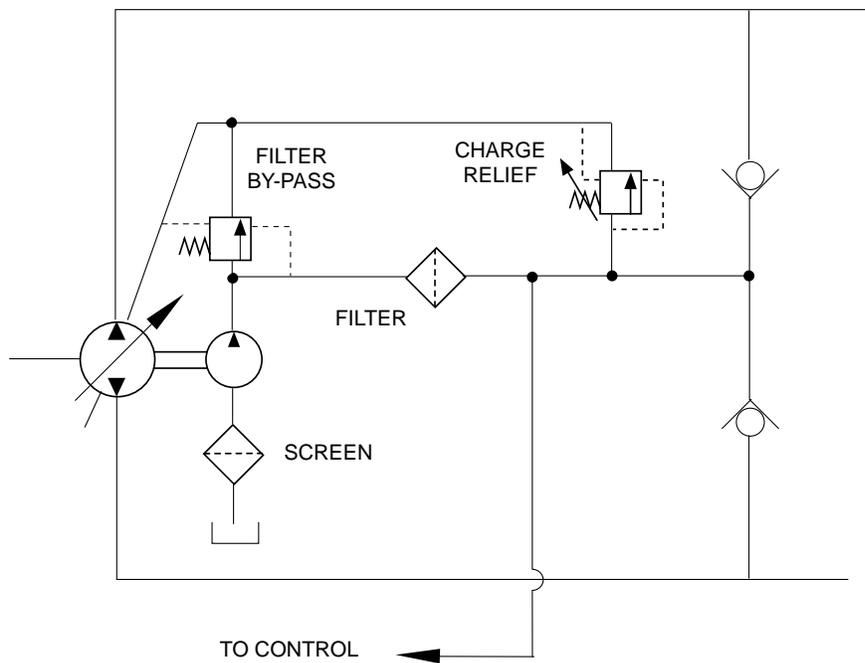


Figure 4-4

4.1.9 Loop Flushing Shuttle

There are two primary fluid loops in a closed circuit transmission, each having a distinct design function and associated circuit elements. These are the main system power loop and the charge cooling loop. Fluid is transferred between the loops by leakage, and if so equipped, by the loop flushing shuttle. Fluid quality is controlled primarily by the charge/cooling loop, but the demand for high quality fluid is in the power loop. The loop flushing shuttle is the circuit element which is available to control this transfer of quality fluid.

The loop flushing shuttle removes some set volume of fluid from the power loop which must be made up by the charge pump. The fluid removed is presumably “dirty” and/or “hot” while the make-up fluid is cool and clean. This greatly improves the heat and contamination transfer out of the power loop, improving fluid quality and extending component life. While some hydrostatic systems can operate satisfactorily without a loop flushing shuttle, its addition can reduce circuit problems where the power loop generates heat or contaminants.

Sauer-Sundstrand offers loop flushing shuttles in several configurations. Most closed circuit motors, as well as the Series 42 pump family, are available with integral loop flushing. A remote mounted loop flushing shuttle is also available.

Integral loop flushing valves reduce the number of external components to be purchased and plumbed into the system. Motor-mounted shuttle valves also increase the volume of fluid flushing and cooling the motor case.

The loop flushing shuttle is recommended for any of the following conditions:

- Sustained operation at low pressure and high speed. This condition can create high mechanical losses, low power loop leakage, and build-up of heat in the power loop. In general, low pressure is below 1000 PSI and high speed is above 2/3 full angle rated (continuous) speed. Motor-mounted shuttle valves are recommended for this condition.
- Cylinders in the hydrostatic circuit. Cylinders allow ingress of contaminants at a significantly higher rate than other circuit elements.
- Flow restricting valves in the power loop. These valves create heat in the loop. Note, however, that a loop flushing shuttle must be placed upstream of any divider-combiner valve. (See Article 4.10.)
- Frequent operation of high pressure relief valves. This creates heat. This restriction also applies to the use of modulating relief valves in the main power loop.
- Long power loop lines. Pressure drop will be higher than normal, increasing heat generation in the loop. In general, long lines exceed 150 diameters in length.
- Partial flow charge pressure filtration. A loop flushing shuttle is required to remove more fluid from the power loop and, therefore, to replenish higher volumes of cool, clean make-up fluid.

A higher degree of system reliability is obtained with a loop flushing shuttle. A shuttle valve can improve an unsatisfactory condition of built-in contamination or marginal cooling. A motor-mounted shuttle can also reduce the length of the case drain line from the transmission to the reservoir. A shuttle valve can potentially reduce cooler size. The final determination is made by evaluating trade-offs between the shuttle, cooler, filter, configuration of lines and expected life.

4.1.10 Pressure Controls

Closed circuit hydraulic systems require a means to limit or control the maximum loop pressure to some design value. Since hydraulic unit life decreases substantially at high pressures, unit life may be unacceptably short if system pressure is not controlled. The type of pressure protection selected will effect the response time and power dissipation of the system.

Relief valves are fast acting and typically do not have large pressure overshoots, provided the flow capacity of the valve is not exceeded. Full flow relief valves are uncommon in most hydrostatic pumps. However, high pressure relief valves dissipate power, sometimes in considerable amounts, and this is a source of heat generation in the system. Hydrostatic units should be sized and pressure relief valves set so that the maximum pressure is reached less than 2% of total operating time.

The **pressure limiter**, shown in Figure 4-5 is a non-power dissipative high pressure control, and therefore limits the heat generated in the system. The pressure limiting function can be used as an integral part of the machine control system. However, if the pressure limiter setting is high, the amount of time at the pressure limit must be balanced against required life.

Unlike a relief valve, the pressure limiter changes the displacement of the pump in response to high system pressure. When system pressure reaches the pressure limiter setting, the pressure limiter ports control fluid to the servo system to destroke the swashplate to the position required to maintain the set system pressure. Pressure limiters can be used to control winch payout loads because the swashplate will travel into the opposite direction if required to maintain the set system pressure. In a typical propel application, the pressure limiters act to reduce pump displacement during acceleration or forward driving. During deceleration or dynamic braking, the pressure limiter acts to increase pump displacement.

The response time and overshoot of the pressure limiter are a function of the pump displacement and the rate of pressure overload. If the pump is at full displacement when the pressure limiter activates, it will take longer for the pump to destroke than if it was at half stroke. Similarly, if the system experiences a high pressure spike, there will be greater overshoot than if the load developed slowly. To dissipate these overshoots in the short time before the pressure limiter becomes fully activated, the pressure limiter is equipped with an internal relief valve to trim pressure overshoot, so an external relief valve is usually not

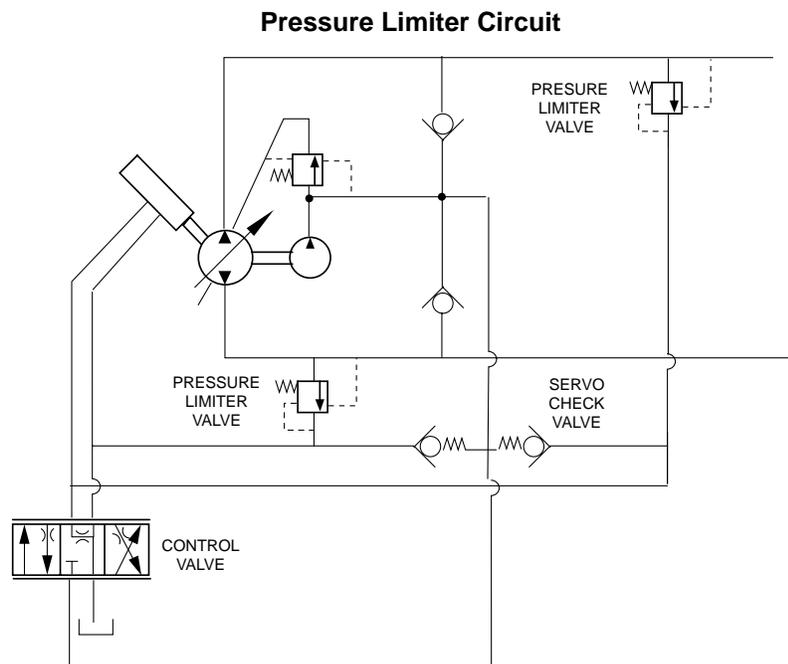


Figure 4-5

necessary. However, systems with control valves in the power loop are an exception to this as they have fast flow change characteristics. High flow, high pressure relief valves are required with directional or switching valves in the power loop. Testing is recommended to determine the amount and duration of pressure overshoot when using control valves in the main power loop.

