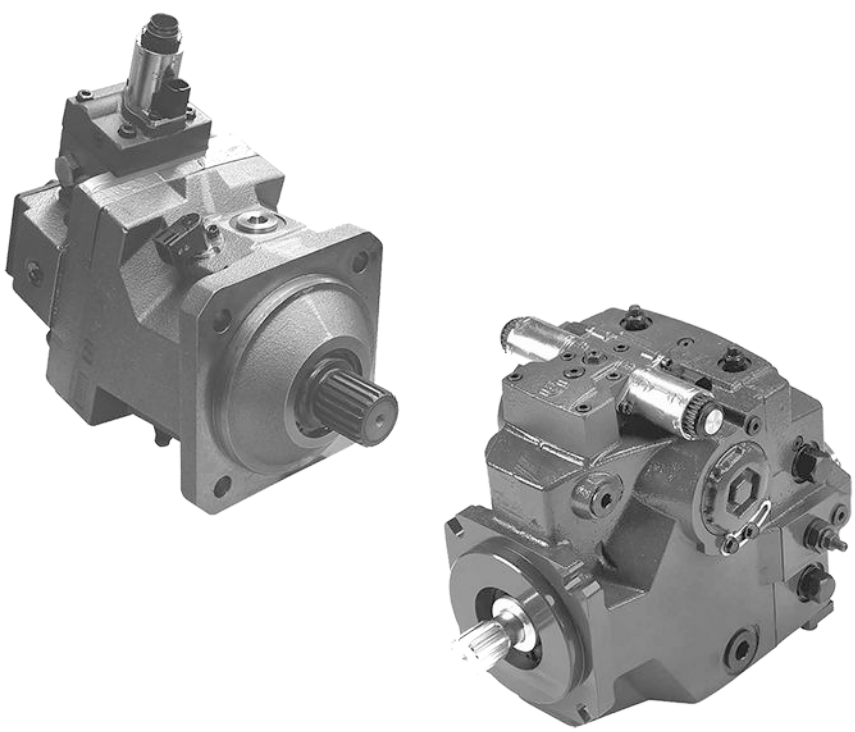
Applications Manual

Selection of Driveline Components



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Revision history

Table of revisions

Date	Changed	Rev
July 2015	Minor edits	0304
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Introduction

Applications Manuals

Content included in these manuals

These applications manuals provide design theory and detailed calculations for building hydraulically powered machines.

The original document was written as one manual with four sections.

The current set of manuals includes the four documents listed below. The section numbers from the original document are listed in parenthesis after the current document title.

- *Selection of Driveline Components* BLN-9885 (originally Section 1)
- *Pressure and Speed Limits for Hydrostatic Units* BLN-9884 (originally Section 2)
- *Transmission Circuit Recommendations* BLN-9886 (originally Section 4)
- *Fluids and Lubricants* 520L0463 (originally Section 3)

Other Reference Manuals

- *Hydraulic Fan Drive Systems Technical Information* 520L0824
- *Hydraulic Fan Drive Systems Design Guidelines* 520L0926

Selection of Driveline Components

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 have been defined, and that the fundamental design parameters have been established for each mode of operation. These typically include vehicle speed, gradability, useful life, vehicle weight, and drive configuration. It is also assumed that required engine power has been established.

Design Goal

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

Hydraulic unit life is highly dependent on system pressure. Establish maximum and continuous pressure based on the required life of the driveline. Danfoss document *Pressure and Speed Limits for Hydrostatic Units* BLN-9884 covers this subject in detail.

The figure below *Driveline Element Selection* 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 driveline design includes so many variables (each dependent on the others), and because final component selection is ultimately limited by product availability, several iterations of this procedure may be required before arriving at the optimum system.

Sizing Procedure

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 mechanisms. It must also be large enough to provide sufficient flow to the drive motor to attain the required speed.

Selection of Driveline Components

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

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 the equations presented throughout this document, a sizing flowchart is included to assist with sizing. The flowchart details the sizing procedure and includes numerous design check steps to validate the calculated sizing values.

Design limits for associated mechanical components are not identified.

Machine designers should verify that the design parameters are met for all driveline components.

The steps outlined in this manual are designed to guide you in component selection. For further assistance, contact your Danfoss representative for help interpreting and verifying your results.

Machine Corner Power (CP)

The first step in the sizing process is to determine the value referred to as Machine Corner Power (CP). The concept of Corner Power is abstract and is normally not an attainable value of transmission power. It is useful in the design process because it provides an indication of transmission component size and ratio requirements. Corner Power is representative of the maximum torque and the maximum speed (at full load) that the machine is required to have. These two values of maximum speed and maximum torque (or Tractive Effort) never happen at the same time, but the purpose of Corner Power is to capture both values to define an operating envelope for the machine and to aid in the selection of the hydraulic motor. Refer to the Machine Corner Power graph below for an illustration of the concept.

The concept of Corner Power also applies to hydraulic motors. As demonstrated in the topic *Motor Selection* on page 9, the maximum corner power of a hydraulic motor represents the maximum torque and maximum continuous speed capabilities of that product. Equations are provided in the Motor Selection topic that allow you to select the appropriate motor based on the machine's corner power.

The equations for calculating Corner Power are provided below. For rotary drives (work function), the input values to the equation are the required maximum output torque and the maximum output speed

Selection of Driveline Components

(at full load) of the machine. For propel drives, the input values are maximum tractive effort and maximum vehicle speed (at full load).

For multi-speed drives (e.g. work mode and travel mode), corner power must be calculated for all ranges.

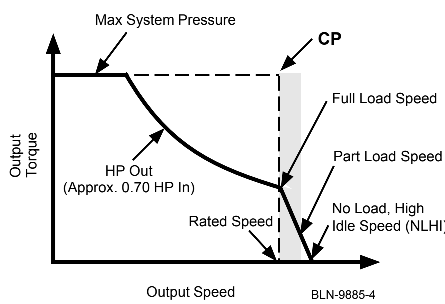
Tractive Effort

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.

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.

The topic *Tractive Effort* on page 26 describes tractive effort in more detail.

Machine Corner Power



Machine Corner Power (CP) is determined by estimating 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.

Warning

Protect yourself from injury. Use proper safety equipment, including safety glasses, at all times.

Warning

Check to ensure that maximum motor speed is NOT exceeded under dynamic braking conditions, when engine speed can exceed No Load High Idle (NLHI) ratings.

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

Selection of Driveline Components

Variable or Fixed Motor

Because the machine corner power is an expression of maximum torque (tractive effort) and maximum vehicle speed, it can be used to establish the effective Transmission Ratio (TR) required to satisfy system demands.

The **effective Transmission Ratio (TR)** is the ratio of the required vehicle corner power divided by the **available power** from the machine's prime mover (engine). 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.

For drives with variable load cycles, determine the normal input power (available power) to the transmission by deducting the average power dedicated to other functions from the maximum engine power available to the drive.

A Transmission Ratio (TR) greater than 1.0 means that there is not enough engine power available to meet all of the operating requirements at the same time.

- Typically, machines with high transmission ratios have high torque (Tractive Effort) requirements at low speed and high speed requirements at low torque (Tractive Effort). In this case, a large fixed motor would satisfy the high torque requirements, but operating the same motor to meet the maximum speed requirement could exceed the speed limit of the motor and require a large displacement pump. For high transmission ratios, use a variable displacement motor; it can be used at high displacement to satisfy the maximum torque requirement and then shifted to a smaller displacement to satisfy the machine's maximum speed requirement. A fixed motor could be used with a multi-ratio gearbox for machines with a high transmission ratio, but usually a variable motor will be the most cost effective solution.
- If the transmission ratio is low, that means that there is probably enough engine power available to achieve the maximum torque and speed requirements simultaneously. In those cases, a fixed motor is suitable for the task.
- In cases of extremely high transmission ratio, a variable motor may not satisfy the need. In these cases, a multi-speed gearbox may also be required. Some applications use 2-speed, 3-speed, or 4-speed gearboxes to meet the vehicle requirements; but a 2-speed gearbox is most common.

The rule for selecting a fixed or 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,
- If TR is greater than 14, use a multi-ratio gear box between the motor and the final drive.

There is no direct relationship between transmission ratio and final drive ratio. The final drive ratio is calculated based on the displacement of the motor that has been chosen, the maximum pressure, the loaded radius of the wheels, and the required maximum tractive effort.

The transmission ratio is only used to help determine the motor type, not the motor size. Refer to the topic *Final Drive Selection* on page 11 to calculate the Final Drive Ratio (FD).]

