SAUER DANFOSS

Applications Manual

Section 1

Selection of Driveline Components

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1.1 Selection of Driveline Components

1.1.1 Introduction

This section presents a method of sizing driveline components for typical closed loop hydrostatic transmissions. Although the method was developed for propel systems, it may be used for winch or reel applications, or other circuits with very slight modifications. The terminology used in this procedure also tends to reflect off-highway mobile applications.

It is assumed that the specific functional requirements of the application are defined, and that the fundamental design parameters have been established for each mode of operation. These typically include vehicle speed, gradeability, useful life, vehicle weight, and drive configuration. It is also assumed that, from these parameters, required engine power has been established.

The **goal** of this design method is to optimize performance and cost of the driveline system by selecting appropriate driveline components. Typically, smaller hydraulic components cost less than larger components, but have lower torque capability.

Because hydraulic unit life is highly dependent on system pressure, it is recommended that a design maximum and continuous pressure be established based on the **required life** of the driveline. Section 2, "Pressure and Speed Limits," covers this subject in detail.

Figure 1-1 shows the components typically found in a closed loop hydrostatic drive system as well as the design parameters and degree of design flexibility associated with each component. Because the driveline design includes so many variables, each dependent on the others, and because final component selections are ultimately limited by product availability, several iterations of this procedure may be required before arriving at the optimum system.

The **sizing procedure** starts with values for the machine maximum torque and required speed. From these values, a hydraulic motor size can be selected. This motor selection is then made compatible with ratings of available output gear drives. From a motor size, a pump size can be established. The pump must be capable of accepting the required input power, and it must be compatible with the pump drive means. It must also be large enough to provide sufficient flow to the drive motor to attain the required speed.

Driveline Element Selection

Driving Element	Design Parameter	Design Flexibility
Engine	Power Speed	No
Gearing	Ratio	Sometimes
Pump	Size Pressure Speed	Yes
Motor	Size Pressure Speed	Yes
Gearing	Ratio	Usually
Load	Speed Weight	No

Figure 1-1

Optimizing the size of the hydraulic units depends on selecting the correct gear ratios. By matching machine corner power with motor corner power, the required unit sizes can be quickly determined. The gear ratios can usually be adjusted to provide some optimization of hydraulic unit component size.

Along with equations presented throughout this article, a sizing flowchart is included near the end of the article to assist with sizing. The flowchart details the procedure described in this article and includes numerous "design check" conditions to determine acceptability of the design values. Note that design limits for associated mechanical components are not identified. Machine designers should verify that all design parameters are met for all driveline components.

While the methods described in this article may be useful, they do not represent the only approach to sizing hydraulic units. Contact Sauer-Sundstrand if questions of interpretation exist.

1.1.2 Machine Corner Power

The first step in the sizing process is to determine the value referred to as **machine corner power (CP)**. Although the concept of “corner power” (CP) is abstract and is not normally an attainable value of transmission power, it is useful in the design process because it provides a useful indication of transmission size and ratio requirements. Mathematically, it represents the highest levels of torque and speed performance required of the machine. Figure 1-2 illustrates the concept of machine corner power.

The equations for calculating corner power are shown below. For rotary drives, the input values into the equation are the required maximum output torque and maximum output speed of the machine. For propel drives, the input values are maximum tractive effort and maximum vehicle speed.

Tractive effort refers to the amount of force available at the wheel or wheels of the vehicle and represents the maximum possible pull a vehicle could exert if it had no resistance to movement. Section 1.2 describes tractive effort in more detail.

Ideally, tractive effort or output torque requirements should be derived from actual tests of the machine. However, for establishing tractive effort design values, an analytical approach based on machine parameters and functional modes of operation has been used successfully (see Section 1.2). For multi-speed drives (e.g., work mode and travel mode), corner power must be calculated for all operating ranges.

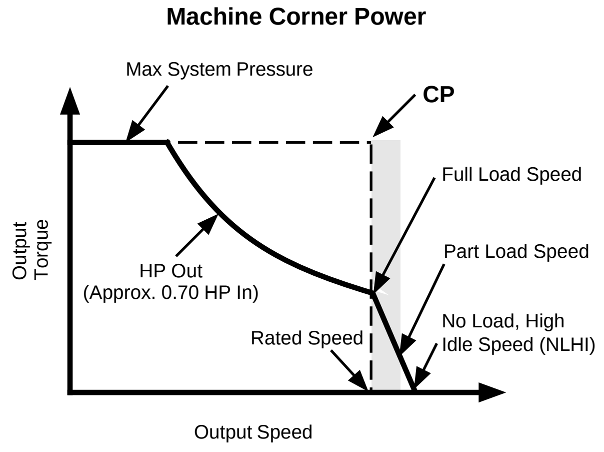


Figure 1-2

Machine corner power (CP) is determined by the maximum torque and maximum output speed required. It is normally greater than actual transmission output power. Maximum output speed is assumed to be at engine rated speed. However, under part load conditions slightly higher speed may be obtained.

SI System	US System	Description	
Rotary Drives			
1) Machine CP = $\frac{TQ \cdot ND}{9549}$	Machine CP = $\frac{TQ \cdot ND}{63\,025}$	CP = machine corner power TQ = maximum drive output torque ND = maximum drive output design speed	kW (hp) Nm (in lbf) rpm
Propel Drives			
Machine CP = $\frac{TE \cdot S}{3600}$	Machine CP = $\frac{TE \cdot S}{375}$	TE = maximum vehicle tractive effort S = maximum vehicle design speed	N (lbf) kph (mph)

1.1.3 Variable or Fixed Motor

Because the machine corner power is an expression of maximum torque and maximum speed, it can be used to establish the **effective transmission ratio (TR)** required to meet system demands. The effective transmission ratio is the ratio of required vehicle corner power to normal transmission output power. This ratio is similar to the ratio spread of a similarly sized mechanical transmission and indicates the amount of hydrostatic ratio which is required. Systems with high transmission ratios normally benefit from variable or two-position drive motors.

Although the normal output power is probably not established yet in the design process, approximate its value to be 70% of normal input power. The normal input power to meet the machine requirements should already be established. For drives with variable load cycles, determine the normal input power by deducting average power to other functions from maximum engine power available to the drive.

The rule for selecting a fixed or a variable drive motor is as follows:

- If TR is greater than 4, use a variable motor
- If TR is less than 2, use a fixed motor
- If TR is between 2 and 4, evaluate both variable and fixed motors for suitability.

SI / US System

2) $$TR = \frac{\text{Machine CP}}{0.7 \cdot HP}$$

TR < 2, use fixed motor
TR > 4, use variable motor

Description

TR = effective transmission ratio
HP = normal input power kW (hp)

1.1.4 Motor Selection

Calculate the required motor corner power from machine corner power and driveline efficiency using Equation (3) below. This establishes the minimum motor size capable of meeting the power requirements of the machine. For multi-speed drives, use the largest corner power for each of the operating ranges.

For transmission circuits using multiple drive motors, the required motor corner power should be interpreted as the required corner power at each motor.

Use Equation (4) to calculate the motor corner power capability based on the design maximum pressure and the design maximum speed required to achieve the desired life of the motor.

Design maximum pressure is the maximum pressure at which the motor is intended to operate to meet the required life. The design maximum pressure may or may not be the same as the maximum pressure rating published in the product literature. Published ratings for maximum pressure assume the pressure is to occur a small percentage of operating time, usually less then 2% of the total, and will result in "normal" life. For applications in which the maximum pressure will occur over a significant portion of the duty cycle, or applications in which additional life is required, the design maximum pressure should be assigned a value less than the published rating for maximum pressure.

Design maximum speed is the maximum speed at which the motor is intended to operate to meet the required life. Although speed has less effect on life than pressure, lower operating speeds will have the effect of increasing life. The value for the design maximum speed must never exceed the maximum speed rating published in the product literature, and will usually be less to allow for motor speed increases as a result of reduced-load or no-load conditions (see Figure 1-2). Section 2, "Pressure and Speed Limits," provides additional information concerning pressure and speed limits with respect to component life.

Ideally, values for the design maximum pressure and design maximum speed would be used in Equation (4) to determine motor CP capability. However, this is difficult at this stage of the sizing process because both the motor displacement and final drive ratio are unknown. Despite this limitation, the next step is to choose a logical motor displacement based on the required motor CP. Table 1-1 can be used as an aid in the preliminary motor selection. Choose a motor displacement with a motor CP at least as large as the required motor CP calculated using Equation (3).

Note that motor CP values in Table 1-1 are based on maximum rated pressure and continuous rated speed. Although the table may eliminate the need to calculate motor CP, use it only as a reference to help make preliminary motor selections. The assumed values for maximum pressure, especially may not provide sufficient life for every application. When in doubt, use Equation (4) to calculate motor capability.

Equation (A) serves as a design check to ensure that a motor with sufficient corner power capability is selected. Motor selection based on corner power results in the smallest motor capable of transmitting the required machine power while achieving system life requirements.