The **pressure override**, shown in Figure 4-6, is another means of limiting system pressure in a hydraulic circuit. Like the pressure limiter, the pressure override acts to destroke the swashplate in response to high system pressures. When the set pressure is reached, the servo control pressure (charge pressure) is blocked, and the two servo pistons are hydraulically shunted. The pressure override does not destroke the swashplate as quickly as the pressure limiter, but again, the response time and pressure overshoot are a function of pump displacement and the rate of pressure overload. High pressure relief valves are required to trim pressure spikes in the short time before the pressure override becomes fully activated. The recommended relief valve setting is 1000 psi higher than the pressure override setting to provide for adequate response and minimize interference between the two pressure controls.

Braking is also a special situation regarding pressure limiting devices. During downhill braking, the

pressure limiter or pressure override will act to increase vehicle speed if the pressure setting is reached. This may decrease the amount of braking power available from the transmission at high vehicle speed as compared with a relief valve pressure control system. Because the pressure limiter and pressure override valves are nondissipative by design, the engine absorptive power limits the amount of high speed braking effort. With relief valves, some of the power required for high speed, steep slope braking can be dissipated over the reliefs by destroying the pump.

Continued braking over relief valves will heat the hydraulic fluid and may damage the transmission, so only emergency situations should be considered for use of reliefs. Since vehicles must also be equipped with mechanical brakes which are suitable for emergency stops, emergency relief valve braking is usually redundant. In other words, **a braking system which is independent of the hydrostatic transmission must be provided which is adequate to stop and hold the vehicle should the condition develop.**

Machines with dual path transmissions which depend on differential output speed to steer must verify that the pressure control device does not prohibit adequate steering control. This applies for both driving and braking.

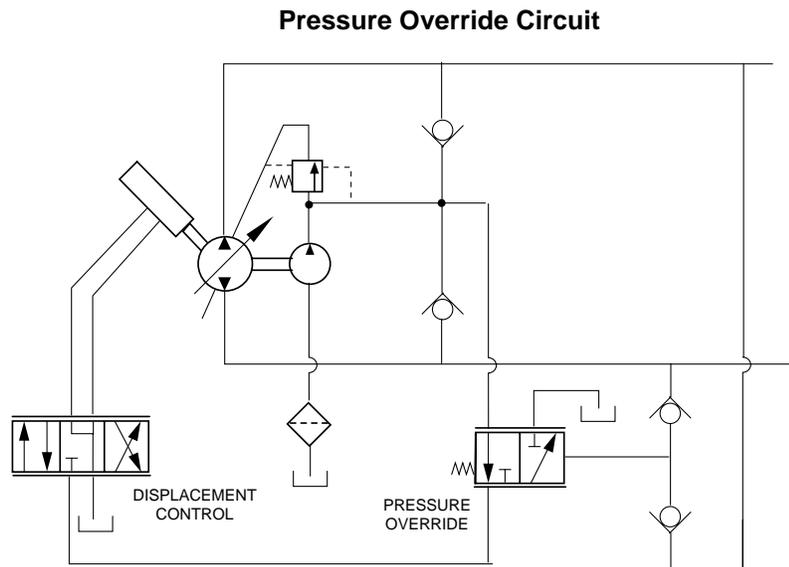


Figure 4-6

4.2 Components for Special Functions

Some applications may require additional circuit components in order to perform specific functions. A few of the frequently used components are discussed in this article which includes suggestions and precautions on their use.

4.2.1 Flow Dividers / Combiners

Flow dividers are used in circuits where two motors in parallel are supplied with a single pump. Typically, the flow divider is expected to provide equal flow rates to the motors to ensure positive traction or to provide synchronized motor speeds. On propel circuits, the flow divider also provides overspeed protection if one motor loses traction.

Vehicle steering usually requires that the leftside and rightside motors operate at different speeds. The flow divider must have the capability to allow this differential flow split. Motor speed ratings must not be exceeded during steering differentials.

The pressure drop through the flow divider generates heat in the system and should be considered when sizing heat exchangers. Motor-mounted loop flushing valves should **not** be used. Instead, an external loop flushing valve should be located between the pump and the flow divider to ensure proper system performance and correct flushing flow. The combining mode should be reserved for the least-used travel direction.

4.2.2 Flow Limiters

As their name implies, flow limiters are used to limit the flow rate to a set value. In hydrostatic circuits, they are used to prevent motor overspeed, especially in multiple motor applications in the event of a loss of traction. Because the flow limiter does not regulate flow below the maximum setting, they do not make good positive traction valves.

In the divide mode, the volumetric tolerance stack-up of the motors and the flow divider may cause the motor inlet to cavitate. Volumetric tolerance stack-up includes the slight difference in motor displacement, the difference in motor volumetric efficiency, and the flow division capability of the flow divider. Anti-cavitation check valves may be required to protect the motor inlets.

Thorough system testing is essential to verify stable charge pressure and motor inlet pressures under all operating modes. This includes all combinations of forward and reverse travel, vehicle steering, and at various pump displacements, pump speeds, and motor displacements.

To prevent the possibility of inlet cavitation with an overrunning load, the flow limiter must be mounted on the outlet side of the motor. Flow limiters should be sized to be active only during an overspeed event to minimize the heat generation in the system. When the flow limiter is active, the flow restriction will cause higher-than-normal back pressure on the motor. Although this is not usually detrimental to the motor, it can effect loop flushing flow and motor life.

System testing must be conducted to verify correct performance.

4.2.3 Counterbalance Valves

Counterbalance valves are normally used in open circuit applications to hold a load while the pump is idling. However, some hydrostatic winch applications use counterbalance valves to maintain a payout tension.

While active, the counterbalance valve generates considerable amounts of heat due to the flow rate and pressure drop across the valve. Because the valve generates high back pressure on the motor, a motor-mounted loop flushing valve cannot be used. An external loop flushing valve should be considered.

4.2.4 Accumulators

An accumulator is rarely necessary in a closed loop circuit. However, the two functions that an accumulator could serve in a hydrostatic system are to absorb shock in the high pressure loop or to provide instantaneous low loop make-up flow. This instantaneous make-up flow requirement is associated with fluid compressibility problems, sometimes called the bulk

modulus effect. Systems with long lines or applications which experience load induced pressure spikes are vulnerable to fluid compressibility and normally require supplemental low loop make-up fluid for a short time duration. However, the accumulator volume adds to the total volume of the system which is subject to the fluid compressibility during a load induced pressure spike.

4.2.5 Cylinders

Generally, open loop circuits are chosen to control hydraulic cylinders. However, a closed loop circuit can be utilized if the correct precautions are observed.

One type of cylinder found on many mobile vehicles is the hydraulic release parking brake. A 2-position 3-way valve is typically used to either release the brake with charge pressure or to apply the brake by venting the cylinder to tank. This brake control sub-circuit is actually an open circuit which simply utilizes charge pressure as a convenient pressure source for brake release. Although there is no sustained charge flow requirement, there is an instantaneous flow requirement. In the vast majority of installations, this charge flow requirement is insignificant. However, it could become significant if the brake release line is very long, if there are multiple brakes being released, or if there are other charge flow demands occurring at the same time.