Selection of Driveline Components

SI / US System	Description
2) $TR = \dfrac{\text{Machine CP}}{HP}$	TR = effective transmission ratio HP = normal input power kW (hp) HP = 0.7 * Available prime mover power
TR < 2, use fixed displacement motor TR > 4, use variable displacement motor TR > 14, use multi-ratio gearbox	

Motor Selection

Calculate the required motor corner power from machine corner power and driveline efficiency using equation (3) Required Motor CP. This establishes the minimum motor size capable of meeting the power requirement 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) Maximum Motor CP to calculate the maximum motor corner power based on the design maximum pressure and the design maximum speed and 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 will occur for only a small percentage of the operating time, usually less than 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 *Machine Corner Power* graph).

Danfoss document *Pressure and Speed Limits for Hydrostatic Units* BLN-9884, 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) Maximum Motor CP 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. The table *Hydrostatic Motor Corner Power Chart* can be used as an aid in preliminary motor selection. You should choose a motor with a motor CP at least as large as the required motor CP calculated using Equation (3) Required Motor CP.

Equation (A) Design Check 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.

Selection of Driveline Components

SI System	US System	Description	
3) $\text{Required Motor CP} = \dfrac{\text{Machine CP}}{E \cdot \#}$	$\text{Required Motor CP} = \dfrac{\text{Machine CP}}{E \cdot \#}$	CP = corner power E = final drive efficiency # = number of motors	kW [hp] (%/100)
4) $\text{Maximum Motor CP} = \dfrac{DM \cdot NM \cdot PM}{600\,000}$	$\text{Maximum Motor CP} = \dfrac{DM \cdot NM \cdot PM}{396\,000}$	DM = maximum motor displacement NM = design maximum speed PM = design maximum pressure	cc [in^3]/rev rpm bar [psi]
A) Design Check: Maximum Motor CP ≥ Required Motor CP			

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 the 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 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 transmission ratio and maximum motor speed, in addition to meeting the torque requirements.

Selection of Driveline Components

Hydrostatic Motor Corner Power Chart

Motor	Max Pressure (psid)	Max Working Pressure (psid)	Max Speed at Max Angle (rpm)	Cont Speed at Max Angle (rpm)	Max Speed at Min Angle (rpm)	Cont Speed at Min Angle (rpm)	Fixed Corner Power (HP)	Fixed Corner Power (kW)	Variable Corner Power (HP)	Variable Corner Power (kW)
Series 15	4500	4350	----	4000	----	----	42	31	----	----
Series 40 - M25	5000	4350	----	4000	----	----	77	57	----	----
Series 40 - M35	5000	4350	----	3600	5300	4200	97	72	113	84
Series 40 - M44	5000	4350	----	3300	4850	3900	112	83	132	99
Series 40 - M46	5000	4350	----	3600	5000	4500	128	95	160	119
LV/LC25	6000		----	----	5000	4400	----	----	102	76
LV/LC30	5000		----	----	5150	4450	----	----	103	77
LV/LC35	4350		----	----	5300	4500	----	----	106	79
KV/KC38	6000		----	----	5200	4650	----	----	163	122
KV/KC45	5000		----	----	5050	4500	----	----	156	116
Series 90 - 55cc	7000		4250	3900	5100	4600	231	173	273	204
Series 90 - 75cc	7000		3950	3600	4700	4250	291	217	344	256
Series 90 - 100cc	7000		3650	3300			356	266	----	----
Series 90 - 130cc	7000		3400	3100			435	324	----	----
H1B060	7000	6525		----	7250	5900	----	----	382	285
H1B080	7000	6525		----	6600	5300	----	----	457	341
H1B110	7000	6525		----	5950	4800	----	----	570	425
H1B160	7000	6525		----	5250	4250	----	----	734	547
H1B250	7000	6525		----	4500	3650	----	----	985	734
51V060	7000				7000	5600	----	----	363	270
51V080	7000				6250	5000	----	----	432	322
51V110	7000				5600	4500	----	----	534	398
51V160	7000				5000	4000	----	----	691	515
51V250	7000				4250	3400	----	----	917	684

These values for corner power capability are based on maximum pressure and maximum speed ratings.

Refer to *Pressure and Speed Limits for Hydrostatic Units* BLN-9884 for detailed information on ratings of units and expected life.

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 *Pressure and Speed Limits for Hydrostatic Units* BLN-9884).

The two methods are as follows:

Selection of Driveline Components

1. Using the *Sizing Flow Chart* on page 19, size the final drive ratio using the design maximum pressure and the maximum torque requirement. Use equation (5) Required FD on the following page for this calculation. After the pump is sized and all speed conditions have been met, estimate the continuous pressure, using the *Sizing Flow Chart* on page 19, and compare it with the maximum design continuous pressure.
2. As an alternate method, calculate the final drive ratio required for all modes of operation (travel mode, work mode, etc.). Calculate the final drive ratio 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 the 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 ratio from the values calculated for the various operating modes.

For variable or two-position motors, only final drive ratios from those modes utilizing maximum motor displacement can be calculated, since the motor minimum displacement is not yet known.

The next step is to check motor speed limits using the limits obtained from *Pressure and Speed Limits for Hydrostatic Units* BLN-9884, or the respective Technical Information manual.

Motor speed will usually be satisfactory unless the final drive ratio is significantly higher than required (Gearbox limits must also be met). Equation (6) NMR=FD•NMD is used to determine the required motor speed at maximum motor displacement based on the final drive ratio calculated in equation (5) Required FD. 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 swashplate angle. Use design check (C) NMR ≤ NML to ensure that the speed limit of the motor is not exceeded. If a variable motor is specified, use equation (7) NVR=FD•NMD and design check (D) NVR ≤ NVL to determine if the speed required at the minimum motor displacement exceeds the maximum reduced angle speed limit. As explained in *Pressure and Speed Limits for Hydrostatic Units* BLN-9884, the maximum speed limit of a variable motor increases with decreasing angle, up to a certain value (the maximum reduced angle speed limit or cutoff point on the speed/angle curve). At low swashplate angles (i.e., below the angle cutoff point), a decrease in angle does not result in a greater maximum speed limit.

Note that 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 with the smallest possible swashplate angle (i.e., at the cutoff point of the speed/angle curve), then increase the motor's maximum displacement and recalculate the final drive ratio.

Refer to *Pressure and Speed Limits for Hydrostatic Units* BLN-9884 for more information concerning speed limits.

Both SM (vehicle speed required at max angle) and SV (vehicle speed required at min angle) are customer defined conditions

Selection of Driveline Components

SI System	US System	Description			
Rotary Drives		DM	=	max motor displacement	cc (in³)/rev
5) Required FD = $\dfrac{Torque \cdot 20\pi}{DM \cdot PM \cdot E \cdot EM}$	Required FD = $\dfrac{Torque \cdot 2\pi}{DM \cdot PM \cdot E \cdot EM}$	E	=	final drive efficiency	(%)/100
		FD	=	final drive ratio	d'less
		E M	=	motor mechanical efficiency	(%)/100
		LR	=	wheel loaded radius	mm (in)
Propel Drives		NMD	=	non-propel design speed at max angle	rpm
Required FD = $\dfrac{TE \cdot LR \cdot 20\pi}{DM \cdot PM \cdot E \cdot EM \cdot \#}$	Required FD = $\dfrac{Torque \cdot 2\pi}{DM \cdot PM \cdot E \cdot EM \cdot \#}$	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
B) Design Check: FD ≥ Required FD		NVR	=	req'd motor speed at min angle	rpm
Rotary Drives		NVL	=	motor speed limit at min angle	rpm
6) NMR = FD • NMD	NMR = FD • NMD	PM	=	maximum pressure	bar (psid)
		SM	=	vehicle speed req'd at max angle	kph (mph)
Propel Drives		SV	=	vehicle speed req'd at min angle	kph (mph)
NMR = $\dfrac{FD \cdot SM \cdot 2650}{LR}$	NMR = $\dfrac{FD \cdot SM \cdot 168}{LR}$	TE	=	vehicle tractive effort	N (lbf)
		TQ	=	max drive output torque	Nm (in•lbf)
		#	=	number of motors	
C) Design Check: NMR ≤ NML					
Rotary Drives					
7) NVR = FD • NVD	NVR = FD • NVD				
Propel Drives					
NVR = $\dfrac{FD \cdot SV \cdot 2650}{LR}$	NVR = $\dfrac{FD \cdot SV \cdot 168}{LR}$				
D) Design Check: NVR ≤ NVL					

Input Gearing

The use of input gearing is usually customer defined and determined by the machine configuration. For vehicles with multiple hydraulic systems, use of an input splitter box is common. Splitter boxes 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) NP = NE•IR to determine the relationship between the prime mover speed, pump speed, and input gear ratio.