SI System

3) $$\text{Required Motor CP} = \frac{\text{Machine CP}}{E \cdot \#}$$

4) $$\text{Motor CP} = \frac{0.95 \cdot DM \cdot NM \cdot PM}{600\,000}$$

US System

$$\text{Required Motor CP} = \frac{\text{Machine CP}}{E \cdot \#}$$

$$\text{Motor CP} = \frac{0.95 \cdot DM \cdot NM \cdot PM}{396\,000}$$

A) **Design Check:** Motor CP $\geq$ Required Motor CP

Description

CP	= corner power	kW (hp)
E	= final drive efficiency	(%)
#	= number of motors	
DM	= maximum motor displacement	cc (in^3)/rev
NM	= design maximum speed	rpm
PM	= design maximum pressure	bar (psi)

For **variable motor systems**, the transmission CP is determined only by the motor. For various pump sizes, actual applied motor CP may be varied by adjusting the minimum motor angle.

For **fixed motor systems**, the transmission CP is ultimately determined by the pump speed and displacement. Although fixed motor CP must be large enough to accommodate the maximum load and speed, the pump must be large enough to drive the motor at the required design speed. An additional sizing exercise may be required for fixed motor systems after the pump selection has been made.

For either variable or fixed motor systems, it may be necessary to increase the motor size if proper output gearing is not available. Gearing must accommodate both the desired ratio and maximum motor speed, in addition to meeting the torque requirements.

Motor Corner Power Capabilities

		Fixed & Variable Motors at Max Displacement		Variable Motors at Min Displacement	
Series	**Maximum Pressure bar (psid)**	**Continuous Rated Speed (RPM)**	**Motor Corner Power kW (hp)**	**Continuous Rated Speed (RPM)**	**Motor Corner Power kW (hp)**
Series 15	276 (4500)	4000	29 (39)		
Series 40					
M25	345 (5000)	4000	54 (72)	5300	71 (95)
M35	345 (5000)	3600	69 (92)	4800	92 (123)
M46	345 (5000)	3600	90 (121)	4400	110 (148)
Series 20					
21	414 (6000)	3000	101 (136)		
22	414 (6000)	2800	128 (172)	4350	199 (267)
23	414 (6000)	2600	152 (203)	4100	239 (320)
24	414 (6000)	2400	187 (250)		
25	414 (6000)	2200	239 (320)		
26	414 (6000)	1900	283 (379)		
27	414 (6000)	1600	350 (469)		
Series 90					
42	483 (7000)	4200	135 (181)		
55	483 (7000)	3900	164 (219)	4600	193 (259)
75	483 (7000)	3600	206 (276)	4250	243 (326)
100	483 (7000)	3300	252 (338)		
130	483 (7000)	3100	308 (413)		
Series 51					
60	483 (7000)			5600	256 (344)
80	483 (7000)			5000	308 (413)
110	483 (7000)			4500	378 (507)
160	483 (7000)			4000	492 (660)
250	483 (7000)			3400	650 (871)

*These values for corner power capability are based on **maximum** pressure and **rated** (continuous) speed ratings. A 95% volumetric efficiency is assumed for the power calculation. **Refer to Section 2 for detailed information on ratings of units and expected life.***

Table 1-1

1.1.5 Final Drive Selection

After the motor is initially sized, calculate the required final drive ratio. One of two approaches can be taken to determine this ratio. Both take into account the design maximum and continuous pressures allowed to meet the life requirements of the machine (see Section 2). The two methods are as follows:

1. Using the first method as presented in the sizing flowchart (p. 14), size the final drive using the design maximum pressure and the maximum torque requirement. Use Equations (5) on the following page for this calculation. After the pump is sized and all speed conditions have been met, estimate the continuous pressure (see flowchart) and compare with the maximum design continuous pressure.

2. As an alternate method, calculate the final drive required for all modes of operation (travel mode, work mode, etc.). Calculate the final drive from the assumed pressure and torque requirements for each operating mode. For "worst case" or intermittent modes of operation, use the design maximum pressure along with the tractive effort or torque requirement to obtain a value for the final drive ratio. Use design continuous pressure for "typical" or continuous modes of operation, and calculate required final drive ratios for these modes as well. Select the largest final drive from the values calculated for the various operating modes. (Note that for variable or two-position motors, only final drives from those modes utilizing maximum motor displacement can be calculated, since the motor minimum displacement is not yet known).

Regardless of the method used to determine the final drive, the next step is to check motor speed limits using the limits obtained from Section 2.1.4. Motor speed will usually be satisfactory unless the final drive is significantly higher than required. (Gearbox limits must also be met). Equations (6) is used to determine the required motor speed at maximum motor displacement based on the final drive calculated in equation (5). For fixed displacement motors, the maximum motor displacement referred to in the equation is simply the displacement of the motor. For variable motors, use the displacement at the maximum motor angle. Use design check (C) to ensure that the speed limit of the motor is not exceeded. If a variable motor is specified, use equation (7) and design check (D) to determine if the speed required at the minimum motor displacement exceeds the maximum reduced angle speed limit. As explained in Section 2, "Pressure and Speed Limits," the maximum speed limit of a variable increases with decreasing angle, up to a certain value (the maximum reduced angle speed limit or "cutoff" point of the speed/angle curve). At low motor angles, any decrease in angle does not result in a greater maximum speed limit. Note that the reduced angle speed limits cannot be checked until the pump displacement and minimum motor displacement have been established. (This will be done in subsequent steps of this procedure). However, if the speed exceeds the limit associated the smallest possible motor angle (the "cutoff" point of the speed/angle curve), then increase the motor size.

Refer to Section 2 for more information concerning speed limits.

SI System **US System** **Description**

Rotary Drives

5) SI: $$\text{Required FD} = \frac{TQ \cdot 20\pi}{0.95 \cdot DM \cdot PM \cdot E}$$

US: $$\text{Required FD} = \frac{TQ \cdot 2\pi}{0.95 \cdot DM \cdot PM \cdot E}$$

Propel Drives

SI: $$\text{Required FD} = \frac{TE \cdot LR \cdot 0.2\pi}{0.95 \cdot DM \cdot PM \cdot E \cdot \#}$$

US: $$\text{Required FD} = \frac{TE \cdot LR \cdot 2\pi}{0.95 \cdot DM \cdot PM \cdot E \cdot \#}$$

B) **Design Check:** $FD \geq \text{Required FD}$

Rotary Drives

6) SI: $NMR = FD \cdot NDM$

US: $NMR = FD \cdot NDM$

Propel Drives

SI: $$NMR = \frac{FD \cdot SM \cdot 2.65}{LR}$$

US: $$NMR = \frac{FD \cdot SM \cdot 168}{LR}$$

C) **Design Check:** $NMR \leq NML$

Rotary Drives

7) SI: $NVR = FD \cdot NVD$

US: $NVR = FD \cdot NVD$

Propel Drives

SI: $$NVR = \frac{FD \cdot SV \cdot 2.65}{LR}$$

US: $$NVR = \frac{FD \cdot SV \cdot 168}{LR}$$

D) **Design Check:** $NVR \leq NVL$

DM	=	max motor displacement	cc (in^3)
E	=	final drive efficiency	(%)
FD	=	final drive ratio	
LR	=	wheel loaded radius	m (in)
NDM	=	non-propel design speed at max angle	rpm
NML	=	motor speed limit at max angle	rpm
NMR	=	req'd motor speed at max angle	rpm
NVD	=	non-propel design speed at min angle	rpm
NVR	=	req'd motor speed at min angle	rpm
NVL	=	motor speed limit at min angle	rpm
PM	=	maximum pressure	bar (psid)
SM	=	vehicle speed req'd at max angle	kph (mph)
SV	=	vehicle speed req'd at min angle	kph (mph)
TE	=	vehicle tractive effort	N (lbf)
TQ	=	max drive output torque	Nm (in•lbf)
#	=	number of motors	

1.1.6 Input Gearing

The use of input gearing is usually determined by the machine configuration. For vehicles with multiple hydraulic systems, use of an input splitter box is common. They are usually available with various ratios to accommodate pump speed requirements. For machines with only a single hydrostatic system, (or machines utilizing tandem pumps) a direct drive pump may be appropriate, in which case the pump speed is the same as the prime mover speed.

Use Equation (8) to determine the relationship between engine speed, pump speed, and input gear ratio.

SI / US System

8) $NP = NE \cdot IR$

Description

NP = maximum pump design speed rpm
NE = prime mover design speed rpm
IR = pump input ratio

1.1.7 Pump Selection

Pump sizing consists of selecting a pump that will meet the flow (speed) requirements of the motor or motors in the system.

Use equation (9) to determine the required pump displacement. This calculation is based on an assumed pump input speed. Select a pump displacement at least as large as the required displacement. Also, check that the pump speed does not exceed the limit established for the life requirement. If the speed limit is exceeded, choose a different pump and calculate the input speed required and the corresponding input ratio using equations (10) and (11).

With a pump displacement selected, calculate the actual motor speed. The actual speed will usually be slightly higher than the required motor speed because the pump selected above will usually have a displacement slightly greater than the displacement required.

For a fixed motor, determine the actual motor speed and compare with its speed limit using equation (12) and design check (G). Note that equation (12) includes a calculation for an overrunning condition. An overrunning condition is characterized by a speed increase in the pump (and consequently the motors), typically by as much as 15%. The condition is especially common during downhill operation. The speed-up is compounded by the fact that, in addition to the pump speed increase, the volumetric efficiencies are "reversed." This reversal is due to the motor behaving as a pump in overrunning conditions.