A closed circuit installation of a double ended cylinder is similar to a hydrostatic motor installation in that the return flow from the cylinder is the same as the input flow. Generally, no special considerations are required for charge pump sizing. Obviously, pump control and high pressure protection is critical to avoid system damage or heat generation when the cylinder has reached the end of its stroke.

A closed circuit installation of a single ended cylinder requires special considerations due to the unequal volumes of the rod and bore ends of the cylinder. When the cylinder is activated from the bore end, the input flow is greater than the exhaust flow. If the charge pump is not able to make-up this volume difference, the low loop side must be supplemented through a check valve. Conversely, when the rod end is pressurized, the return flow is greater than the input flow. Generally, the internal charge relief valve is unable to pass the increased low loop flow, and an external shuttle and relief valve are necessary to dump the flow to tank.

The input and exhaust flow differentials discussed above are a function of the ratio of the rod and bore end areas of the cylinder. The greater the area difference, the greater the flow differential that must be accommodated with external circuitry.

4.3 Remote Variable Pressure Limiter

4.3.1 Introduction

A single sided Remotely Variable Pressure Limiter (RVPL) circuit can be created by properly connecting a high pressure pilot-operated relief valve to a standard Series 90 pump. Adding a RVPL circuit allows for a pressure limiter setting that is adjustable and lower than the internal pressure limiter. This added control feature is useful for applications requiring adjustable constant pressure control, such as drill rig pull-downs and constant tension winches. More detailed information about applications is presented after the explanation of the circuit.

4.3.2 Circuit

Figures 4-8 & 4-9 illustrate the external circuitry required for a single sided RVPL. A high pressure line must be routed from the appropriate system pressure gage port on the pump (or other system pressure source) to the inlet of a remotely mounted pilot pressure relief valve (the RVPL valve). The output of this valve is then returned to the appropriate servo pressure gage port on the pump. Figure 4-7 shows the series 90 pump gage ports.

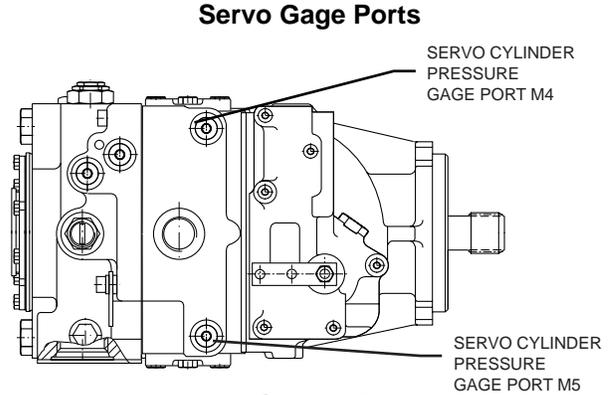


Figure 4-7

The external RVPL valve becomes a second pressure limiter valve which is connected in parallel with the internal pressure limiter valve. Of these two valves, the one which is set at a lower pressure will limit system pressure. The internal pressure limiter should be set at a maximum desired system pressure, and it will limit maximum system pressure, even if the external RVPL valve is adjusted to a higher setting. Lower system pressures are maintained by adjusting the external RVPL valve to settings lower than the internal pressure limiter.

**Remotely Variable Pressure Limiter Circuit For Series 90 Pumps
(Counterclockwise Rotation)**

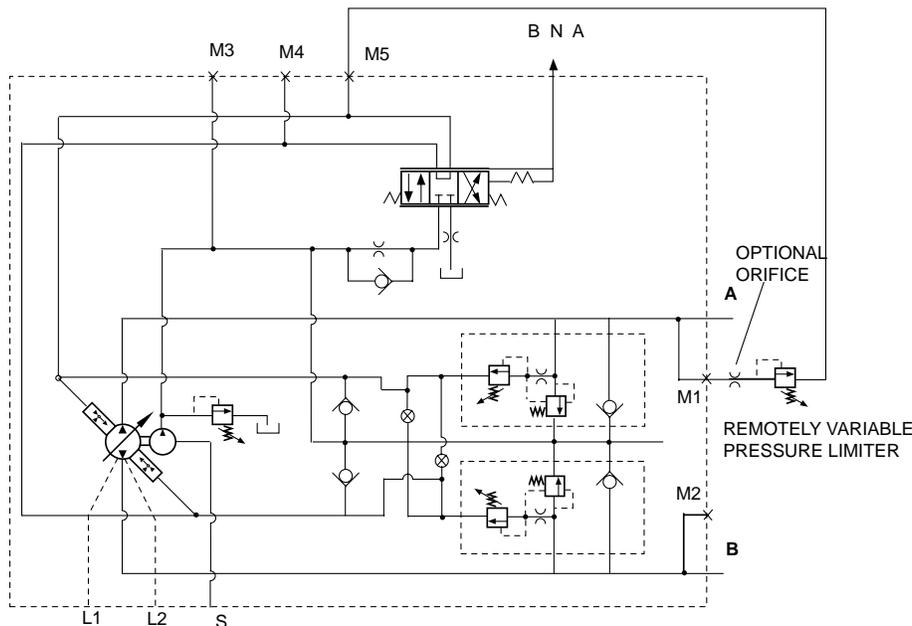


Figure 4-8

**Remotely Variable Pressure Limiter Circuit For Series 90 Pumps
(Clockwise Rotation)**

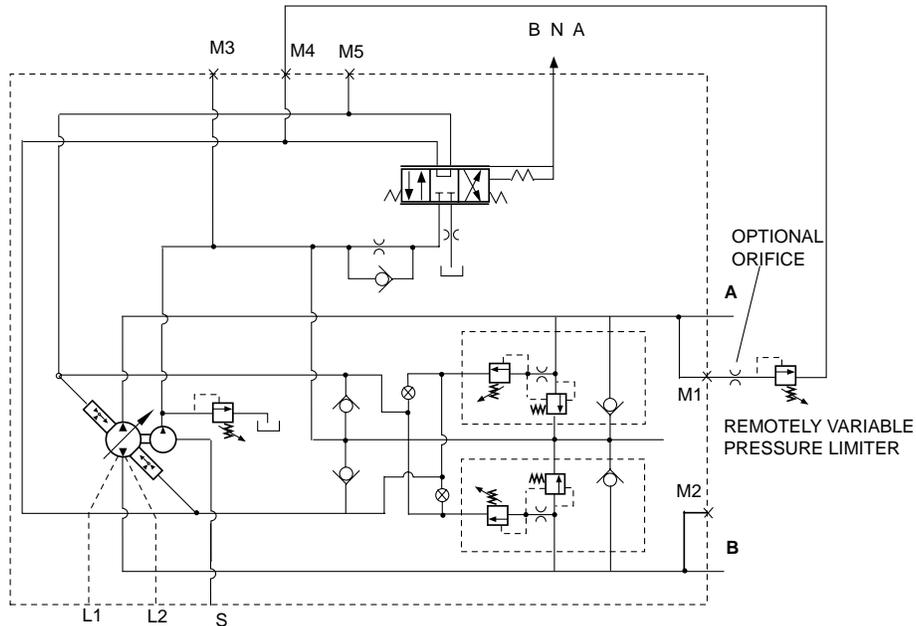


Figure 4-9

The external RVPL valve will always be connected to the system port in which it is desired to have the RVPL function. However, the appropriate servo connection is dependent upon pump rotation shown in Table 4-1.

Warning—If the servo gage port plumbing is reversed, the remote pressure limiter will not function.

If a two-level pressure limiter is desired without readjustment of the external RVPL valve, the external valve can be switched in and out of the circuit with an on/off valve (manually, hydraulically or solenoid operated) placed in series. This valve will switch pressure limiter operation between the internal and the external valve settings. This logic can even be extended to more than two pressure limiter levels, using additional external RVPL valves and on/off valves.

Do not attempt to control the system delta pressure near zero psi. The system gage pressure must always be above charge pressure and preferably no less than 750 psi.