Selection of Driveline Components

	SI / US System	Description			
8)	$NP = NE \cdot IR$	NP	=	maximum pump design speed	rpm
		NE	=	prime mover design speed	rpm
		IR	=	pump input ratio	

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 calculated displacement. Also, check that the desired pump speed does not exceed the rated maximum speed for the pump. If the rated 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 that is selected will usually have a displacement slightly greater than the calculated displacement.

Fixed Motor

For a fixed motor, determine the actual motor speed and compare with its rated maximum speed 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 at the pump (and consequently the motors), typically by as much as 15%. The condition is especially common during downhill operation. Not only is there an increase in pump speed, but during either downhill operation or vehicle deceleration using hydrostatic braking; the motor becomes the pump and the pump becomes the motor. The net result is that the motor will turn faster for any given pump speed than what would be experienced during normal propel operation.

A 15% increase in engine speed is just an estimate; check with the engine manufacturer for specific details concerning the engine's ability to provide dynamic braking and its maximum, or [not-to-exceed] operating speed.

Selection of Driveline Components

SI / US System

9) $$DPR = \frac{NMR \cdot DM \cdot \#}{NP \cdot EVP \cdot EVM}$$

D) Design Check: $DP \geq DPR$

E) Design Check: $NP \leq NPL$

10) $$NPR = \frac{DM \cdot NMR \cdot \#}{DP \cdot EVP \cdot EVM}$$

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

Without Overrunning Condition:

12) $$NM = \frac{DP \cdot NE \cdot IR \cdot EVP \cdot EVM}{DM \cdot \#}$$

With Overrunning Condition:

$$NM = \frac{DP \cdot NE \cdot IR \cdot 1.15}{EVP \cdot EVM \cdot DM \cdot \#}$$

G) Design Check: $NM \leq NML$

Description

DM	=	max motor displacement	cc (in^3)/rev
DP	=	max pump displacement	cc (in^3)/rev
DPR	=	required max pump displacement	cc (in^3)/rev
IR	=	pump input ratio	dimensionless
NMR	=	required 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	=	required pump speed	rpm
EVP	=	pump volumetric efficiency	%/100
EVM	=	motor volumetric efficiency	%/100
#	=	number of motors	dimensionless

Assume EVP = EVM = 95% for first pass thru the sizing exercise

Variable Motor

For a variable motor, the procedure for assuring that the rated maximum speed is not exceeded is somewhat more involved.

The steps are as follows:

1. Determine if the rated maximum speed is exceeded using the method above, reference equation (12).
2. Determine the minimum motor displacement using equation (13).
3. Calculate the angle associated with this displacement using equation (14). Select an available minimum swashplate angle using design check (H) and determine the actual motor speed using equation (15).
4. Determine the reduced swashplate angle speed from *Pressure and Speed Limits for Hydrostatic Units* BLN-9884, or by using equation (16). Use design check (I) to ensure that the minimum angle speed limit is not exceeded.

The Sizing Flowchart below details the above procedure.

Selection of Driveline Components

SI / US System

13) $$DV = \frac{DP \cdot NE \cdot IR \cdot EVP \cdot EVM}{NVR \cdot \#}$$

14) All Swashplate Motors:

$$TANV = TANM \cdot (DV / DM)$$
$$AV = ARCTAN\ (TANV)$$

All H1B andSeries 51 Bent-Axis Motors:

$$SINV = 0.53 \cdot (DV / DM)$$
$$AV = ARCSIN\ (SINV)$$

Note: Sin 32° = 0.53, 32° = Maximum angle of H1B and S51 bent axis motors

H) Design Check: $AV \geq$ Min Available

IF "YES", then proceed; IF "NO", increase pump displacement in Equation 13

15) Without Overrunning Condition:

$$NV = \frac{DP \cdot NE \cdot IR \cdot EVP \cdot EVM}{DV \cdot \#}$$

With Overrunning Condition:

$$NV = \frac{DP \cdot NE \cdot IR \cdot (1.15)}{EVP \cdot EVM \cdot DV \cdot \#}$$

NOTE: 1.15 is an estimate of engine max speed capability, contact engine supplier for additional information

16) All Swashplate Motors:

$$NVL = NML \cdot \sqrt{DM / DV}$$

Series 51 Bent-Axis Motors:

$$NVL = NML \cdot (0.53 / SINV)$$

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

IF "YES", then proceed

	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 displacemnt	cc (in^3)/rev
DV	= min motor displacement	cc (in^3)/rev
IR	= pump input ratio	dimensionless
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	dimensionless
SINV	= sine of motor at min angle	dimensionless
SV	= vehicle speed req'd at min angle	kph (mph)
TANM	= tangent of motor at max angle	dimensionless
TANV	= tangent of motor at min angle	dimensionless
EVP	= pump volumetric efficiency	%/100
EVM	= motor volumetric efficiency	%/100
#	= number of motors	dimensionless

Assume EVP = EVM = 95% for first pass thru the sizing exercise

Continuous Pressure

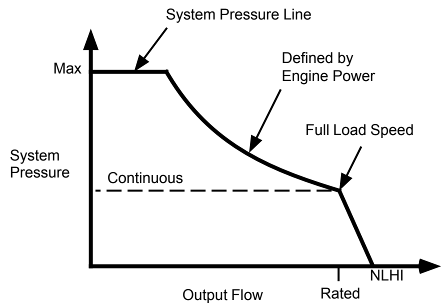
The final (but crucial) step in the procedure is to estimate the continuous pressure based on the components selected.

The Pump Selection graph below 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 *Sizing Flow Chart* on page 19 equations provide a check to ensure that the continuous pressure is below the pressure required to meet the design life.

Selection of Driveline Components

Pump Selection



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) $PC = \frac{HP \cdot 600\,000}{DP \cdot NE \cdot IR}$	$PC = \frac{HP \cdot 396\,000}{DP \cdot NE \cdot IR}$
Propel, Motor at Max Angle	
18) $FD = \frac{NML \cdot LR}{2650 \cdot SM}$	$FD = \frac{NML \cdot LR}{168 \cdot SM}$
Non-Propel, Motor at Max Angle	
$FD = \frac{NML}{NMD}$	$FD = \frac{NML}{NMD}$
Propel, Motor at Min Angle	
$FD = \frac{NML \cdot LR}{2650 \cdot SV}$	$FD = \frac{NML \cdot LR}{168 \cdot SV}$
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	mm (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)

System Sizing Flow Chart

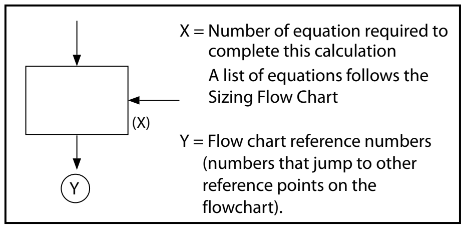
The flowchart is designed to be used as a sizing algorithm to assist in the selection of system components. It provides a concise step-by-step run-through of the sizing process. It is intended to accompany the previous sections and to expand the equations presented with the text.

The symbols used in the flowchart are explained in the box below. The equations used to calculate the quantities are included following the flowchart, along with the 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.