SI / US System

9) $$\mathrm{DPR} = \frac{\mathrm{NMR} \bullet \mathrm{DM} \bullet \#}{(0.95)^2 \bullet \mathrm{NP}}$$

D) **Design Check:** $\mathrm{DP} \geq \mathrm{DPR}$

E) **Design Check:** $\mathrm{NP} \leq \mathrm{NPL}$

10) $$\mathrm{NPR} = \frac{\mathrm{DM} \bullet \mathrm{NMR} \bullet \#}{(0.95)^2 \bullet \mathrm{DP}}$$

11) $$\mathrm{IR} = \frac{\mathrm{NPR}}{\mathrm{NE}}$$

Without Overrunning Condition:

12) $$\mathrm{NM} = \frac{\mathrm{DP} \bullet \mathrm{NE} \bullet \mathrm{IR} \bullet (0.95)^2}{\mathrm{DM} \bullet \#}$$

With Overrunning Condition:

$$\mathrm{NM} = \frac{\mathrm{DP} \bullet \mathrm{NE} \bullet \mathrm{IR} \bullet 1.15}{(0.95)^2 \bullet \mathrm{DM} \bullet \#}$$

G) **Design Check:** $\mathrm{NM} \leq \mathrm{NML}$

Description

DM	=	max motor displacement	cc (in^3)/rev
DP	=	max pump displacement	cc (in^3)/rev
DPR	=	req'd max pump displacement	cc (in^3)/rev
IR	=	pump input ratio	
NMR	=	req'd motor speed at max angle	rpm
NE	=	prime mover design speed	rpm
NM	=	design maximum speed	rpm
NML	=	motor speed limit at max angle	rpm
NP	=	max pump design speed	rpm
NPL	=	pump speed limit at max angle	rpm
NPR	=	req'd pump speed	rpm
#	=	number of motors	

For a variable motor, the procedure for assuring the speed limit is not exceeded is somewhat more involved. The steps are as follows:

1. Determine if the maximum displacement speed limit is exceeded using the method above (12).
2. Determine the motor minimum displacement using equation (13).
3. Calculate the angle associated with this displacement using equation (14). Select an available minimum angle using design check (H) and determine the actual motor speed using equation (15).
4. Determine the reduced angle speed limit from Section 2.1.3 or by using equation (16). Use design check (I) to ensure that the minimum angle speed limit is not exceeded.

The flowchart in section 1.1.9 details the above procedure.

SI / US System

13)

$$DV = \frac{DP \bullet NE \bullet IR \bullet (0.95)^2}{NVR \bullet \#}$$

14) **All Swashplate Motors:**

$$TANV = TANM\ (DV / DM)$$

$$AV = ARCTAN\ (TANV)$$

Series 51 Bent-Axis Motors:

$$SINV = 0.53\ (DV / DM)$$

$$AV = ARCSIN\ (SINV)$$

H) **Design Check:** $AV \leq$ Min Available

15) **Without Overrunning Condition:**

$$NV = \frac{DP \bullet NE \bullet IR \bullet (0.95)^2}{NVR \bullet \#}$$

With Overrunning Condition:

$$NV = \frac{DP \bullet NE \bullet IR \bullet (1.15)}{(0.95)^2 \bullet DV \bullet \#}$$

16) **All Swashplate Motors:**

$$NVL = NML \bullet (DM / DV)^{1/2}$$

Series 51 Bent-Axis Motors:

$$NVL = NML \bullet (0.53 / SV)$$

I) **Design Check:** $NVL \geq$ Max Reduced Angle Value

Description

AV = min angle for a variable motor Degrees
DM = max motor displacement cc (in^3)/rev
DP = max pump displacement cc (in^3)/rev
DPR = req'd max pump displacement cc (in^3)/rev
DV = min motor displacement cc (in^3)/rev
IR = pump input ratio
NE = prime mover design speed rpm
NM = motor speed at max angle rpm
NML = motor speed limit at max angle rpm
NMR = req'd motor speed at max angle rpm
NV = motor speed at min angle rpm
NVL = motor speed limit at min angle rpm
NVR = req'd motor speed at min angle rpm
NP = max pump design speed rpm
NPL = pump speed limit at max angle rpm
SINM = sine of motor at max angle
SINV = sine of motor at min angle
SV = vehicle speed req'd at min angle kph (mph)
TANM = tangent of motor at max angle
TANV = tangent of motor at min angle
\# = number of motors

1.1.8 Continuous Pressure

The final (but crucial) step in the procedure is to estimate the continuous pressure based on the components selected.

Figure 1-3 shows the relationship between system pressure and system flow for a hydrostatic pump. The figure shows that the continuous system pressure usually occurs near maximum pump flow and normal input power.

The flowsheet equations provide a check to ensure the continuous pressure is below the pressure required to meet the design life.

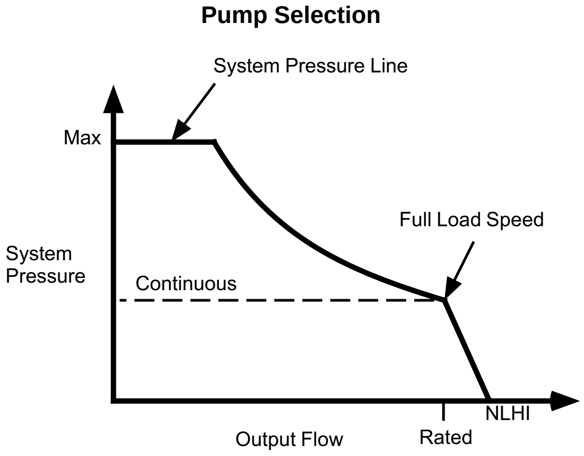


Figure 1-3

Continuous system pressure at maximum pump flow may be estimated from the normal input power to the drive. For many systems, the continuous pressure determined in this manner is a good indicator of typical system pressure experienced in the drive.

	SI System	US System
17)	$PC = \frac{HP \bullet 600\,000}{DP \bullet NE \bullet IR}$	$PC = \frac{HP \bullet 396\,000}{DP \bullet NE \bullet IR}$
	Propel, Motor at Max Angle	
18)	$FD = \frac{NML \bullet LR}{2.65 \bullet SM}$	$FD = \frac{NML \bullet LR}{168 \bullet SM}$
	Non-Propel, Motor at Max Angle	
	$FD = \frac{NML}{NMD}$	$FD = \frac{NML}{NMD}$
	Propel, Motor at Min Angle	
	$FD = \frac{NML \bullet LR}{2.65 \bullet SV}$	$FD = \frac{NML \bullet LR}{168 \bullet SV}$
	Non-Propel, Motor at Min Angle	
	$FD = \frac{NML}{NVD}$	$FD = \frac{NML}{NVD}$

Description		
DP	= max pump displacement	cc (in^3)/rev
FD	= final drive ratio	
HP	= normal power input to drive	kW (hp)
IR	= pump input ratio	
LR	= wheel loaded radius	m (inch)
NE	= prime mover design speed	rpm
NMD	= non-propel speed at max angle	rpm
NML	= motor speed limit at max angle	rpm
NVD	= non-propel speed at min angle	rpm
PC	= estimated continuous pressure	bar (psid)
SM	= vehicle speed req'd at max angle	kph (mph)
SV	= vehicle speed req'd at min angle	kph (mph)

1.1.9 Sizing Flowchart

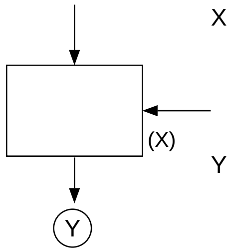
The flowchart included in this section is designed to be used as a sizing "algorithm" to assist in the selection of system components. It provides a concise step-by-step run-through of the sizing process. It is intended to accompany the previous sections and to expand the equations presented with the text.

The symbol designations used in the flowchart are explained in the upper righthand box of the first page. The equations (p. 19) used to calculate the quantities are included following the flowchart, along with definitions of the symbols.

Be aware that the flowchart does **not** consider any torque / speed limits associated with various mechanical components, e.g., pump drives or final drive gearboxes.