If independent control of both system pressures is required, the RVPL circuit shown can be separately applied to both sides of the pump. It is also possible to control both system pressures to the same level with a single external RVPL valve. However, the external circuitry becomes more complex, and Sauer-Sundstrand application engineering should be contacted for assistance.

PUMP LOGIC CHART			
ROTATION	PORT	SYSTEM GAGE PORT	SERVO GAGE PORT
RH (CW)	A	M1	M4
	B	M2	M5
LH (CCW)	A	M1	M5
	B	M2	M4

Table 4-1

4.3.3 Component and Circuit Performance

Ideally, the external RVPL valve should have flow characteristics close to that of the internal pressure limiter valve. The critical characteristic is the **pressure rise rate** versus flow, which is the slope of a pressure versus flow graph for the RVPL valve. The desired pressure rise rate is 250 psi/gpm. A pressure rise rate is illustrated in Figure 4-10. If the pressure rise rate of the RVPL valve is higher (more than 250 psi/gpm), response of the RVPL will be slower than that of the internal pressure limiter. If the pressure rise rate of the RVPL valve is significantly lower (less than 250 psi/gpm), the RVPL response will be faster. The faster RVPL (high flow capability) can create instability or overpressurize the servo.

Applications attempting to utilize faster RVPL responses should be thoroughly tested to insure adequate performance and life.

If it is desired to utilize an RVPL valve with an inadequate pressure rise rate (flow capacity too high), the desired pressure rise rate can be approximated by placing an orifice in series with the external RVPL valve. This configuration is shown as being optional on the circuit diagram. If the RVPL valve has very little pressure rise, an orifice of .070 inch diameter will provide an acceptable pressure rise rate in the RVPL circuit. If the pressure rise rate of the RVPL valve is an intermediate value (low but not negligible), a larger orifice may be used. The desired final result is a combination of the external RVPL valve and the orifice which has a pressure rise rate of 250 psi/gpm at a flow rate of 1 gpm. The valve will typically operate correctly while maintaining constant system pressure.

To minimize poor RVPL response with cold oil, 3/8 inch (#6) lines are recommended to and from the external RVPL valve, particularly for long runs (10 feet or more). The pump gage ports are #6 SAE straight thread ports.

With the desired 250 psi/gpm pressure rise rate of the external RVPL valve, the swashplate destroking performance of the RVPL will be similar to that of the internal pressure limiter. However, it should be noted that the RVPL will experience more pressure overshoot (pressure rise above its steady state setting) during load transients requiring rapid destroking. This will occur because the internal system pressure relief valve will not operate in tandem with the external RVPL, as it does with the internal pressure limiter.

The system relief valve limits **pressure overshoot** of the internal pressure limiter when swashplate response is not sufficient to limit system pressure to 200-400 psi above the limiter setting. When controlling with the external RVPL, the swashplate destroking response rate will be a function of the system pressure overshoot above the steady state RVPL setting. If the pressure rise rate of the RVPL circuit is a constant 250 psi/gpm, the following table gives estimated 17° to 0° destroking times for various levels of overshoot (assumes constant level of overshoot during destroking). Note that the use of a high flow RVPL valve with a series orifice will result in slower response, particularly at the higher overshoot levels. This is because the pressure rise rate for an orifice is nonlinear, requiring more pressure overshoot to obtain a given flow.

Destroke Time (17° to 0°) for Pressure Overshoot

Series 90 Pump	Pressure Overshoot			
	250 psi	500 psi	750 psi	1000 psi
42cc	0.21 sec	0.11 sec	0.07 sec	0.05 sec
55cc	0.35 sec	0.18 sec	0.12 sec	0.09 sec
75cc	0.45 sec	0.23 sec	0.15 sec	0.11 sec
100cc	0.65 sec	0.32 sec	0.22 sec	0.16 sec

Table 4-2

The proposed RVPL circuit has the capability to push the pump swashplate over center, even at low system pressures, to maintain system pressure near the RVPL setting. This is required in constant tension winch applications, where the cable must pay out as well as pay in to maintain constant pressure. In this application, the pump displacement control will be biased for full pay-in speed, and the RVPL will push the pump swashplate over center to allow pay out.

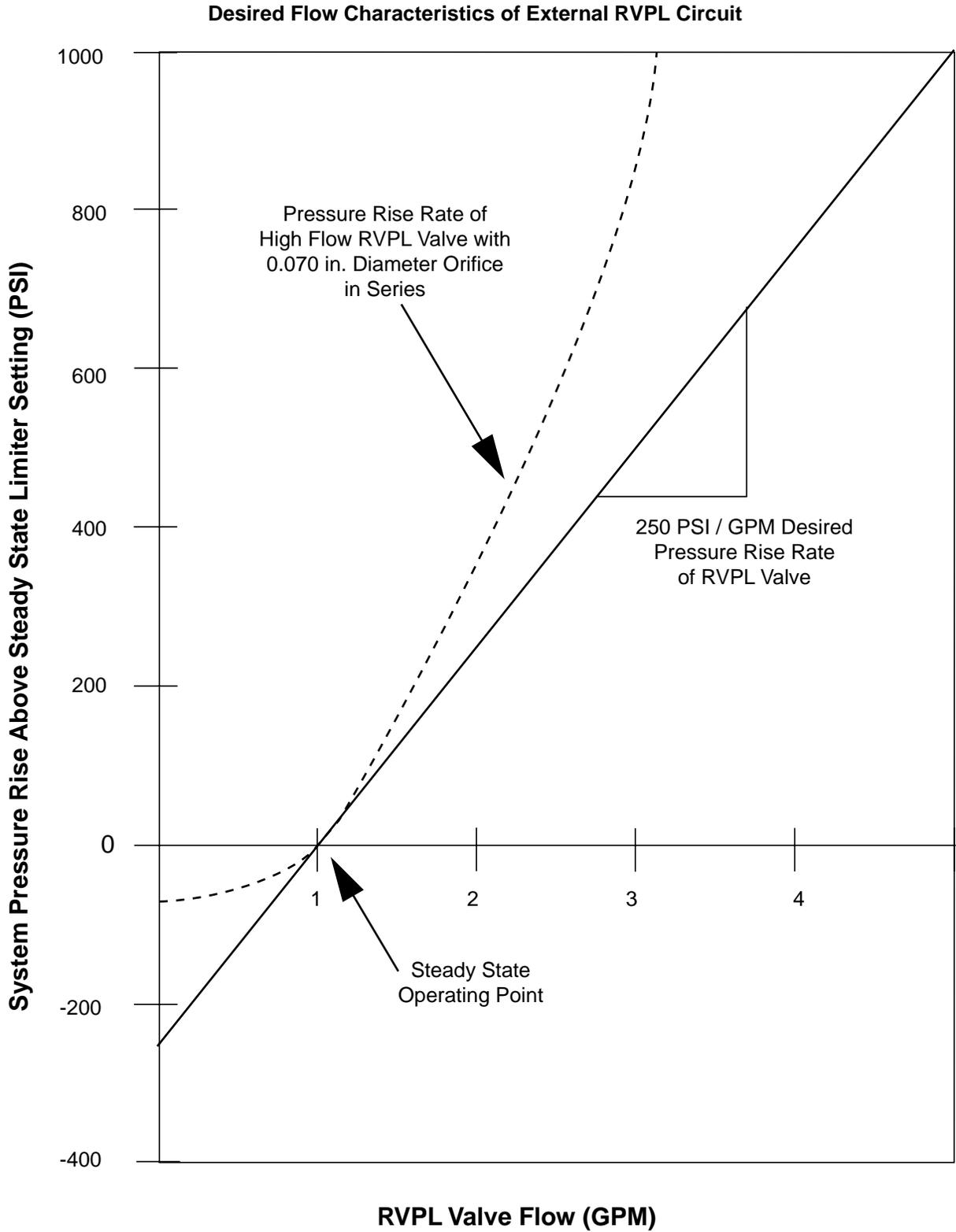


Figure 4-10

4.3.4 Applications

As mentioned, the RVPL circuit is useful for applications requiring adjustable constant pressure control, such as drill rig pulldowns and constant tension winches. The circuit is normally used with a displacement control (manual, hydraulic or electrical). The displacement control provides speed control when the RVPL is set at a pressure higher than is required to move the load. This allows precise load positioning when the system is not in the constant pressure mode. When operating at constant pressure, the RVPL is adjusted to maintain a desired pressure level, and the displacement control is set to limit the maximum load speed. The RVPL is designed to behave in a manner similar to the internal pressure limiter to reduce pump displacement when the pressure reaches the RVPL setting.