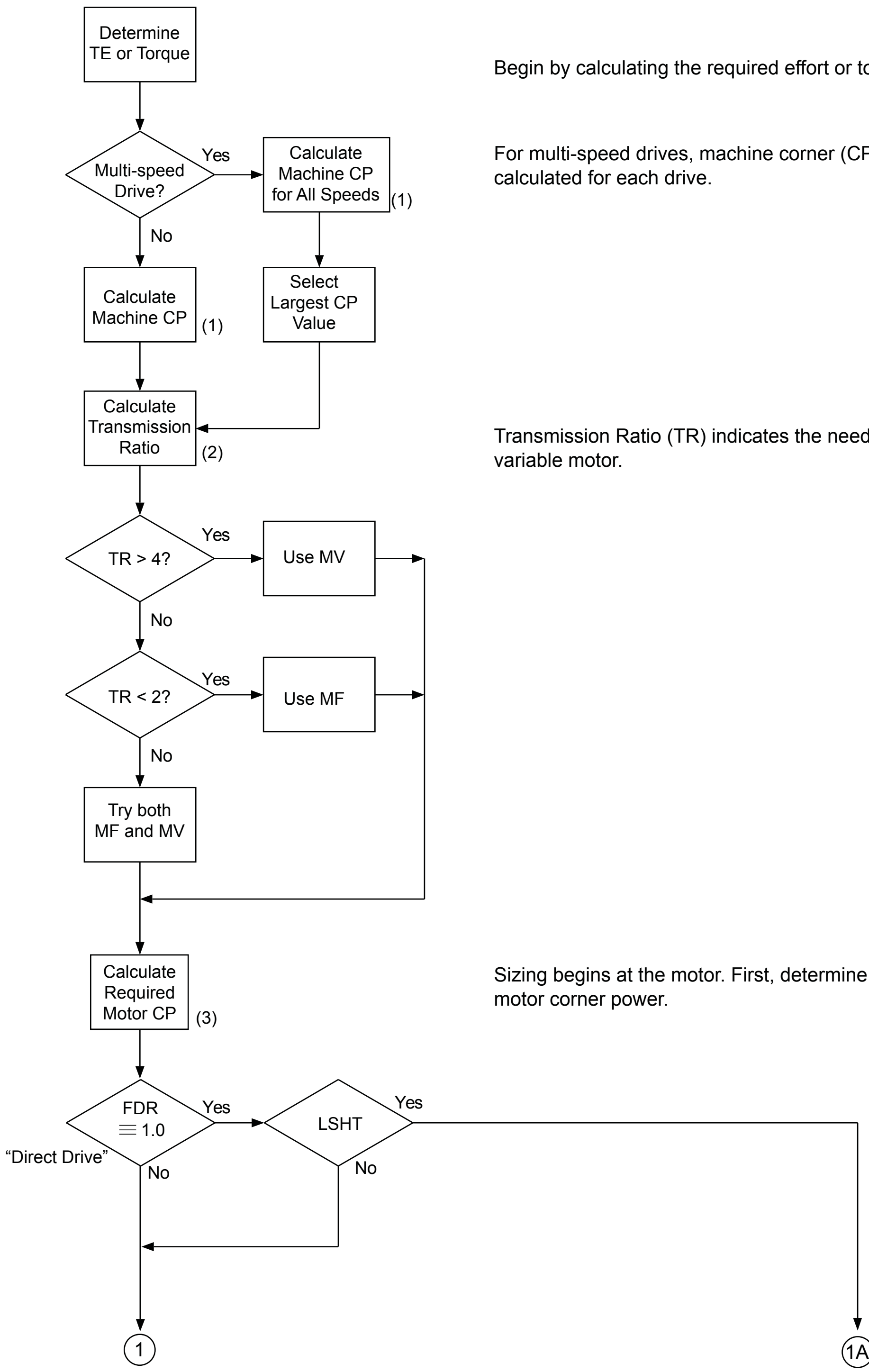
Use the flowchart on the following pages to assist in sizing a hydrostatic transmission. The number of the equation required to complete this calculation is shown at the lower right corner of the boxes (X). This equation can be found in the tables following the flowchart.

Selection of Driveline Components



Selection of Driveline Components

Sizing Flow Chart



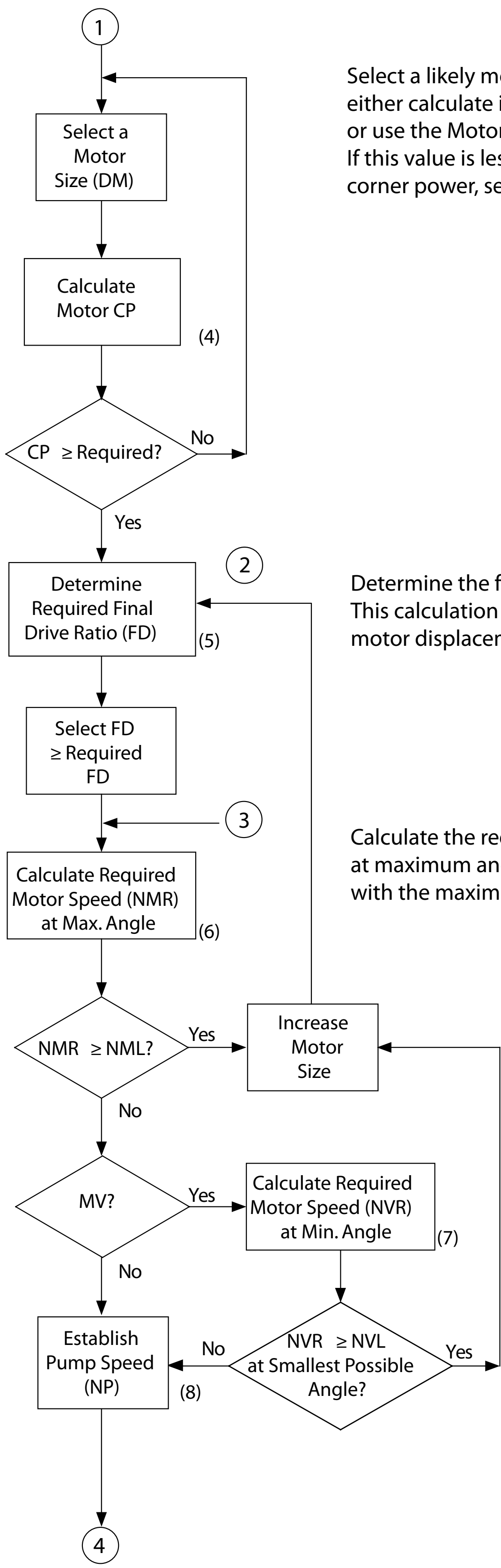
Begin by calculating the required effort or torque.

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.

Selection of Driveline Components

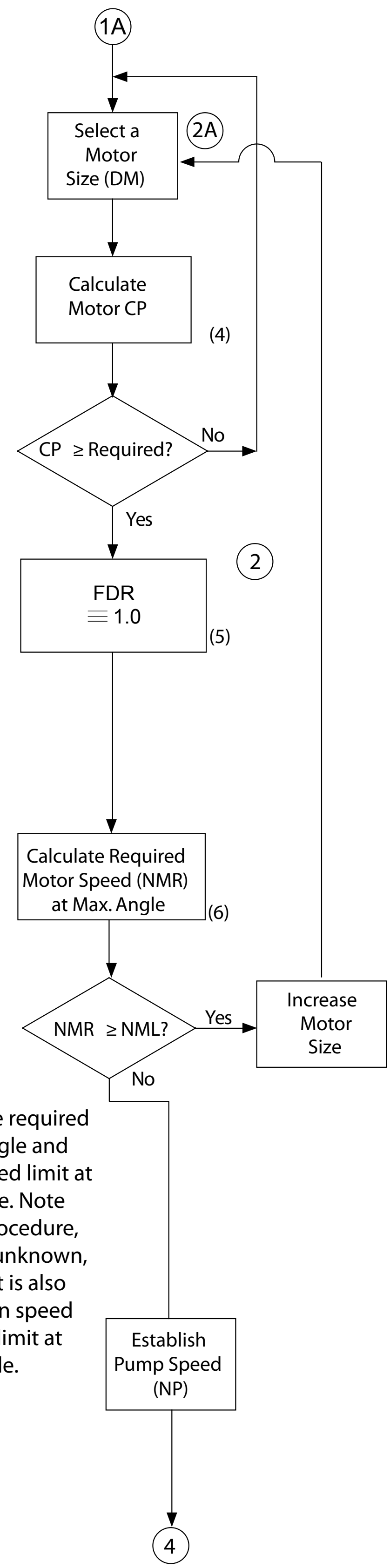


Select a likely motor displacement and either calculate its corner power capability or use the Motor Power Capabilities table. If this value is less than the required corner power, select a larger motor.

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.