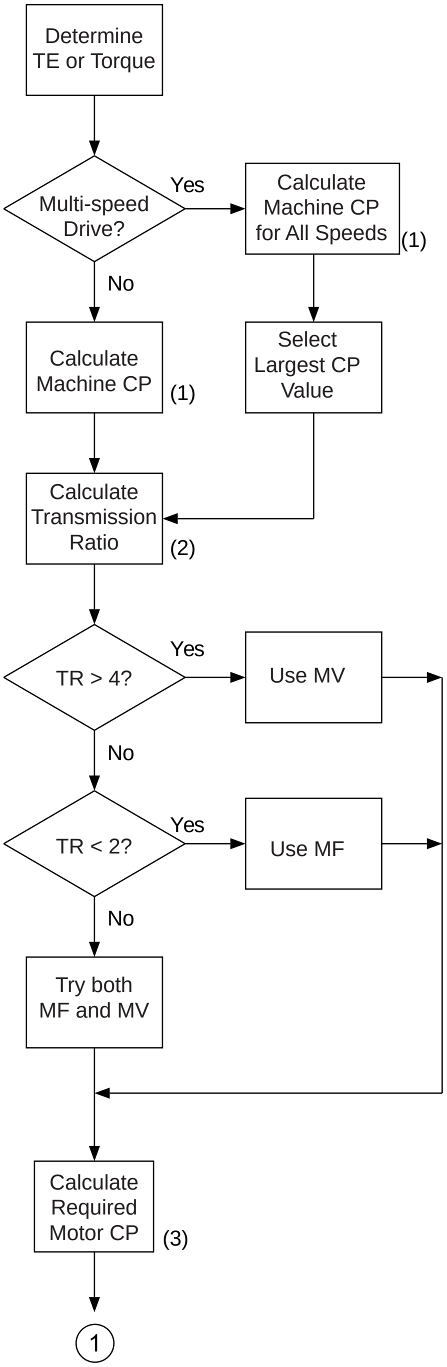
Sizing Flowchart

Use the following flowchart to assist in sizing a hydrostatic transmission. Note that the required equations are shown at the lower right corner of the boxes and can be found on the tables following the flowchart.



X = Equation number required as shown on p.19.

Y = Flow chart reference numbers (numbers that jump to other reference points on the flowchart).

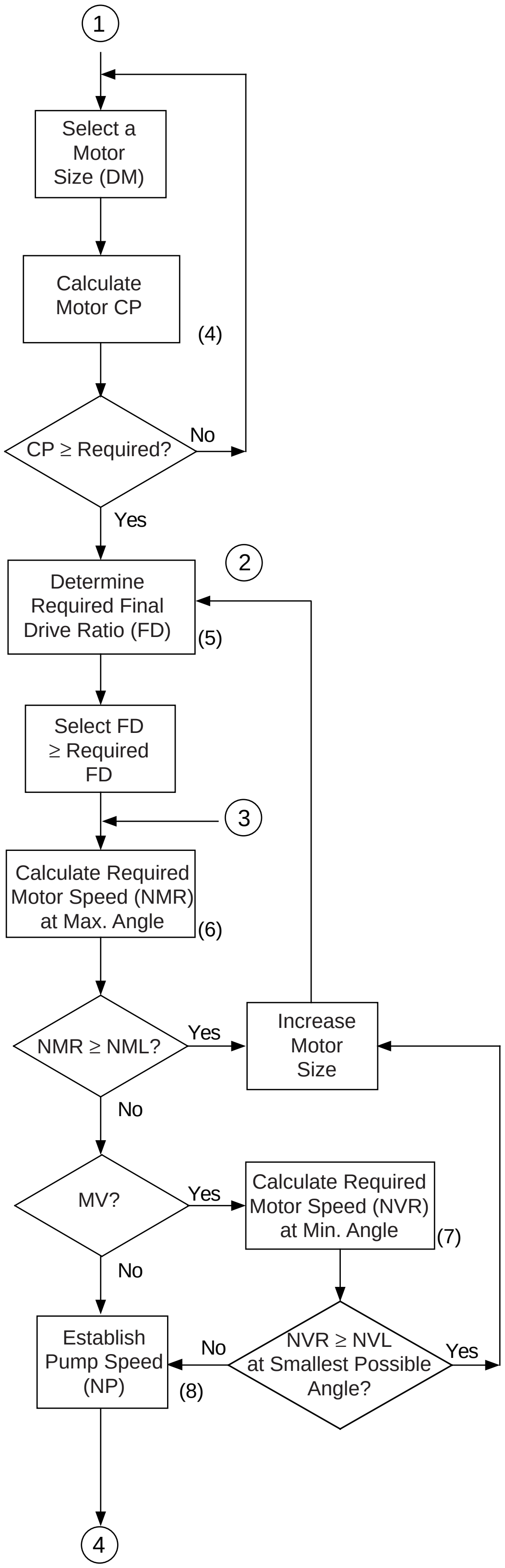


Begin by calculating the required effort or torque.

For multi-speed drives, machine corner (CP) power must be calculated for each drive.

Transmission Ratio (TR) indicates the need for a fixed or variable motor.

Sizing begins at the motor. First, determine the required motor corner power.

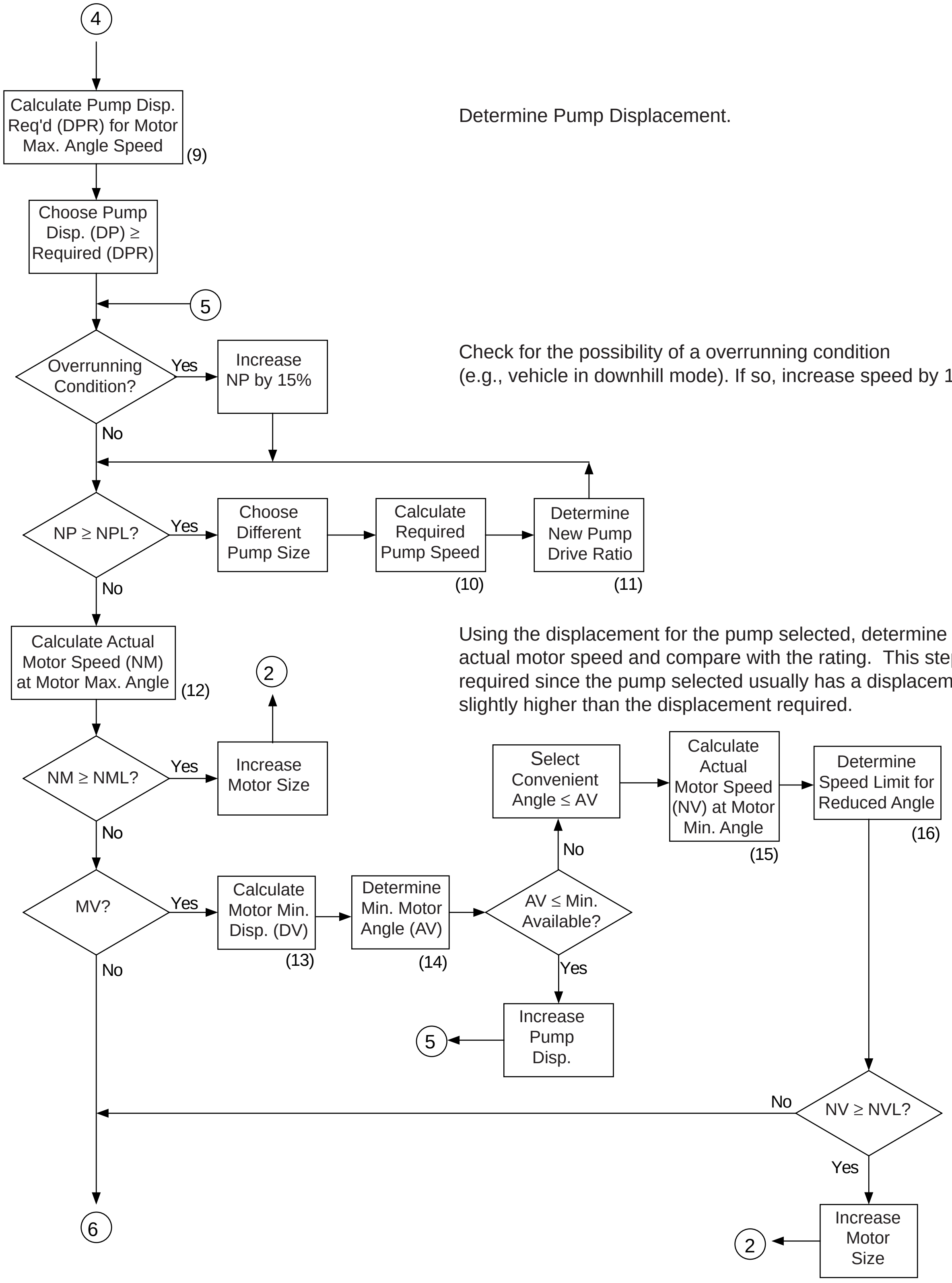


Select a likely motor displacement and either calculate its corner power capability or use Table 1-1. If this value is *less than* the required corner power, select a *larger* motor.

Determine the final drive ratio (FD) required. This calculation is based on the maximum motor displacement.

Calculate the required motor speed at maximum angle and compare with the maximum speed limit allowed.

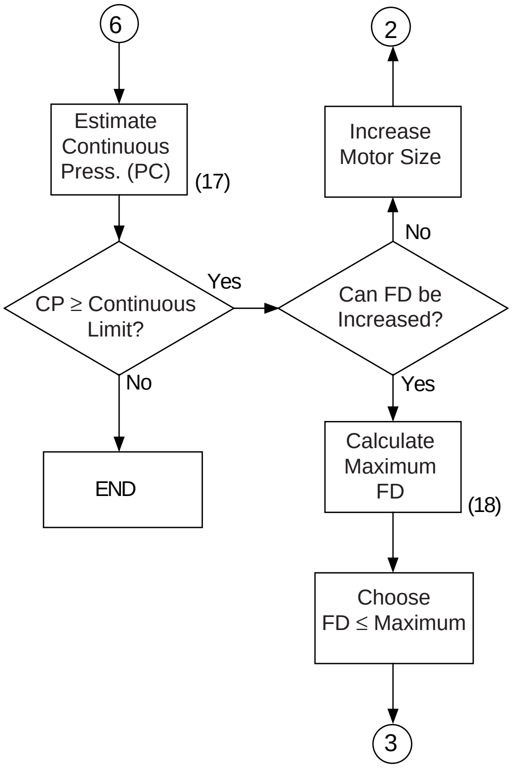
For variable motors, calculate the required motor speed at the minimum angle and compare with the maximum speed limit at the smallest possible motor angle. Note that at this point in the sizing procedure, the required minimum angle is unknown, so the reduced angle speed limit is also unknown. However, the unknown speed limit will be less than the speed limit at the smallest possible motor angle.