In **drilling applications**, motor output torque is limited by limiting pressure with the RVPL. The RVPL can be used to control drilling torque, with the internal pressure limiter set at a higher pressure.

On a **constant tension winch**, the displacement control is typically set to allow full pump displacement to run full speed cable pay in, while constant cable tension is maintained by means of the RVPL. The ability of the RVPL to drive the pump swashplate over center for a winch pay out condition makes it especially suitable for a mooring winch application. On a mooring winch, the pump's displacement control is set to pay in the cable at a certain speed. If the cable tension increases to the point of the RVPL setting, the pump's displacement will decrease to maintain the pressure setting. If the tension is great enough, the RVPL will drive the swashplate over center and the pump will pay out cable. Consideration must be given if the load can drive the swashplate to its maximum displacement when paying out cable. Any flow generated by the load that the pump cannot take in will either drive the pump faster or drive the pressure higher. A relief valve can be used to limit the maximum pressure condition, but excessive heat will be produced. In most cases the prime mover, acting in a dynamic braking mode, will be the limiting factor. A separate method of braking the load must be provided to prevent engine or pump overspeed.

High inertia vehicles, such as railroad maintenance machines, need control of deceleration forces. The pump has a capability to be driven to a neutral swashplate condition before the vehicle reaches a zero speed. Without some method to prevent this from occurring, the internal pressure limiter will drive the pump towards maximum displacement during dynamic braking. The pump, now acting as a motor, will feed a large amount of torque into the engine due to the pump's high pressure and displacement. This may be more torque than what the engine can handle in a braking mode. A circuit utilizing the RVPL, while sensing travel direction, can be used to limit system pressure while decelerating. By setting the RVPL at a pressure level less than the internal pressure limiter, torque can be absorbed by the prime mover at a safe level. The system is designed such that the internal pressure limiter controls the acceleration torque while the RVPL controls the deceleration torque.

Certain applications require that **maximum torque** levels be **controlled**, whereby the load is dependent on a second hydraulic drive. For example, a drilling operation may require the maximum torque on the drill head to be controlled. The torque, however, will depend on the rate of pull-down of the drill head. An RVPL circuit could be used to limit torque in the following manner: both the drilling pump and the pull-down pump would have their pressure limiter settings set above the RVPL. The RVPL would sense system pressure from the drilling circuit, but instead of feeding the RVPL signal back into the drilling pump, the signal would feed into the pull-down pump to bring it out of stroke. Drill torque is then controlled by the pull-down rate of the drill head.

As another example, a **harvesting machine** may include two hydraulic systems: one system to propel the machine and one system to feed crop into the machine. In this application, the RVPL would sense the pressure of the system gathering crop into the machine, but its signal would be tied to the propel system. When the gathering system pressure increases due to heavier crop, the RVPL will operate to slow down the vehicle speed.

When two or more pumps are used in parallel, an external pressure limiter is also recommended. For more detailed information for this kind of application, see Article 4.5, "Ganged Pumps."

4.4 Wheel Assist

4.4.1 Description

Additional traction is often needed on machines working in poor traction conditions. The need may only be to prevent stalling of the machine, or to increase the drawbar pull or to assist in steerability. Axles, or wheels, not usually used for the propel function can be fitted with hydraulic motors to give auxiliary tractive force. Each vehicle has its own particular needs and each must be analyzed according to those needs.

Some of the considerations that will influence the size or type of wheel assist are:

- weight distribution due to loading or grade
- coefficients of traction
- auxiliary drive wheel speed
- wheel diameter
- final drive ratios.

As a general rule, the wheels which are offering the additional traction must not lose traction (spin out) before the main drive axle wheels spin out. This is especially true when fluid from the main propel circuit is used for the wheel assist function. The traction coefficient must be known, as well as the load distribution or the weight shift distribution. Variations in the traction coefficient and load distribution may exist due to changes in traction and operating grades.

Wheel speeds will change due to shifting into a different final drive ratio or to changes in variable motor displacements.

Often the assist and main drive wheels are of different diameter which then requires a different rpm for any given vehicle speed. Likewise the turning radius affects a speed change between the inside wheel and the outside wheel. Maximum speed limits of the wheel assist motors must be considered closely due to these variations as well as any downhill speedup. Flow dividers may be necessary to prevent full flow from going to any one motor in any circumstance.

Controlling a wheel assist function can be done in a variety of ways. Variable motors, using main system drive flow as the power source, can be stroked for additional traction, manually or automatically, at some designated pressure setting. Pressure compensated motors stroked at a pre-selected pressure may work well when traction coefficients are rather consistent. For instance, if the main drive wheels are losing traction at a pressure less than the wheel assist pressure set point, the wheel assist motors would not shift offering any additional traction.

Sensing a percentage of main drive wheel slip can be the signal to the control system to add wheel assist traction. Typically, a separate pump and motor arrangement would be used. In this case, the main drive could be mechanical or hydraulically driven.

Application assistance is always available when wheel assist drives are encountered.

The wheel assist shown in Figure 4-11 is accomplished with two variable motors and wheel ends on the steerable wheels. These variable motors are typically capable of both zero and full displacement. While in a mode of low torque requirements, the variable motors are in zero displacement. When more tractive force is required, the motors are shifted to full displacement.

The wheel assist shown in Figure 4-12 can accomplish the additional tractive force required with a separate designated pump for the assist function. This system can produce a fixed or variable tractive force by the use of an electronic pressure control. By controlling pressure level from zero to maximum by operator input, just the right amount of assist traction is possible. At a zero pressure command, vehicle coast is possible.

Operator Controlled Wheel Assist

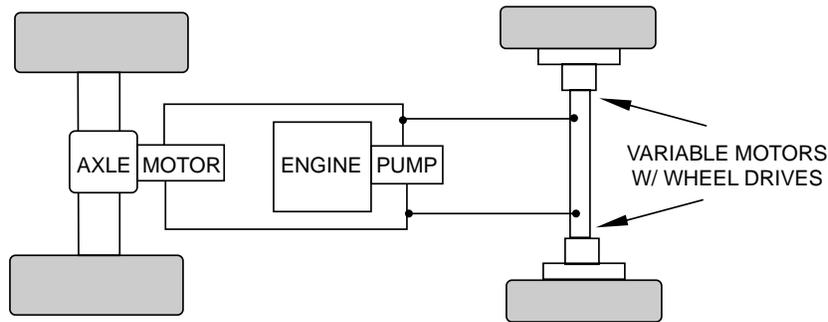


Figure 4-11

Pressure Controlled Wheel Assist

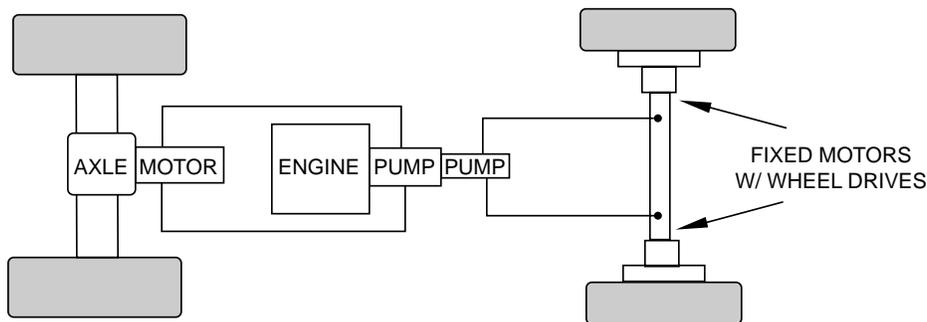


Figure 4-12

4.5 Ganged Pumps

4.5.1 Introduction

The full through drive torque capability of the Series 90 pump is often used to eliminate the need for multi pad pump drives. The Series 90 pump family has found additional success with the through torque drive by “ganging” pumps together. A ganged pump is defined normally as two or more Series 90 pumps bolted together with the output flows combined to

drive one load. The objective of the ganged pump is to simulate the performance of one large pump with the flow of two or more smaller pumps. Two smaller pumps will have higher individual speed capabilities than a large pump and therefore could offer an input speed advantage. “Ganging” pumps has created an entirely new set of larger pump displacements for additional application flexibility.