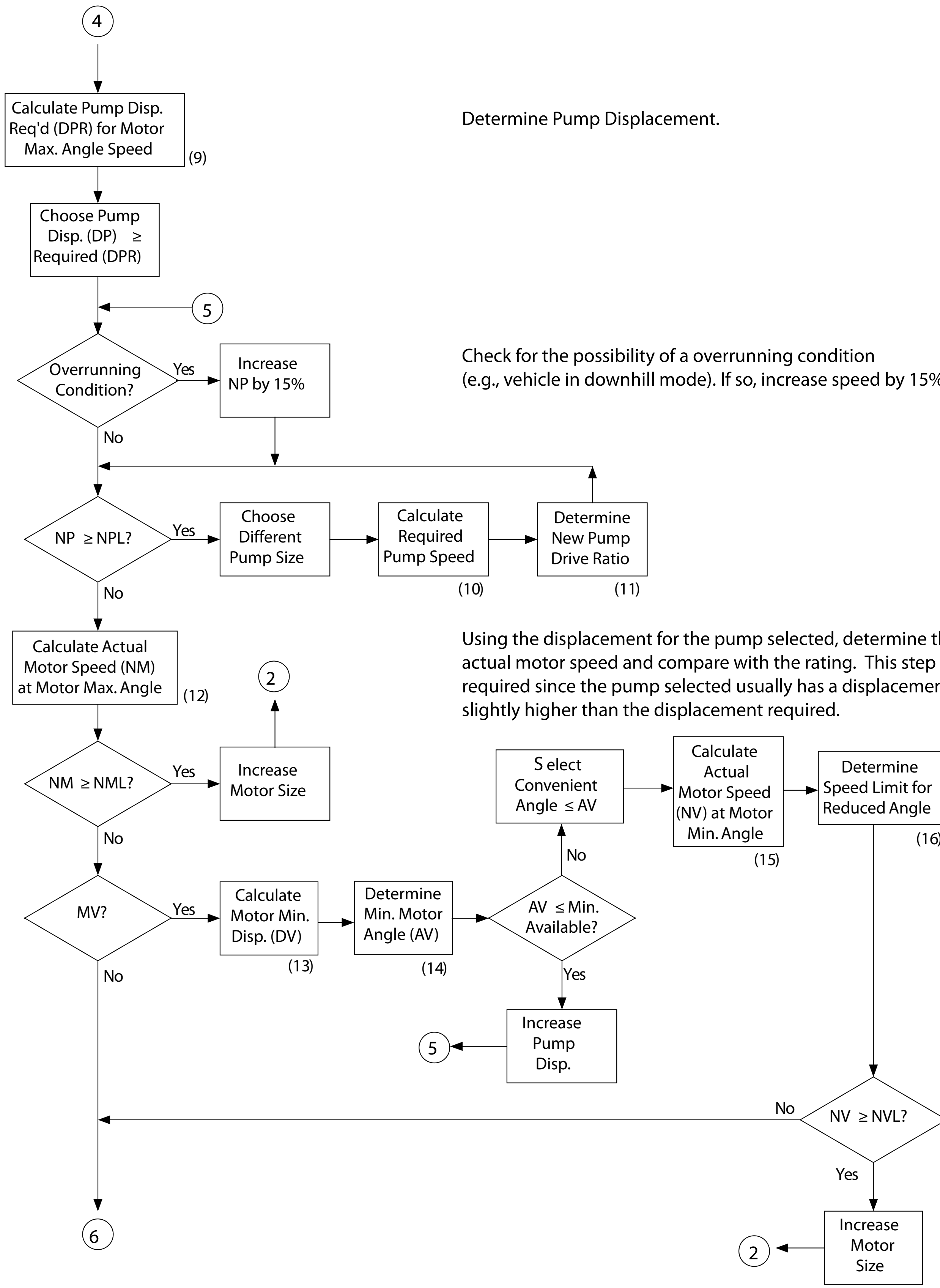


Selection of Driveline Components

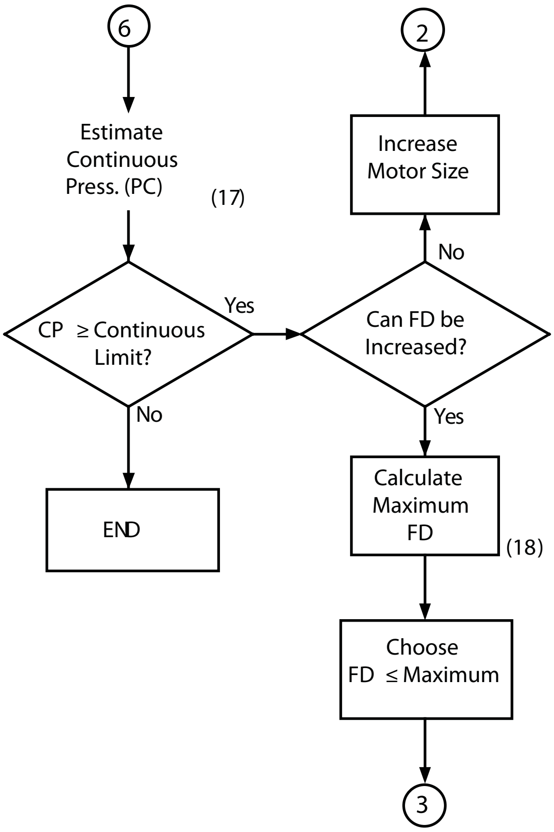
Determine Pump Displacement.

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.



Selection of Driveline Components



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.

Selection of Driveline Components

Equations

Step	Equations Required		Comments
	Metric System	US System	
1	Machine CP = $\frac{TE \cdot S}{3600}$	Machine CP = $\frac{TE \cdot S}{375}$	Propel Drive
	Machine CP = $\frac{TQ \cdot ND}{9549}$	Machine CP = $\frac{TQ \cdot ND}{63025}$	Non-Propel Drive
2	TR = $\frac{\text{Machine CP}}{0.7 \cdot \text{Available power}}$	Same	
3	Required Motor CP = $\frac{\text{Machine CP}}{E \cdot \#}$	Same	
4	Motor CP = $\frac{0.95 \cdot DM \cdot NM \cdot PM}{600\,000}$	Motor CP = $\frac{0.95 \cdot DM \cdot NM \cdot PM}{396\,000}$	
5	Required FD = $\frac{TE \cdot LR \cdot 0.02\pi}{0.95 \cdot DM \cdot PM \cdot E \cdot \#}$	Required FD = $\frac{TE \cdot LR \cdot 2\pi}{0.95 \cdot DM \cdot PM \cdot E \cdot \#}$	Propel Drive
	Required FD = $\frac{TQ \cdot 20\pi}{0.95 \cdot DM \cdot PM \cdot E}$	Required FD = $\frac{TQ \cdot 2\pi}{0.95 \cdot DM \cdot PM \cdot E}$	Non-Propel Drive
6	NMR = $\frac{FD \cdot SM \cdot 2650}{LR}$	NMR = $\frac{FD \cdot S \cdot 168}{LR}$	Propel Drive
	NMR = $FD \cdot NDM$	Same	Non-Propel Drive
7	NVR = $\frac{FD \cdot SV \cdot 2650}{LR}$	NVR = $\frac{FD \cdot SV \cdot 168}{LR}$	Propel Drive
	NVR = $FD \cdot NDV$	Same	Non-Propel Drive
8	NP = $NE \cdot IR$	Same	
9	DPR = $\frac{NMR \cdot DM \cdot \#}{(0.95)^2 \cdot NP}$	Same	
10	NPR = $\frac{DM \cdot NMR \cdot \#}{DP \cdot (0.95)^2}$ (choose DP $\geq$ DPR)	Same	
11	IR = $\frac{NPR}{NE}$	Same	

Selection of Driveline Components

Step	Equations Required — Metric System	Equations Required — US System	Comments
12	$NM = \frac{DP \cdot NE \cdot IR \cdot (0.95)^2}{DM \cdot \#}$	Same	Normal Operation
	$NM = \frac{DP \cdot NE \cdot IR \cdot 1.15}{(0.95)^2 \cdot DM \cdot \#}$	Same	Overrunning Conditions
13	$DV = \frac{DP \cdot NE \cdot IR \cdot (0.95)^2}{NVR \cdot \#}$	Same	
14	$TANV = TANM \cdot (DV / DM)$ $AV = \text{Arctan}(TANV)$	Same	All Swashplate Motors
	AV = Refer to Technical Information manual	Same	H1B & Series 51 Bent Axis Motors
15	$NV = \frac{DP \cdot NE \cdot IR \cdot (0.95)^2}{DV \cdot \#}$	Same	Normal Operation
	$NV = \frac{DP \cdot NE \cdot IR \cdot 1.15}{(0.95)^2 \cdot DV \cdot \#}$	Same	Overrunning Conditions
16	$NVL = NML \cdot (DM / DV)^{1/2}$	Same	All Swashplate Motors
	NVL = Refer to Technical Information manual	Same	H1B & Series 51 Bent Axis Motors
	$NVL \geq$ Reduced Angle Value		
17	$PC = \frac{HP \cdot 600\,000}{DP \cdot NE \cdot IR}$	$PC = \frac{HP \cdot 396\,000}{DP \cdot NE \cdot IR}$	
18	$FD = \frac{NML \cdot LR}{2650 \cdot SM}$	$FD = \frac{NML \cdot LR}{168 \cdot SM}$	Propel, Motor at Max Angle
	$FD = \frac{NML}{NMD}$	Same	Non-Propel, Motor at Max Angle
	$FD = \frac{NVL \cdot LR}{2650 \cdot SV}$	$FD = \frac{NVL \cdot LR}{168 \cdot SV}$	Propel, Motor at Min Angle
	$FD = \frac{NVL}{NVD}$	Same	Non-Propel, Motor at Min Angle

Selection of Driveline Components

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)	mm [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	—

Tractive Effort

Tractive Effort

For vehicle propel drives, motion resistance and required tractive efforts 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 pull ratio requirements have been estimated and are listed in the table *Pull Ratio Requirements for Vehicle Propel Drives* below. To establish the 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 calculated tractive effort values by testing an actual vehicle.