Determine Pump Displacement.

Check for the possibility of a overrunning condition (e.g., vehicle in downhill mode). If so, increase speed by 15%.

Using the displacement for the pump selected, determine the actual motor speed and compare with the rating. This step is required since the pump selected usually has a displacement slightly higher than the displacement required.



Continuous pressure can be estimated based on the input horsepower.

If the final drive can be increased to reduce pressure, calculate the largest final drive that will keep motor speed under limits. Reselect a final drive no larger than this value.

Step	Equations Required		Comments
	Metric System	US System	
1	$\text{Machine CP} = \frac{TE \bullet S}{3600}$ $\text{Machine CP} = \frac{TQ \bullet ND}{95496}$	$\text{Machine CP} = \frac{TE \bullet S}{375}$ $\text{Machine CP} = \frac{TQ \bullet ND}{63025}$	Propel Drive Non-Propel Drive
2	$TR = \frac{\text{Machine CP}}{0.7 \bullet HP}$	Same	
3	$\text{Required Motor CP} = \frac{\text{Machine CP}}{E \bullet \#}$	Same	
4	$\text{Motor CP} = \frac{0.95 \bullet DM \bullet NM \bullet PM}{600\,000}$	$\text{Motor CP} = \frac{0.95 \bullet DM \bullet NM \bullet PM}{396\,000}$	
5	$\text{Required FD} = \frac{TE \bullet LR \bullet 0.2\pi}{0.95 \bullet DM \bullet PM \bullet E \bullet \#}$ $\text{Required FD} = \frac{TQ \bullet 20\pi}{0.95 \bullet DM \bullet PM \bullet E}$	$\text{Required FD} = \frac{TE \bullet LR \bullet 2\pi}{0.95 \bullet DM \bullet PM \bullet E \bullet \#}$ $\text{Required FD} = \frac{TQ \bullet 2\pi}{0.95 \bullet DM \bullet PM \bullet E}$	Propel Drive Non-Propel Drive
6	$NMR = \frac{FD \bullet SM \bullet 2.65}{LR}$ $NMR = FD \bullet NDM$	$NMR = \frac{FD \bullet S \bullet 168}{LR}$ Same	Propel Drive Non-Propel Drive
7	$NVR = \frac{FD \bullet SV \bullet 2.65}{LR}$ $NVR = FD \bullet NDV$	$NVR = \frac{FD \bullet SV \bullet 168}{LR}$ Same	Propel Drive Non-Propel Drive
8	$NP = NE \bullet IR$	Same	
9	$DPR = \frac{NMR \bullet DM \bullet \#}{(0.95)^2 \bullet NP}$	Same	
10	$NPR = \frac{DM \bullet NMR \bullet \#}{DP \bullet (0.95)^2}$ (choose $DP \geq DPR$)	Same	
11	$IR = \frac{NPR}{NE}$	Same	

Step	Equations Required		Comments
	Metric System	US System	
12	$NM = \frac{DP \bullet NE \bullet IR \bullet (0.95)^2}{DM \bullet \#}$	Same	Normal Operation
	$NM = \frac{DP \bullet NE \bullet IR \bullet 1.15}{(0.95)^2 \bullet DM \bullet \#}$	Same	Overrunning Conditions
13	$DV = \frac{DP \bullet NE \bullet IR \bullet (0.95)^2}{NVR \bullet \#}$	Same	
14	$TANV = TANM \bullet (DV / DM)$	Same	All Swashplate Motors
	$AV = \text{Arctan}\ (TANV)$		
	$SINV = 0.53 \bullet (DV / DM)$	Same	Series 51 Bent Series Motors
	$AV = \text{Arcsin}\ (SINV)$		
15	$NV = \frac{DP \bullet NE \bullet IR \bullet (0.95)^2}{DV \bullet \#}$	Same	Normal Operation
	$NV = \frac{DP \bullet NE \bullet IR \bullet 1.15}{(0.95)^2 \bullet DV \bullet \#}$	Same	OverrunningConditions
16	$NVL = NML \bullet (DM / DV)^{1/2}$	Same	All Swashplate Motors
	$NVL = NML \bullet (0.53 / SV)$	Same	Series 51 Bent-Axis Series Motors
	$NVL \geq$ Reduced Angle Value		
17	$PC = \frac{HP \bullet 600\ 000}{DP \bullet NE \bullet IR}$	$PC = \frac{HP \bullet 396\ 000}{DP \bullet NE \bullet IR}$	
18	$FD = \frac{NML \bullet LR}{2.65 \bullet SM}$	$FD = \frac{NML \bullet LR}{168 \bullet SM}$	Propel, Motor at Max Angle
	$FD = \frac{NML}{NMD}$	Same	Non-Propel, Motor at Max Angle
	$FD = \frac{NVL \bullet LR}{2.65 \bullet SV}$	$FD = \frac{NVL \bullet LR}{168 \bullet SV}$	Propel, Motor at Min Angle
	$FD = \frac{NVL}{NVD}$	Same	Non-Propel, Motor at Min Angle

Definition of Terms

The following list of terms describe the variables used in the sizing equations:

AV	Minimum angle for a variable motor	Degrees
CP	Corner power	kW (hp)
DM	Maximum motor displacement	cc (in^3)/rev
DV	Minimum motor displacement	cc (in^3)/rev
DP	Maximum pump displacement	cc (in^3)/rev
DPR	Required maximum pump displacement	cc (in^3)/rev
E	Final drive efficiency	%
FD	Final drive ratio	
HP	Normal power input to drive	kW (hp)
IR	Input ratio (pump speed /prime mover speed)	
LR	Wheel loaded radius (rolling radius)	m (inch)
ND	Design speed for non-propel	rpm
NMD	Non-propel design speed at motor max angle	rpm
NVD	Non-propel design speed at motor min angle	rpm
NE	Prime mover input speed (engine, electric motor)	rpm
NML	Motor speed limit at maximum angle	rpm
NPL	Pump speed limit	rpm
NVL	Motor speed limit at minimum angle	rpm
NM	Motor speed at maximum angle	rpm
NP	Pump speed	rpm
NV	Motor speed at minimum angle	rpm
NMR	Required motor speed at maximum angle	rpm
NPR	Required pump speed	rpm
NVR	Required motor speed at minimum angle	rpm
PC	Estimated continuous pressure	bar (psid)
PM	Maximum system pressure	bar (psid)
S	Maximum vehicle speed	kph (mph)
SM	Vehicle speed required with motor at max angle	kph (mph)
SINM	Sine of motor maximum angle	
SINV	Sine of motor minimum angle	
SV	Vehicle speed required with motor at min angle	kph (mph)
TE	Tractive effort requirement	N (lbf)
TANM	Tangent of motor maximum angle	
TQ	Torque requirement (non-propel)	Nm (in lbf)
TR	Transmission ratio	
TANV	Tangent of motor minimum angle	

1.2 Tractive Effort

For vehicle propel drives, motion resistance and required tractive effort are directly related to vehicle weight. For a particular class or type of vehicle, the ratio of tractive effort to vehicle weight is relatively constant. This term is commonly called a “pull ratio” and it is a convenient design parameter.

The elements constituting a particular class or type of vehicle are machine function, drive configuration, grade, and terrain. Values for motion resistance contributing to the pull ratio requirements have been estimated and are listed in Table 1-2. To establish required pull ratio, sum the motion resistance values for machine function, drive configuration, grade and rolling resistance. Calculate required tractive effort from pull ratio and vehicle weight.

PR = MF + DC + GR + RR

where

PR = Pull ratio
MF = Machine function motion resistance
DC = Drive configuration motion resistance
GR = Grade motion resistance
RR = Rolling resistance

TE = (PR) (WT)

where

TE = Vehicle tractive effort (lb)
WT = Vehicle weight (lb)

The tractive effort to weight ratio, or pull ratio, is the sum of all expected demands on vehicle motion resistance. We recommend verifying the tractive effort values which are used for design by actual vehicle test.

To determine **machine function** (MP) motion resistance, consider all functions and modes of operation separately. Usually, the functions performed in the worst ground conditions predominate. For transmissions with multi-speed mechanical gearboxes, designers should consider the functions performed for each range. This usually requires examining several possible work situations and selecting the one with the highest rolling resistance and/or grade.

The pull ratio listed for “propel forces main work drive” is approximate. For propel drives which interact with the main work drive (cutters, planers, etc.), it is appropriate to make an accurate determination of the required motion resistance by testing a working machine.