4.5.2 Circuit Recommendations

In order for two pumps to operate and perform as one, it is critical that each pump reacts similarly to the common load. The Series 90 Pressure Limiter automatically adjusts pump displacement to limit pressure when the limiter setting is reached, but is not practical to assume that the valves in two pumps will be set exactly the same. Therefore, the lowest valve in either pump would attempt to regulate the pressure of the entire system by controlling the displacement of one pump. To prevent this undesirable effect, a single external valve should be used to control both of the ganged pumps simultaneously.

The concept shown in the figure below is commonly referred to as a “Remote Variable Pressure Limiter” (RVPL) and details can be obtained from Sauer-Sundstrand Application Engineering. Electrical Displacement Controls (EDC’s) are shown in the figure. The check valves are necessary to isolate the pump servos from one another when the system pressure

is less than the RVPL setting. The remote pressure limiter is a single side circuit. Two RVPL circuits are necessary to control both sides of the loop.

Both pumps must have a displacement control and it is recommended that they be of the same style (manual, hydraulic or electric). Do **not** use a cover plate and slave the second pump from the first. The displacement control adds the necessary swashplate feedback so that both pump displacements are matched closely to one another within the accuracy and tolerance of the two controls. At pressures below the RVPL setting the displacement control response time for each pump is not affected by the RVPL circuit. However, when the RVPL does function, the system pressure limiter response time will approximately double due to the total volume of two servos and added plumbing that needs to be pressurized before the pumps will destroke.

A ganged pump with the RVPL regulating port “A”

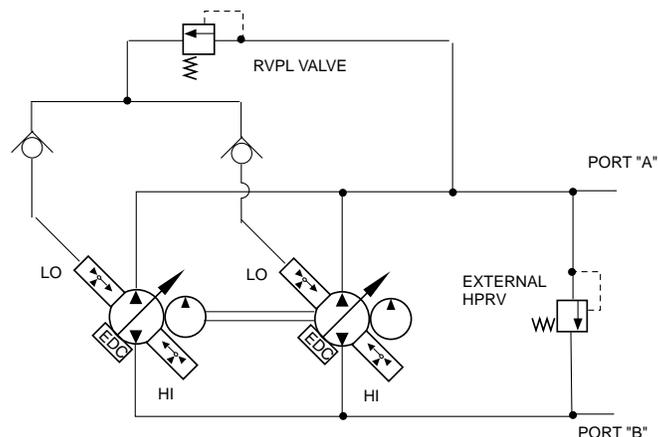


Figure 4-13

Since the internal pressure limiters (PL) of each pump needs to be functionally removed, it is necessary that they be ordered, or adjusted, to avoid any interference with the remote valve. The internal PL's should be set at a *minimum* of 35 bar (500 psid) *above* the setting of the remote pressure limiter valve. Since the high pressure relief function of the Series 90 Multi Function Valve is also eliminated when the integral PL's are functionally removed, it is recommended that an external high pressure relief valve capable of short duration high flow be added to the circuit at a pressure of 35 bar (500 psid) above the remote pressure limiter setting.

It is recommended that both pumps be controlled together. For example, an idle pump in neutral would be forced over center when system pressure is created by a second individually controlled pump. Additionally, the displacement change resulting from the operation of the pressure limiter is a result of a system of sources acting on the pump swashplate. The pumps should be controlled together so the net forces on each pump are similar and, therefore, the performance of the pressure limiter optimized.

Table 4-1 (p. 19) identifies the appropriate system pressure and low pressure servo gage ports that are used in the RVPL circuit based on input rotation. The port labels shown in the table are consistent with the Series 90 Technical Manual, pump outline and installation drawings.

Other control concepts and configurations are possible, but their acceptance would depend on the application and performance expectations. Any ganged pump application should be tested thoroughly for control and driveline performance. Ganged Series 90 pumps have been a success in many applications. For application assistance, or information on other control concepts, please consult your Sauer-Sundstrand representative.

4.6 Case Drain Routing Variations

The routing of case drains in a closed circuit system should allow for adequate cooling of the hydraulic fluid, but with minimum external plumbing. The figures in this section show the recommended case drain routings for various pump/motor configurations.

Figure 4-14 shows the simplest method for routing case flows for a pump/motor combination without loop flushing. The figure shows that all of the case flow goes through the motor housing. To ensure the cases of the pump and motor are full of fluid at all times, case outlets should come out at the top of the unit and case inlets should come in at the bottom.

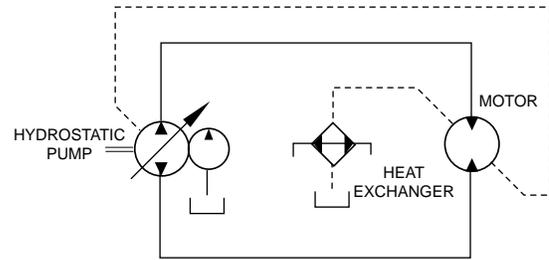


Figure 4-14

Figure 4-15 shows one method of case drain routing for a system which incorporates loop flushing. This method combines case flows for both the pump and the motor.

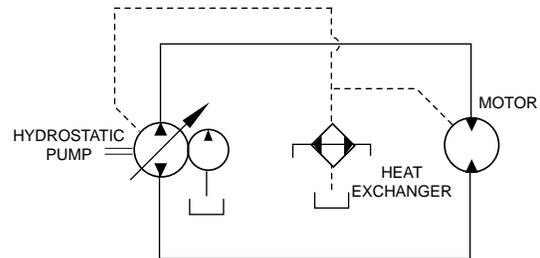


Figure 4-15

An alternative routing for a loop flushing system is shown in Figure 4-16. Here all of the case flow goes through the pump. This system uses somewhat simpler plumbing than the previous method. However, it has the disadvantage that the pump case temperatures will be greater.

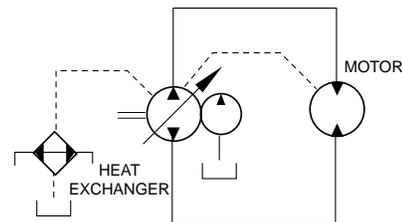


Figure 4-16

4.7 One Charge Supply / Two Pumps

It is possible to have one charge pump supply charge flow to two separate pumps, providing charge flow is great enough. The main advantage is that a single line from the reservoir may simplify plumbing.

Figure 4-17 shows that only one of the pumps is equipped with a charge pump, which supplies charge flow to the second pump through its charge pressure gage port. The charge pump must be sized large enough to provide adequate charge flow to both pumps for all operating conditions. This is especially important at low speeds, since charge flow is less and leakage losses are greater.