To determine Machine Function (MF) 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 speed 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** in the table **Pull Ratio Requirements for Vehicle Propel Drives** below is approximate. For propel drives which interact with work functions (cutters, planers, etc.), make an accurate determination of the required motion resistance by testing a working machine.

Transport mode should be used only for 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 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.

Grade motion Resistance (GR), or Gradability, is a function of terrain 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 of 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

Tractive Effort

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 (WT) 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 the power unit plus any towed trailer or wagon. For drawbar vehicles, maximum weight is only the weight of the power unit.

Simulating a trailer load: If we make the assumption that the coefficient of rolling resistance is the same for the vehicle and the trailer, then the weight of the trailer and its load can be added to the weight of the vehicle. If the coefficient of rolling resistance is different, then pro-rate the trailer weight before adding it to the vehicle weight with the following formula:

$$\text{Weight to add to the Vehicle} = \left\{ (\text{Total Trailer Weight}) * \frac{(RR_{\text{Trailer}})}{(RR_{\text{Vehicle}})} \right\}$$

Typical minimum design values of pull ratio for some common vehicles have been determined and are listed in the **Minimum Tractive Efforts Requirements** table below. 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.

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

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.

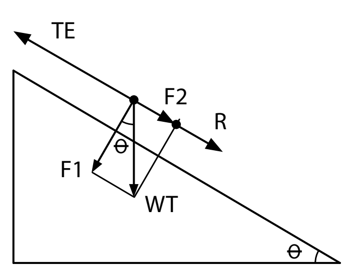
Tractive Effort

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

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.

Pull Ratio



Tractive Effort

Derivation of Maximum Gradability

Given:

TE = Tractive Force of Vehicle, N (lbf)

WT = Weight of Vehicle, N (lbf)

RR = Coefficient of Rolling Resistance, d'less

Find:

θ = Gradabiltiy Angle, degrees

Normal Force, $F1 = WT * \cos\theta$, N (lbf)

Tangential Force, $F2 = WT * \sin\theta$, N (lbf)

Rolling Resistance Force, $R = RR * F1 = RR * WT * \cos\theta$, N (lbf)

Maximum Gradability occurs when force uphill balances forces downhill:

$TE = R + F2 = RR * WT * \cos\theta + WT * \sin\theta$

Let $X = \cos\theta$, $\sin\theta = \sqrt{\left(1 - X^2\right)}$

$$TE = RR * WT * X + WT * \sqrt{\left(1 - X^2\right)}$$

$$\left(TE - RR * WT * X\right)^2 = \left(WT\right)^2 * \left(1 - X^2\right)$$

$$\left(TE^2 - WT^2\right) - \left(2 * TE * RR * WT * X\right) + \left(\left(1 + RR^2\right) * WT^2 * X^2\right) = 0$$

Solve Quadratic Equation for "X":

$$X = \frac{2 * TE * RR * WT \pm \sqrt{4 * TE^2 * RR^2 * WT^2 - 4 * \left(1 + RR^2\right) * WT^2 * \left(TE^2 - WT^2\right)}}{2 * \left(1 + RR^2\right) * WT^2}$$

$$X = \frac{RR * TE \pm \sqrt{WT^2 * \left(1 + RR^2\right) - TE^2}}{\left(1 + RR^2\right) * WT}$$

$$\text{Gradability, } \% = 100 * \tan\theta = 100 * \tan\left\{\cos^{-1}\left(\frac{RR * TE \pm \sqrt{WT^2 * \left(1 + RR^2\right) - TE^2}}{\left(1 + RR^2\right) * WT}\right)\right\}$$

Acceleration

Acceleration

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

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

t = (W) (V) (g) (F)

t = time (seconds)

W = Vehicle weight (lbs.)

V = Vehicle velocity (ft. per sec.) V = (MPH) (1.467)

g = Gravity (32.2 ft. per sec. per sec.)

F = Drawbar pull (lbs.) (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 effect 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 drives may also have acceleration time requirements and should not be overlooked during the equipment selection stage.

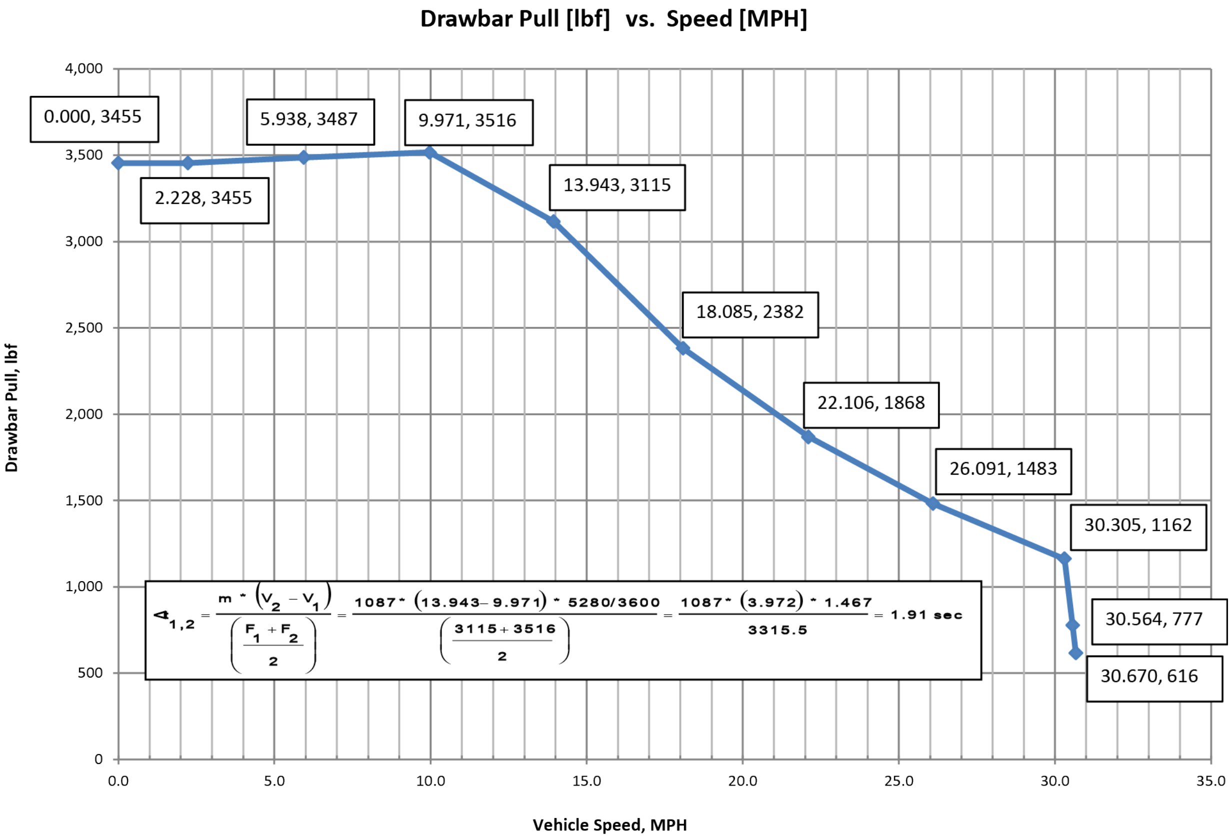
An example is attached using computer generated (P-Cubed) performance data.