“Transport” mode should be used only for those machines, or specific modes of operation, in which traveling or carrying is the only requirement. It is assumed that the vehicle operates at a relatively constant speed in the transport mode.

The component of pull ratio due to **drive configuration** (DC) results from geometry effects when steering. The particular form of drive for the vehicle affects the motion resistance. “Skid steer” configurations imply turning with differential side-to-side torque and no variable geometry. “Dual path variable steer geometry” configurations are usually wheeled machines with a single trailing pivot or caster wheel. “Single path track” or “single path wheel” configurations imply a geometry adjustment of the ground engaging elements to achieve steering.

Pull Ratio Requirements for Vehicle Propel Drives

Machine Function	**MF**
Dozing (All Wheel / Track Drive)	.90
Drawbar (All Wheel / Track Drive)	.80
Drawbar (Single Axle Drive)	.60
Dig and Load (All Wheel / Track Drive)	.50
Propel Forces Main Work Drive	.30 (Typ)
Stop and Go Shuttle	.15
Transport (No Work Interaction)	.00
Drive Configuration	**DC**
Skid Steer Track	.40
Skid Steer Wheel	.30
Dual Path Variable Steer Geometry	.20
Single Path Track	.10
Single Path Wheel	.00
Grade (Intermittent)	**GR**
10% Grade	.10
20% Grade	.20
30% Grade	.29
40% Grade	.37
50% Grade	.45
60% Grade	.51
Rolling Resistance	**RR**
Sand	.25
Wet Soil, Mud	.20
Fresh Deep Snow	.16
Loose Soil, Gravel	.12
Grassy Field, Dry Cropland	.08
Packed Soil, Dirt Roadway	.05
Pavement	.02
Steel on Steel Rails	.004

Table 1-2

Pull Ratio may be used to determine tractive effort in vehicle propel drives. Pull ratios are based on working vehicle weight. In general, this is loaded weight. For vehicles having a separate transport mode, empty weight may be appropriate.

Motion resistance due to **grade** (GR) is a function of slope. Select the maximum grade at which the particular machine function is performed. The maximum grade is assumed to be intermittent, with the average grade one-half to two-thirds the maximum.

Rolling resistance (RR) affects motion resistance depending on the condition of the terrain. Rolling resistance values listed here are typical and may vary depending on location, particular conditions and drive configuration. These may be adjusted with more specific data. These values apply for typical rubber-tired vehicles. High flotation tires and tracked crawlers may show somewhat lower values in poor terrain.

Vehicle weight is the maximum weight for the function being considered. For most vehicles, this is the loaded weight. Empty weight may be appropriate for some transport modes. For shuttle and transport vehicles, maximum weight is the gross combined weight of power unit plus any towed trailer or wagon. For drawbar vehicles, maximum weight is only the power unit.

Typical minimum design values of pull ratio for some common vehicles have been determined and are listed in Table 1-3. These values may be useful for checking intended tractive effort requirements. Vehicle performance testing is highly recommended to verify suitability in an actual working environment.

Minimum Tractive Effort Requirements

	Assumed Operating Conditions		Minimum Pull Ratio	
Vehicle Type	Function and Terrain	Working Grade	Loaded	Empty (Ref)
Crane, Tracked	Transport in Wet Soil	30%	.89	
Crane, Wheeled	Transport in Wet Soil	30%	.49	
Crawler Dozer	Dozing, Wet Soil	10%	1.60	
Crawler Loader	Dig and Load, Loose Soil	10%	1.12	1.30
Excavator, Tracked	Transport in Wet Soil	40%	.97	
Farm Tractor, 2WD	Plow in Loose Dirt	15%	.82	
Farm Tractor, 4WD	Plow in Loose Dirt	15%	1.02	
Garbage Packer	Crane, Wheeled	15%	.27	
Grader	Grading Wet Soil	15%	.65	
Harvesting Machine	High Speed, Grassy Field	15%	.23	
Harvesting Machine	Low Speed, Mud	15%	.35	
Harvesting Machine	Climb Obstacle		.45	
Commercial Lawn Mower	Mow on Grassy Field	30%	.37	
Lift Truck, Cushion Tire	Stop and Go, Pavement	5%	.22	
Lift Truck, Pneumatic Tire	Stop and Go, Gravel	5%	.32	
Lift Truck, Rough Terrain	Stop and Go, Loose Soil	25%	.52	
Locomotive, Switcher	Shuttle Rail Cars	3%	.19	
Log Feller, Dual Path Steer	Accelerate With Load, Wet Soil	10 %	.65	
Log Forwarder, Wheeled	Transport in Wet Soil	30 %	.49	
Mining Scoop, Wheeled	Scoop in Gravel, Rock	10 %	.72	
Paver	Paving on Firm Soil	10 %	.45	
Road Planer	Plane Highway	10 %	.52	
Roller	Roll Packed Soil	10 %	.30	
Skid Steer Loader	Dig and Load, Loose Soil	10 %	1.02	1.25
Snow Groomer	Grooming Snow on Steep Slope	60 %	1.07	
Soil Stabilizer	Stabilize Wet Soil	15%	.65	
Street Sweeper	Dump Load in Loose Soil	10%	.22	
Trash Compactor	Blading Uphill	30 %	.94	
Wheel Loader, Articulated	Dig and Load, Loose Soil	0 %	.62	.80

Table 1-3

Pull ratio and tractive effort requirements are based on typical vehicles being operated in normal fashion. Specific requirements may vary. Vehicle testing is recommended to verify that performance is satisfactory and that sufficient life of the driveline components will be obtained.

1.3 Acceleration

Often ignored during a vehicle transmission sizing proposal are vehicle acceleration and deceleration times. This data is important to know especially for high inertia vehicles. An acceptable tractive force for steady state running may be inadequate for acceptable acceleration. Tractive force minus rolling resistance is the force left to accelerate on level terrain.

A simple formula for calculating average **acceleration** or **deceleration time** on level terrain is:

$$T = \frac{W \cdot V}{G \cdot F}$$

where

T = seconds
W = Vehicle weight (lb)
V = Vehicle velocity (ft/s)
= (MPH) (1.467)
G = Gravity (32.2 ft/s^2)
F = Drawbar pull (lb)
= Tractive force minus rolling resistance

Available tractive force will change with vehicle speed due to engine power and/or pump and motor displacement and power train ratio. Calculating acceleration time requires a summation of forces as they change with vehicle speed. For example, air resistance may be a factor at high vehicle speeds.

Rolling resistance will have an affect on any vehicle's ability to accelerate as well as the ability to transmit all available force to the wheel before wheel slip.

Deceleration time is calculated by this same method, if only engine dynamic braking is used. Tractive force will vary with pump displacement and the capability of the engine to absorb torque.

Large centrifugal type loads or long conveyor belt drive may also have acceleration time requirements and should not be overlooked during the equipment selection stage.

An example is attached using computer generated (C33) performance data...

Acceleration Time

Gear #1 Draw Bar Pull (lb)	Gear #1 Speed (MPH)	Time to Accelerate (sec)	Cumulative Time (sec)	Acceleration (ft/sec^2)
3141	1.51	2.57	2.57	3.04
3471	6.83	2.62	5.19	3.13
3335	12.43	3.67	8.86	2.61
2344	18.97	4.77	13.63	1.86
1701	25.02	4.12	17.75	1.42
1386	29.01	0.59	18.34	1.02
823	29.42	0.66	19.01	0.58
427	29.68	0.63	19.64	0.30
229	29.81	0.30	19.93	0.20