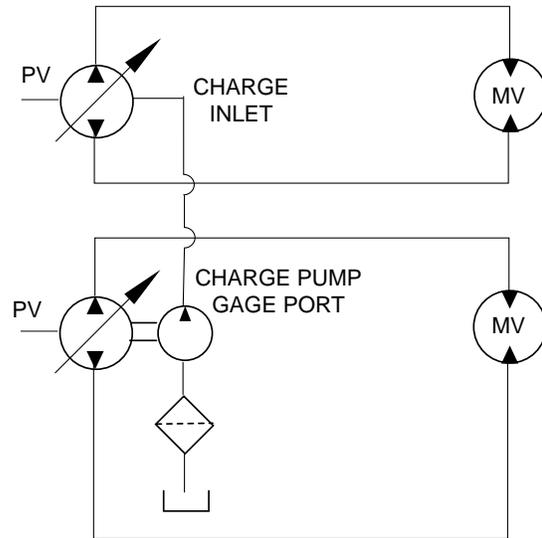


Figure 4-17

4.8 Charge From Implement Return Flow

Often charge flow for a closed circuit system can be provided by an implement pump, whose primary function is to power an open circuit load such as a cylinder, motor, etc. Figure 4-18 shows such an arrangement, whereby one or more pumps are without charge pumps and receive all of their charge flow from implement pump return flow. The main advantage to this method is the single inlet suction line from the reservoir.

Care must be taken that the implement pump return flow from the load always exceeds the closed circuit charge flow requirements for any operating condition. Also, a pressure filter should be placed between the charge inlet and the return flow to keep contaminants from the implement circuit out of the hydrostatic system.

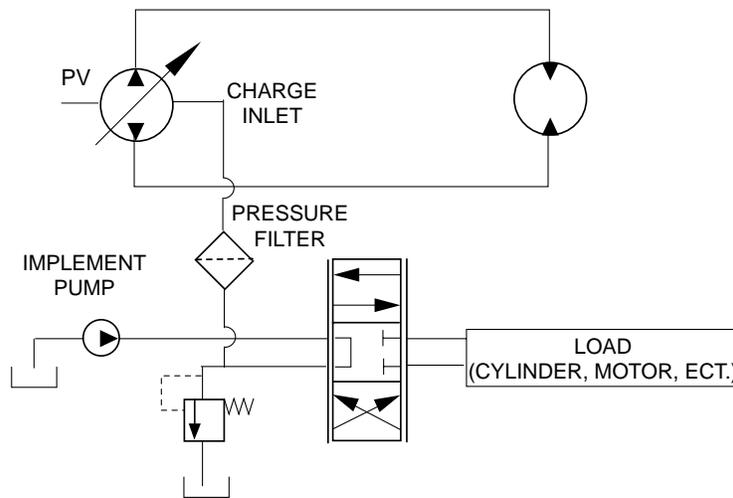


Figure 4-18

4.9 Multiple Motors

Several motors can be combined in the same closed circuit, but are normally only used in parallel. Series operation of axial piston or bent axis motors loads the bearings excessively causing substantial reductions in life expectancy. Contact Sauer-Sundstrand if series circuits are under consideration.

Several motors driven in parallel from one pump allow a potential motor overspeed situation. If one motor is at a reduced load due to poor traction or other unusual situation, it may overspeed and damage the unit if not sized to accept all the pump flow. Flow dividers, divider-combiners or flow limiters have been used successfully to prevent motor overspeed.

Motor overspeed is a safety concern when the transmission is providing retarding power. For swashplate motors, maximum speed may be limited by block lift and accompanying loss of drive. Operation above maximum speed is not permitted.

Flow dividers create heat, even during normal operation, as they continuously act during vehicle turns, with load differentials, or due to normal tolerance effects. Care must be used in sizing heat exchangers when using flow dividers.

Flow divider performance is sensitive to the droop built into the valve. A large droop should be used to improve control stability and reduce the amount of time the valve regulates. A flow droop of at least 30% prior to the valve restricting at the pressure limit is recommended. A cross-valve orifice may be required to achieve this. This may affect motor sizing and maximum speed expected and should be accounted for when sizing units.

A **divider-combiner valve** can pressurize both motor ports when operating in the combining mode. The frequency of this occurring must be anticipated as this condition greatly reduces expected hydraulic unit life. The combining mode should be used in the least frequent travel direction. We recommend testing to determine the level and duration of both motor ports being pressurized.

A loop flushing shuttle should be used to keep cool fluid flowing into the power loop. Flow in a loop flushing shuttle must be accounted for when using flow dividers or divider/combiners in a multiple motor circuit. Because the charge relief valve in the shuttle flows oil, a differential flow can exist from one side of the loop to the other. If the shuttle is installed between one motor and the flow control valve, charge flow causes the valve to operate, even though all motors may be at the same speed. A loop flushing shuttle should be installed between the pump and flow control valve in order to avoid this problem.

Motor speed may be controlled with the use of **flow limiters**, which operate at a set maximum flow. This type of valve only functions if overspeed is imminent, and it typically operates a small amount of time. Flow limiters should be installed to restrict motor return flow to avoid cavitating the motor inlet when they function. As this will pressurize both motor ports and reduce expected life, frequency of operation must be determined.

Sauer-Sundstrand is available for obtaining suggestions on transmission circuit design. Contact your Sauer-Sundstrand representative if questions of suitability exist. In any case, we recommend testing the final circuit for sufficient time and under severe enough conditions to determine acceptable performance and life.

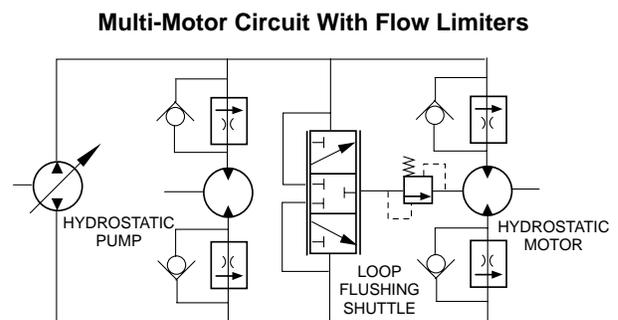


Figure 4-19

4.10 Flow Dividers and Divider / Combiners

4.10.1 Introduction

This article describes how flow dividers and divider-combiner valves are used in closed circuit applications and how special considerations must be taken when using these components.

Flow dividers and flow divider-combiner valves are normally used in vehicle propel circuits where two motors are plumbed in parallel and supplied by a single flow source. In such installations, they serve potentially two purposes; to prevent one motor from overspeeding if it loses contact with the ground, and to ensure uniform output speed of both motors.

The basic difference between flow dividers and flow divider-combiners is simply that the flow divider is not designed to combine flow. In the divide mode, their

performance is effectively the same. A circuit using flow dividers must either be a unidirectional system or it must bypass the divider during reverse flow. Because most vehicle propel systems are bi-directional, this article focuses on the divider-combiner with the understanding that the comments made with respect to the divide mode of operation are valid for both types of valves.

The information presented in this article is based on field experience using flow divider-combiners in closed loop hydraulics. Valve suppliers should be consulted regarding the performance, ratings, and sizing for each application.

4.10.2 Heat Generation

The pressure drop across the divider-combiner is typically on the order of 200 to 600 psi in the divide mode. Many valves will have an even greater pressure drop in the combine mode. This pressure drop is a source of heat generation in the circuit which should be accounted for when sizing the heat exchanger. The heat generation can be minimized if the divider-combiner is by-passed during modes of operation when it is not needed in the circuit.

4.10.3 Design Flow Range

Flow divider-combiners are designed for optimum performance within a specified range of flow rates. Operation outside of the specified range will significantly reduce the reliability of the valve and will likely cause circuit problems as well. Consult the valve supplier for correct sizing on each application.

4.10.4 Loop Flushing

If loop flushing is desired in a circuit using a flow divider-combiner, it must be located between the pump and the flow divider-combiner. If loop flushing is attempted at the motors, problems can occur during both the divide and combine modes of operation.

During the divide mode of operation, the low loop side of the motor should be at charge pressure. However, on propel applications where the two motors are effectively tied together by the ground, the high pressure side of one motor can actually be subjected to a vacuum (see section 4.10.8, Recommendations for a Trouble Free Circuit) due to volumetric discrepancies in the divider-combiner and the motors. If this occurs,

the loop flushing shuttle will shift and attempt to draw fluid from the inlet side of the motor, compounding the vacuum problem.