Acceleration

Prime Mover Speed (rpm):	2800	Limiting Pressure (psid):	5500
Input Power (hp):	200	Charge Pressure (psid):	348

Delta System Pressure (psid)	5500	5500	5500	4961	4012	3356	2871	2477	2000	1800
Pump 1	**H1P165 - H1, 10.07cir, 18deg Max Angle, 1.59cir charge pump**									
Pump Drive Ratio:	1.00	Pump Drive Efficiency (%):	100							
Swashplate Angle (deg)	2.8	5.0	7.4	9.5	11.7	13.8	15.9	18.0	18.0	18.0
Displacement (cir)	1.50	2.70	4.00	5.20	6.40	7.60	8.80	10.07	10.07	10.07
Displacement (cc)	24.6	44.2	65.5	85.2	104.9	124.5	144.2	165.0	165.0	165.0
Input Shaft Speed (rpm)	2800	2800	2800	2800	2800	2800	2800	2800	2800	2800
Torque at Shaft (in-lb)	1646	2718	3877	4502	4502	4502	4502	4502	3724	3398
Actual Flow (gpm)	10.3	24.8	40.5	55.8	71.5	87.0	102.3	118.6	119.2	119.5
Power Loss (hp)	33.7	34.8	35.9	32.2	26.1	23.3	22.2	22.2	20.0	19.1
Charge Pump Loss (hp)	6.44	6.44	6.44	6.44	6.44	6.44	6.44	6.44	6.44	6.44
Volumetric Efficiency (%)	57	76	84	88	92	94	96	97	98	98
Torque Efficiency (%)	87	92	94	94	94	93	92	91	90	89
Overall Efficiency (%)	49	70	78	83	87	88	89	89	87	87
Total Pump Flow (gpm)	10.3	24.8	40.5	55.8	71.5	87.0	102.3	118.6	119.2	119.5
Final Drive Case 1										
Motor 1	**90M100 - Series 90, 6.1cir, 17deg Max Angle, AxialPiston**									
Rolling Radius (in):	15.00	Coeff. Rolling Resistance:	0.02	Final Drive Ratio:	6.5					
Weight Carried by Motor/Wheel (lb):	17500	Rolling Resistance (lb):	350	Final Drive Efficiency (%):	95					
Swashplate Angle (deg)	17.0	17.0	17.0	17.0	17.0	17.0	17.0	17.0	17.0	17.0
Displacement (cir)	6.10	6.10	6.10	6.10	6.10	6.10	6.10	6.10	6.10	6.10
Displacement (cc)	100.0	100.0	100.0	100.0	100.0	100.0	100.0	100.0	100.0	100.0
Shaft Speed (rpm)	162	432	726	1015	1317	1610	1900	2207	2226	2234
Torque at Motor Shaft (in-lb)	5047	5085	5121	4634	3743	3119	2650	2261	1794	1598
Power Loss (hp)	3.5	4.9	5.9	6.0	5.5	5.5	5.8	6.5	6.2	6.1
Volumetric Efficiency (%)	83	92	95	96	97	98	98	98	99	99
Torque Efficiency (%)	95	95	96	96	96	96	95	94	92	91
Overall Efficiency (%)	79	88	91	93	93	94	93	92	91	90
Torque at Wheel (in-lb)	31162	31401	31619	28613	23113	19260	16367	13962	11078	9865
Wheel Speed (rpm)	25	67	112	156	203	248	292	340	342	344
Tractive Force (lb)	2077	2093	2108	1908	1541	1284	1091	931	739	658
Motor 2	**90M100 - Series 90, 6.1cir, 17deg Max Angle, AxialPiston**									
Rolling Radius (in):	15.00	Coeff. Rolling Resistance:	0.02	Final Drive Ratio:	6.5					
Weight Carried by Motor/Wheel (lb):	17500	Rolling Resistance (lb):	350	Final Drive Efficiency (%):	95					
Swashplate Angle (deg)	17.0	17.0	17.0	17.0	17.0	17.0	17.0	17.0	17.0	17.0
Displacement (cir)	6.10	6.10	6.10	6.10	6.10	6.10	6.10	6.10	6.10	6.10
Displacement (cc)	100.0	100.0	100.0	100.0	100.0	100.0	100.0	100.0	100.0	100.0
Shaft Speed (rpm)	162	432	726	1015	1317	1610	1900	2207	2226	2234
Torque at Motor Shaft (in-lb)	5047	5085	5121	4634	3743	3119	2650	2261	1794	1598
Power Loss (hp)	3.5	4.9	5.9	6.0	5.5	5.5	5.8	6.5	6.2	6.1
Volumetric Efficiency (%)	83	92	95	96	97	98	98	98	99	99
Torque Efficiency (%)	95	95	96	96	96	96	95	94	92	91
Overall Efficiency (%)	79	88	91	93	93	94	93	92	91	90
Torque at Wheel (in-lb)	31162	31401	31619	28613	23113	19260	16367	13962	11078	9865
Wheel Speed (rpm)	25	67	112	156	203	248	292	340	342	344
Tractive Force (lb)	2077	2093	2108	1908	1541	1284	1091	931	739	658
Total Vehicle Performance										
Vehicle Weight (lb):	35000	Coeff. Rolling Resistance:	0.02	Rolling Resistance (lb):	700					
Input Power (hp)	73.1	120.8	172.3	200.0	200.0	200.0	200.0	200.0	165.5	151.0
Output Power (hp)	26.0	69.8	118.0	149.3	156.4	159.4	159.8	158.4	126.7	113.2
Power Loss (hp)	47.2	51.0	54.3	50.7	43.6	40.6	40.2	41.6	38.7	37.7
System Efficiency (%)	36	58	69	75	78	80	80	79	77	75
Tractive Force (lb)	4155	4187	4216	3815	3082	2568	2182	1862	1477	1315
Vehicle Speed (mph)	2.228	5.938	9.971	13.943	18.085	22.106	26.091	30.305	30.564	30.670
Vehicle Speed (fpm)	196	523	877	1227	1591	1945	2296	2667	2690	2699
Drawbar Pull @ 0% Grade (lb)	3455	3487	3516	3115	2382	1868	1483	1162	777	616
Gradeability (%)	9.94	10.03	10.11	8.95	6.83	5.35	4.24	3.32	2.22	1.76

Acceleration



Assumes average force between two speeds						
Drawbar pull	Speed	Speed	Time to accel	Cumulative Time	Accel	
(lb)	(mph)	(ft/sec)	(sec)	(sec)	(ft/sec^2)	g's
3455	0.000	0.000	0.000	0.000	0.000	0.000
3455	2.228	3.267	1.028	1.028	3.179	0.099
3487	5.938	8.709	1.704	2.732	3.208	0.100
3516	9.971	14.625	1.836	4.568	3.235	0.100
3115	13.943	20.449	1.909	6.477	2.866	0.089
2382	18.085	26.525	2.403	8.880	2.192	0.068
1868	22.106	32.422	3.016	11.896	1.719	0.053
1483	26.091	38.266	3.791	15.687	1.364	0.042
1162	30.305	44.447	5.081	20.768	1.069	0.033
777	30.564	44.827	0.425	21.193	0.715	0.022
616	30.670	44.983	0.244	21.437	0.567	0.018

Charge Pump Sizing

Introduction

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

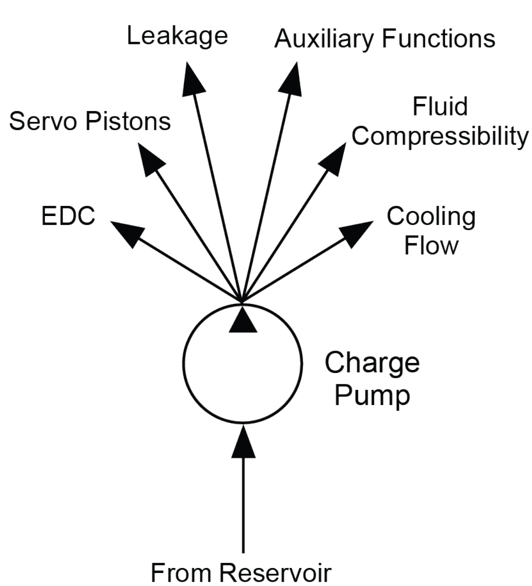
The primary function of the charge pump is to replenish fluid lost through leakage. In closed circuit hydrostatic systems, continual internal leakage of high pressure fluid is inherent in the design of the components used in such a system, and will generally increase as the displacement of the system's 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.

Another function of the charge pump includes providing fluid for servo pistons on those systems having servo-controlled transmissions. 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 additional flow.

The *Charge Pump Functions* illustration shows the functions that the charge pump may be required to provide in a given application.

Charge Pump Functions



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 is only an approximation for a simple system containing high speed piston components. The best way to size a charge pump is to individually consider each demand imposed on the charge pump. Many of these requirements do not occur in a simple hydrostatic circuit.