Table 1-4

```
              A TYPICAL MACHINE PERFORMANCE

DATA BELOW CONSTITUTES VALID PREDICTIONS OF TRANSMISSION PERFORMANCE WITH THE
GIVEN PARAMETERS. TRANSMITTAL OF THIS DATA DOES NOT CONSTITUTE SAUER-SUNDSTRAND
APPROVAL OF THE PUMP AND MOTOR SIZE SELECTION OR THE APPLICATION PARAMETERS.

                          HS AXLE
  INPUT   INPUT   LIMITING   CHARGE     CHARGE PUMP  TRACTIVE FORCE
  RPM     POWER   PRESSURE   PRESSURE   POWER LOSS   LIMIT
  2800.   200.00  5500.0     300.0      3.00         10000000.0

GEAR MESH          FINAL DRIVE    PUMP / INPUT    MOTOR/OUTPUT
NO. OF TEETH
GEAR SPEED RATIO       6.5       1.000        1.000
GEAR EFFICIENCY       0.950       1.000        1.000


   SERIES 90 LOSSES
      PUMP PERFORMANCE FOR A  9.73 CUBIC IN. UNIT (MAX. ANGLE= 17.00 DEG.)
SWASH  ANGLE            8.60     3.00     6.00     9.00     12.00    15.00    17.00    17.00    17.00    17.00
DISPLACEMENT            4.82     1.67     3.34     5.04     6.76     8.53     9.73     9.73     9.73     9.73
RPM OF UNIT             2800     2800     2800     2800     2800     2800     2800     2800     2800     2800
ACTUAL  TORQUE          4434     1650     3129     4434     4434     4434     4434     3312     2527     2135
UNIT HP LOSS            43.88    46.77    44.86    39.76    23.67    18.27    16.81    13.06    11.13    10.36
ACTUAL  FLOW-GPM        47.73    8.27     29.35    51.26    75.67    98.64    113.88   114.91   115.58   115.90
VOLUMETRIC EFF.-%       81.76    40.90    72.40    83.89    92.29    95.43    96.56    97.43    98.00    98.27
TORQUE  EFF.-%          95.07  88.50      93.56    95.15    95.34    95.07    94.72    93.53    91.92    90.65
UNIT EFFICIENCY -%      77.73    36.20    67.74    79.82    87.99    90.73    91.47    91.13    90.09    89.08

SYS. DELTA PR.          5500     5500     5500     5259     3927     3106     2712     2000     1500     1250

MOTOR PERF. FOR 2 MOTOR(S) WITH A  6.10 IN3 DISP. EACH (MAX. ANGLE=17.0 DEG.)
   SERIES 90 LOSSES
SWASH  ANGLE            17.00    17.00    17.00    17.00    17.00    17.00    17.00    17.00    17.00    17.00
DISP./MOTOR             6.10     6.10     6.10     6.10     6.10     6.10     6.10     6.10     6.10     6.10
RPM OF UNIT             835      110      498      906      1382     1822     2113     2143     2162     2171
TORQUE/MOTOR            5118     4665     5066     4901     3697     2916     2533     1850     1369     1128
TOT. MOTOR HP LOSS      17.45    10.28    14.16    16.42    11.23    10.11    10.38    8.31     7.23     6.79
VOLUMETRIC EFF.-%       92.45    70.13    89.55    93.31    96.44    97.56    97.97    98.47    98.78    98.93
TORQUE  EFF.-%          95.85    87.37    94.87    95.98    96.97    96.70    96.19    95.26    94.00    92.97
UNIT EFFICIENCY -%      88.61    61.27    84.96    89.56    93.52    94.34    94.24    93.80    92.85    91.97

TRANS.  OUTPUT  RPM     835.4    109.8    497.7    905.5    1381.8   1822.2   2112.6   2142.5   2161.9   21711
TRANS.  OUTPUT  TORQUE  10236    9331     10131    9802     7394     5832     5066     3699     2738     2256
TRANS. INPUT HP         200.00   76.30    142.04   200.00   200.00   200.00   200.00   150.12   115.27   97.87
TRANS.  OUTPUT  HP      135.68   16.26    80.01    140.83   162.11   168.62   169.81   125.76   93.91    77.73
TOTAL TRANS. HP LOSS    64.33    60.04    62.03    59.18    37.90    31.38    30.19    24.36    21.36    20.14
OVERALL EFFICIENCY      67.84    21.30    56.33    70.41    81.05    84.31    84.91    83.77    81.47    79.42
            VEHICLE PERFORMANCE

TRACTIVE  FORCE         4214     3841     4171     4035     3044     2401     2086     1523     1127     929
WHEEL  MPH              11.47    1.51     6.83     12.43    18.97    25.02    29.01    29.42    29.68    29.81
VEHICLE  MPH            11.47    1.51     6.83     12.43    18.97    25.02    29.01    29.42    29.68    29.81
DRAWBAR  PULL           3514     3141     3471     3335     2344     1701     1386     823      427      229
DRAWBAR  HP             107.5    12.6     63.3     110.6    118.6    113.5    107.2    64.6     33.8     18.2
GRADEABILITY            10.10    9.02     9.98     9.58     6.72     4.87     3.96     2.35     1.22     0.65
HS AXLE FINAL DR.RATIO = 6.500, HS FINAL DR. EFF. =  0.950, HS ROLLING RAD.=  15.00, HS % SLIP =  0.0
ROLL. RESIST COEFF      = 0.020, ROLLING RESIST    =  700.,  VEHICLE WEIGHT =  35000.
```

Table 1-5

1.4 Charge Pump Sizing

1.4.1 Introduction

The charge pump is a critical component of the hydrostatic transmission. It is the heart of the hydrostatic transmission, for without charge flow and charge pressure, the transmission will cease to function.

The primary function of the charge pump is to replenish fluid lost through leakage. In closed circuit hydrostatic systems, continual internal leakage of high pressure fluid is inherent in the design of the components used in such a system, and will generally increase as the displacements of the system's pumps and motors increase. This "make up" fluid from the charge pump is added to the low pressure side of the closed circuit to keep the lines full of fluid and avoid cavitation at the pump.

In addition to the primary function of replenishing fluid, another major function of the charge pump is to provide charge pressure to help return the pistons and keep the slippers against the swashplate.

Other functions of the charge pump include providing fluid for the servo pistons on those systems having servo-controlled transmissions. In addition, if an Electronic Displacement Control (EDC) is used, the charge pump provides flow for the operation of a pressure control pilot valve (PCP). Charge flow also provides a transfer medium for heat dissipation. If the charge pump is used for auxiliary functions, then it must also be sized to provide this flow.

Figure 1-4 illustrates the functions that the charge pump may be required to provide in a given application.

Charge Pump Functions

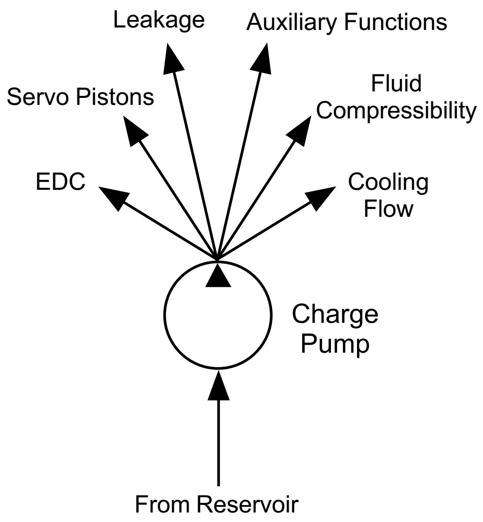


Figure 1-4

1.4.2 Charge Pump Considerations

As a rule of thumb, the charge flow requirement for a simple hydrostatic circuit is approximately 10% of the total displacement of all units in the system. However, this guideline in only an approximation for a simple system containing only high speed, piston components. The best way to size the charge pump is to consider each of the flow requirements demanded from the charge pump, many of which do not occur in a simple hydrostatic circuit.

To properly size a charge pump, several considerations must be taken into account, including the following:

- system pressure
- input speed
- minimum operational input speed
- line size and length
- control requirements
- cooling flow
- non-Sauer-Sundstrand components
- type of loading

Figure 1-5 shows how system pressure and input speed affect leakage in the system. The figure shows that leakage increases with both system pressure and input speed. Changes in pressure have a greater affect on leakage than do changes in speed. However, the affects due to speed are greater at higher pressures.

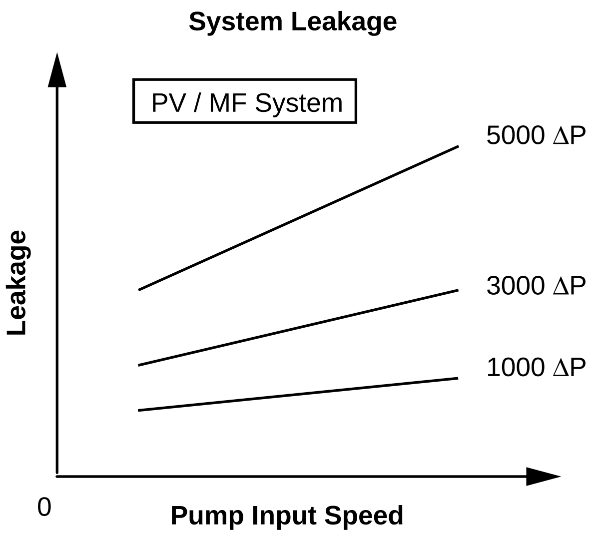


Figure 1-5

Figure 1-6 shows why it is important to also know the **minimum pump input speed**. In addition to the curves showing leakage, the figure includes curves for two charge pump sizes and their respective flows. (Charge pump #1 has the larger displacement.) The figure shows that for a given system pressure and charge pump size, system leakage varies at a rate different than that for charge flow. Disregarding for the moment all other charge pump requirements, other than leakage, the minimum charge pump size for a given speed and pressure has a flow curve which intersects the leakage curve. At low input speeds and high pressures, the potential leakage may actually exceed the flow that the charge pump is capable of providing. Furthermore, the charge pump's volumetric efficiency decreases with decreasing input speed. Therefore, even though leakage may be greater at high pump input speeds, the largest charge pump displacement may be required at a reduced input speed. Both extremes of input speed need to be checked for charge flow requirements. In many cases, the low input speed operational requirement will predominate in the final charge pump size selection.