During the combine mode of operation, the low loop pressure at the motors is elevated by the pressure drop across the divider-combiner, which can be several hundred psi. Thus, the loop flushing relief valve at the motor, being subjected to this considerably higher pressure, would dump significantly more fluid than in the divide mode of operation. This could potentially cause increased case pressures and/or a loss of charge pressure in the combine mode of operation.

Loop Flushing When Used With a Divider-Combiner

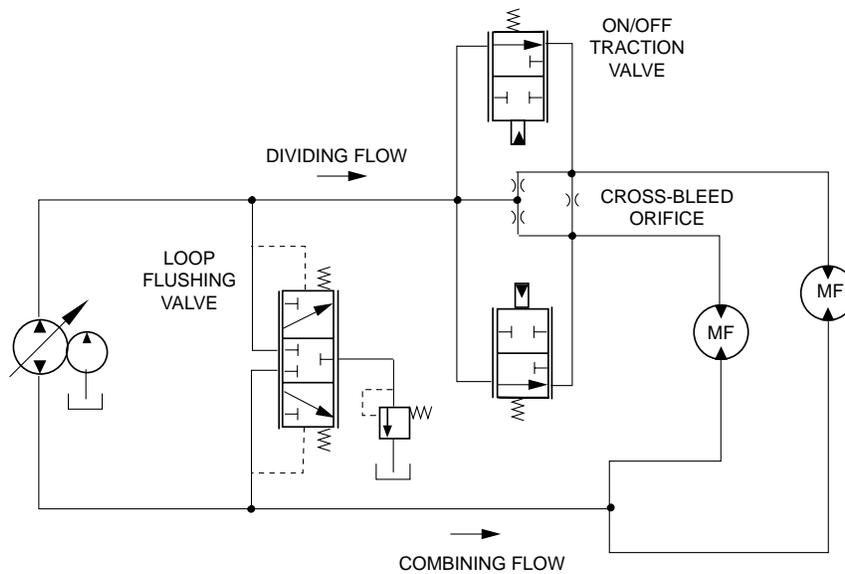


Figure 4-20

4.10.5 Motor Life Reduction

Hydraulic unit life is highly dependent on system pressure. In the combine mode of operation, when the "low loop" pressure is potentially several hundred psi greater than charge pressure, the motor life can be adversely affected. Although the delta pressure across the motor (the pressure that transmits torque) may be at some prescribed design value, the actual pressure

seen by the motor rotating group is the sum of the high and low sides of the loop. Therefore, the frequency of operating in the combine mode must be anticipated and included in the duty cycle to calculate a predicted unit life. Operation in the combine mode should be minimized if long motor life is critical.

4.10.6 Differential Flow for Steering

On a typical propel system, the motors must rotate at moderately different speeds in order for the vehicle to negotiate a smooth turn. To accomplish this, the flow divider-combiner must allow differential flows. The combine mode is often much less capable of handling

this differential flow during a steering event. Motor speed ratings must not be exceeded during the steering event. Consult the valve supplier for specifications on differential flow in both divide and combine modes of operation.

4.10.7 Motor Speed Synchronization (Non-Propel)

Several non-propel types of applications require two (or more) motors operating at the same speed to correctly perform a work function. Depending on the output speed accuracy required, a flow divider-combiner may or may not be an acceptable means of accomplishing the objective.

The system performance will be limited by the volumetric tolerance stack-up of the motors and the divider-combiner. This includes the slight difference in motor displacement, the slight difference in motor volumetric efficiency, and the flow division capability of the divider-combiner. Because the flow divider is not a perfect device, it will not send exactly the same volume of fluid to the two motors. Because the motors

are not perfect devices, their speed given the same volume of fluid will be slightly different. As the system loads up, the motor volumetric efficiency of the two motors will decrease at slightly different rates. The result of these imperfections is that the two motors will probably not rotate at exactly the same speed. The acceptability of a system can only be determined by testing.

If a system requires a high level of accuracy, the best way to control two (or more) motors at the same output speed is with a one pump per motor. A micro-controller used in conjunction with speed sensors at the motors and EDC's at the pumps would provide the best synchronization.

4.10.8 Recommendations for Trouble-Free Circuit

Several circuit problems may arise when using flow dividers in closed loop hydraulics. These problems seem to be very sensitive to system dynamics and do not surface on each and every installation. The severity of the problems may fluctuate on a given machine with fluid temperature, vehicle speed, and grade. To combat these problems, the following recommendations are provided with Figure 4-20 to help design trouble-free circuits.

- The loop flushing valve **must** be located between the pump and the flow divider-combiner per the discussions presented above.
- Bypass the divider-combiner in the combine mode of operation with either check valves or ON/OFF valves. Having eliminated the combiner from the circuit, it will provide no motor speed regulation or limitation in the event of a wheel spin-out.
- Provide anti-cavitation check valves to the inlet side of both motors. This is necessary because the volumetric tolerance stack-up of the motors and the divider-combiner can cause one motor to cavitate in a typical propel application. If one motor requires slightly more fluid to rotate at the same speed as its partner motor (and if the two motors are tied together by the ground) then it will starve. Because its inlet pressure drops, the motor is doing no work. On some machines this could go unnoticed indefinitely. The same scenario occurs if the divider-combiner were to split the flow unevenly.
- Work closely with a divider-combiner supplier to select the most appropriate valve for the application. Take into account flow and pressure ratings, pressure and temperature compensation, differential flow capability, and volumetric effectiveness.
- Test the vehicle extensively in all modes of operation to ensure correct functional performance as well as correct circuit performance.

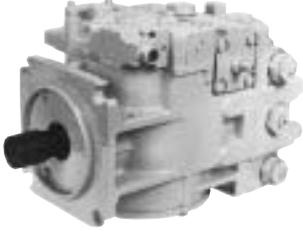
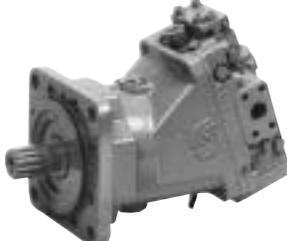
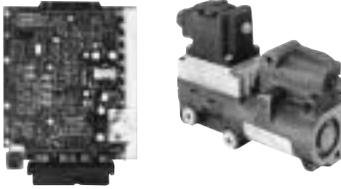
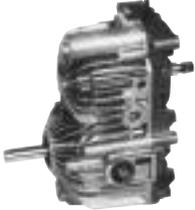
Hydraulic Power Systems

SAUER-SUNDSTRAND Hydraulic Power Systems - Market Leaders Worldwide

SAUER-SUNDSTRAND is a world leader in the design and manufacture of Hydraulic Power Systems. Research and development resources in both North America and Europe enable SAUER-SUNDSTRAND to offer a wide range of design solutions utilizing hydraulic power system technology.

SAUER-SUNDSTRAND specializes in integrating a full range of system components to provide vehicle designers with the most advanced total-design system.

SAUER-SUNDSTRAND is Your World Source for Controlled Hydraulic Power Systems.

 <p>Heavy Duty Axial Piston Pumps and Motors</p>	 <p>Heavy Duty Bent Axis Variable Motors</p>	 <p>Cartridge Motors/ Compact Wheel Drives</p>
 <p>Medium Duty Axial Piston Pumps and Motors</p>	 <p>Microcontrollers and Electrohydraulic Controls</p>	 <p>Hydrostatic Transmission Packages</p>
 <p>Open Circuit Axial Piston Pumps</p>	 <p>Gear Pumps and Motors</p>	 <p>Genuine Service Parts</p>

Worldwide Service Support

SAUER-SUNDSTRAND provides comprehensive worldwide service for its products through an extensive network of Authorized Service Centers strategically located in all parts of the world.

Look to SAUER-SUNDSTRAND for the best in WORLDWIDE SERVICE.