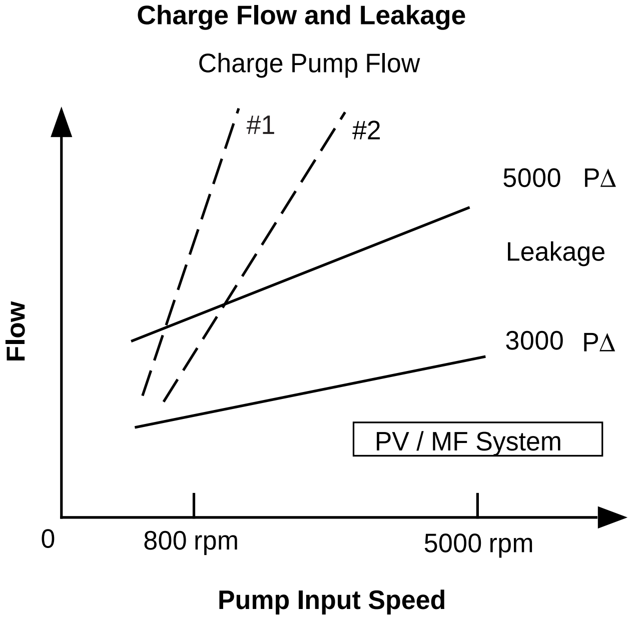


Figure 1-6

If a larger charge pump is selected due to a low input speed, then case flow at the higher speeds will be greater, and larger case drain lines may be required to keep case pressure within limits.

Make sure all components with potential leakage are considered. Any component connected to the charge flow (i.e., connected to the low pressure side of the hydrostatic loop) must have its leakage value included in the total available charge flow. In addition, if these same components also create large drops in pressure, additional charge flow may be required for cooling.

The maximum flow required for the **servo volume** for servo-controlled pumps is dependent on the stroke rate and the servo volume. Normally, the flow required is in the range of 1/2 to 2 gpm. In any case, servo flow must be included in the charge pump sizing requirement when applicable.

If an **Electrical Displacement Control (EDC)** is also used, a small amount of additional charge flow is required, usually 0.5 to 1 gpm. This flow is needed for operation of the Pressure Control Pilot Valve, which directs flow to the control spool of the displacement control. Actual leakage past the control spool is minimal, so this additional flow requirement does not apply to hydraulic or manual displacement controls.

In some applications, special considerations for **cooling flow** requirements are not necessary. Charge pump flow necessary to make up for leakage may be sufficient for cooling. More often, additional cooling flow is required and a loop flushing shuttle is specified. The charge pump size must then take into account this additional charge flow.

The type of loading can also require additional charge flow. Particularly, if the load is erratic or cyclical, a **bulk modulus** effect can occur. The name is derived from a property of the fluid called the bulk modulus, which is defined as the amount a fluid compresses for a given pressure increase. At low pressures, the amount of this fluid compression is small, and for this reason fluids are usually thought of as being “incompressible.” The pressures that can occur in hydrostatic systems, however, are of a magnitude that fluid compressibility can be significant.

The bulk modulus effect occurs when rapid system pressure spikes compress the fluid in the high pressure side of the system. This results in an instantaneous reduction of volume of the return flow in the low pressure side of the system. This reduction of return flow volume must be made by the charge pump in order to maintain proper charge pressure in the low pressure side of the system.

The degree of bulk modulus effect in a given system will depend on several factors. These are the length and size of the pressure conduits (which determine the volume of fluid subjected to the high pressure spikes), the rate of rise of the pressure spike, the magnitude of the pressure spike, and the bulk modulus of the fluid.

Because the bulk modulus effect is so easily overlooked, and because it often results in a tremendous increase in required charge flow, Section 3.5 has been included to bring special attention to this topic.

The required charge pump size is one with a displacement which is able to provide flow for all of the above requirements. If the required charge flow exceeds the capability of all available charge pump displacements, then a gear pump (or some additional charge flow source) must be used. Most Sauer-Sundstrand pumps include an auxiliary pad to mount gear pumps of various displacements.

After a charge pump size is selected, a system must always be tested to be certain charge flow and pressure requirements are met.

Figure 1-7 is a worksheet which may be use to help size a charge pump. Each of the charge flow requirements are included. The sum of the required charge flows represents the total flow required if all charge flow demands need to be met simultaneously. In reality, this is usually not be the case. For example, it may be that for a particular system, a bulk modulus effect may never occur while an auxiliary function is active. Each application needs to be reviewed carefully to determine how much charge flow is required.

1.4.3 Charge Pump Sizing Worksheet

Customer: ____________________ Date: __________

Application: ____________________

Leakage

"Pump" refers to hydrostatic pump, not charge charge. Actually, a portion of all inefficiencies can be attributed to crossport leakage between high and low system loops. Since the charge pump needs to replace only fluid leaking past the rotating kits (case flow), the calculations below are somewhat conservative. If case flow values are available, they should be used instead of the equations below.

System Pressure ________ psi

Pump
- Series ________
- Frame Size ________
- Speed ________ RPM
- Volumetric Efficiency ________ %
- Leakage ________ gpm

Motor #1
- Series ________
- Frame Size ________
- Speed ________ RPM
- Volumetric Efficiency ________ %
- Leakage ________ gpm

Motor #2
- Series ________
- Frame Size ________
- Speed ________ RPM
- Volumetric Efficiency ________ %
- Leakage ________ gpm

Total Leakage ________ gpm

$$\text{Pump Flow} = \frac{\text{Pump Disp x Pump RPM}}{231} \times \frac{\text{Pump Efficiency}}{100}$$

$$\text{Pump Leakage} = \frac{\text{Pump Disp x Pump RPM}}{231} \times 1 - \frac{\text{Pump Efficiency}}{100}$$

$$\text{Motor Speed} = \frac{\text{Pump Flow x Motor Efficiency}}{\text{Motor Disp x \# Motors}}$$

$$\text{Motor Leakage} = \frac{\text{Pump Flow}}{\text{\# Motors}} \times 1 - \frac{\text{Motor Efficiency}}{100}$$

Consult product technical information bulletins for values of volumetric efficiency.

Control Requirements

Control Type
- ____ DDC
- ____ MDC ________ gpm
- ____ HDC ________ gpm
- ____ EDC ________ gpm
- ____ Other ________ gpm

$$\text{Flow} = \frac{\text{Servo Volume x 0.26}}{\text{Stroke Time}}$$

Series	Servo Volume (in^3)
Series 40	
M46	1.5
Series 42	
28cc	1.0
41cc	1.5
Series 90	
42cc	1.0
55cc	1.3
75cc	1.7
100cc	2.5
130cc	3.5
180cc	5.0
250cc	5.0

For most applications with 1-3 second stroke times, assume a value of 0.5 gpm.

For atypical stroke times, use the chart and equation shown at right.

For pumps with EDC controls, add 0.75 gpm to the servo flow to allow for losses in the PCP.

Charge Pump Sizing Worksheet (cont.)

Loop Flushing

Loop Flushing flow ________________ gpm

The amount of loop flushing will normally vary between 2-4 gpm depending on the charge pump displacement, input speed, and relative settings between the pump and motor charge relief valves.

Fluid Compressibility

Magnitude of pressure spike ________________ psi
Time duration ________________ sec.
Bulk modulus ________________ psi
Hose length ________________ feet
Hose I.D. ________________ inches
Hose Volume ________________ in^3

Charge flow required ________________ gpm

Hose Volume = $V = (9.42) \times (\text{I.D.})^2 \times (\text{Length})$

$$Q = \frac{\Delta P \cdot (V)}{(BM) \cdot \Delta t} \cdot 0.26$$

where
Q = additional charge flow required (gpm)
ΔP = change in pressure (gpm)
BM = bulk modulus (psi)
Δt = time duration for pressure change (sec)

Auxiliary Functions

Hydraulically released brakes ________________ gpm
Two-speed motor shifting ________________ gpm
Cylinders ________________ gpm
Other components ________________ gpm

Total auxiliary flow ________________ gpm

Total Charge Flow Required

Leakage + Control + Loop Flushing + Compressibility + Auxiliary = ____________ gpm

Select a preliminary charge pump displacement:

Charge pump displacement ________________ in^3
Volumetric efficiency ________________ %
Charge flow provided ________________ gpm

$$\text{Charge flow} = \frac{(\text{Ch Disp}) \times (\text{Input Speed}) \times (\text{Ch Efficiency})}{231}$$

Is the charge pump capable of providing adequate charge flow?

If not, a larger displacement size must be selected, or an external charge supply must be provided.

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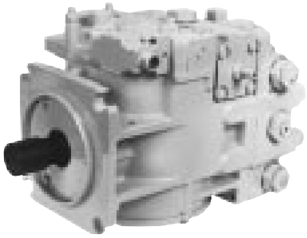
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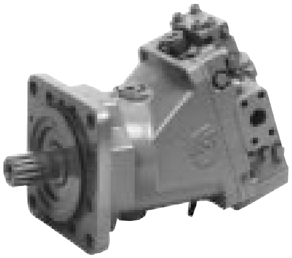
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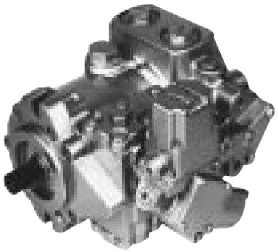
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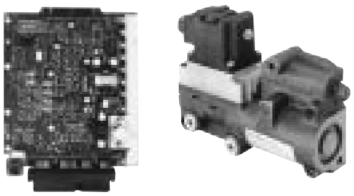
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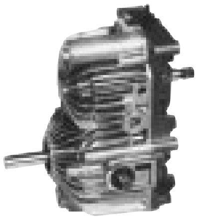
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