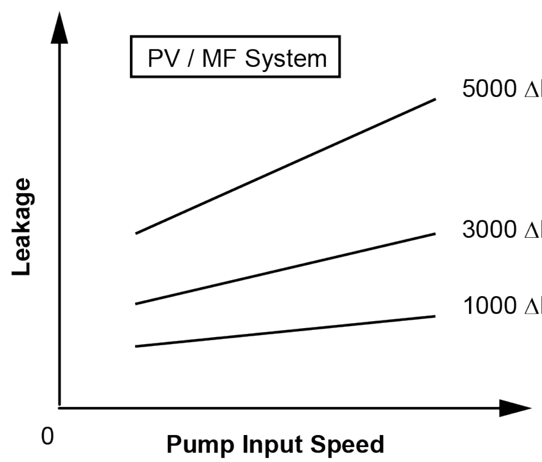
To properly size a charge pump, each of the following factors must be taken into account:

Charge Pump Sizing

- System Pressure, and rate-of-change of system pressure (ΔP)
- Input Speed
- Minimum operational input speed
- Line size, length, and Effective Bulk Modulus of the fluid
- Control requirements
- Non-Danfoss components
- Type of loading

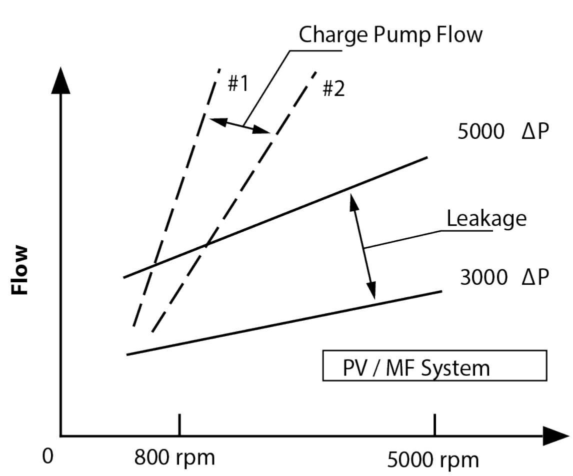
The *System Leakage* graph shows how system pressure and input speed affect leakage in the system. The graph shows that leakage increases with both higher system pressure and higher input speed. Changes in pressure have a greater effect on leakage than changes in speed. However, the affects due to changes in speed are greater at higher system pressures.

System Leakage



The *Charge Flow and Leakage* graph 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. .

Charge Flow and Leakage



Disregarding for the moment all other charge pump requirements, other than leakage; for a given speed and pressure (and temperature) a charge pump has a flow curve which intersects the system leakage curve. At low speeds and high pressures, the potential system leakage may exceed the flow that the

Charge Pump Sizing

charge pump is capable of providing. Furthermore, the charge pump's Volumetric Efficiency decreases with decreasing speed. Therefore, even though leakage rate may be greater at high pump input speeds, the largest charge pump displacement may be required at a reduced input speed. Both extremes of 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.

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

Make sure that 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 "control requirements" for servo-controlled pumps is dependent on the stroke rate and the servo volume. Normally, the flow required is in the range of 2 to 8 lpm [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 2 to 4 lpm [1/2 to 1 gpm]. This flow rate is needed for operation of the Pressure Control Pilot Valve (PCP), which regulates the position of the control spool of the displacement control. 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 valve is specified. The charge pump displacement must then accommodate this additional requirement for 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. **Bulk modulus** is the inverse of a material property, Compressibility, which is defined as the amount that 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 the influence of 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 the return flow rate into the low pressure side of the system. This reduction of return flow rate must be provided by the charge system, 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 diameter 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, a section has been included in *Fluids and Lubricants* **520L0463**. An example calculation is provided below to bring special attention to this topic.

Charge Pump Sizing

A system with 30 feet of one inch I.D. hose has an abrupt change in system pressure from 2000 psid (138 bar) to 5000 psid (345 bar). This change occurs in 100 milliseconds. Calculate the charge flow rate requirement due to the effects of fluid compressibility.

Change in pressure = 5000-2000 = 3000 psid.

Volume under pressure:

$$V = \frac{(30\ \text{ft})(12\ \text{in/ft}) * \pi * (1\ \text{in})^2}{4} = 282\ \text{in}^3$$

$$V = \frac{(914\ \text{cm}) * \pi * (2.54\ \text{cm})^2}{4 * (1000\ \text{cc/liter})} = 4.63\ \text{liters}$$

Using a Bulk Modulus of 150 000 psi (10 345 bar),

The required flow rate is:

$$Q = \frac{(3000\ \text{psid}) * (282\ \text{in}^3)}{(150\,000\ \text{psid}) * (0.100\ \text{sec})} = 56.5\ \ \text{in}^3/\text{second}$$

$$Q = \frac{(207\ \text{bar}) * (4.63\ \text{liters})}{(10\,345\ \text{bar}) * (0.100\ \text{sec})} = 0.93\ \ \text{liters/second} * (60\ \text{sec/min}) = 55.6\ \ \text{liters/min}$$

The required charge pump displacement is the one 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 Danfoss pumps include an auxiliary pad to mount gear pumps of various displacements.

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

Use the Charge Pump Sizing Worksheet below to size a charge pump. Each of the charge flow requirements is 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 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.

Charge Pump Sizing Worksheet

Leakage:

"Pump" refers to hydrostatic pump, not the charge pump. Actually, only 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 (i.e. case flow), the calculations below are conservative. If case flow values are available, they should be used instead of the equations below.

Charge Pump Sizing

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
	______	lpm

$$\text{Pump Flow} = \frac{\text{Pump Disp, cir * Pump RPM}}{231\,\text{in}^3/\text{gal}} * \frac{\text{Pump Efficiency, \%}}{100}, \text{gpm}$$

$$\text{Pump Flow} = \frac{\text{Pump Disp, cc * Pump RPM}}{1000\,\text{cc/liter}} * \frac{\text{Pump Efficiency, \%}}{100}, \text{lpm}$$

$$\text{Pump Leakage} = \frac{\text{Pump Disp, cir * Pump RPM}}{231\,\text{in}^3/\text{gal}} * \left(1 - \frac{\text{Pump Efficiency, \%}}{100}\right), \text{gpm}$$

$$\text{Pump Leakage} = \frac{\text{Pump Disp, cc * Pump RPM}}{1000\,\text{cc/liter}} * \left(1 - \frac{\text{Pump Efficiency, \%}}{100}\right), \text{lpm}$$

$$\text{Motor Speed} \quad \frac{(\text{Pump Flow, gpm}) * \left(231\,\text{in}^3\,/\,gal\right)(\text{Motor Efficiency, \%/100})}{\left(\text{Motor Disp, in}^3/\text{rev}\right) * \#\ \text{Motors}}, RPM$$

$$\text{Motor Speed} = \frac{(\text{Pump Flow, lpm}) * \left(1000\,\text{cm}^3\,/\,liter\right)(\text{Motor Efficiency, \%/100})}{\left(\text{Motor Disp, cm}^3/\text{rev}\right) * \#\ \text{Motors}}, RPM$$

$$\text{MotorLeakage} = \left(\frac{\text{Pump Flow}}{\#\ \text{Motors}}\right) * \left(1 - \frac{\text{Motor Efficiency, \%}}{100}\right),\ \text{gpm (lpm)}$$

Note : Consult product technical information bulletins
for values of Volumetric Efficiency.

Charge Pump Sizing

Control Requirements:

Control Type

______	DDC	
______	MDC	________ gpm (lpm)
______	HDC	________ gpm (lpm)
______	EDC	________ gpm (lpm)
______	NFP	________ gpm (lpm)
______	Other	________ gpm (lpm)

For most applications with 1 -3 second stroke times, assume a value of 0.5 gpm (2 lpm).
For atypical stroke times use the chart and the equation below:

$$\text{Flow Rate} = \frac{\text{Servo Volume, in}^3 * 0.26}{\text{Stroke Time, sec}}, \text{gpm}$$

$$\text{Flow Rate} = \frac{\text{Servo Volume, cm}^3 * 0.06}{\text{Stroke Time, sec}}, \text{lpm}$$

For pumps with EDC controls, add 0.75 gpm (3 lpm)to the servo flow to allow for losses in the PCP.

Series	Servo Volume (in^3)	Servo Volume (cm^3)
Series 40, M46	1.50	24.5
Series 42, 28 cc/rev	1.00	16.4
Series 42, 41 cc/rev	1.50	24.5
Series 90, 55 cc/rev	1.30	21.3
Series 90, 75 cc/rev	1.70	27.9
Series 90, 100 cc/rev	2.50	41.0
Series 90, 130 cc/rev	3.50	57.4
Series 90, 180 cc/rev	5.00	81.9
Series 90, 250 cc/rev	5.00	81.9
H1P045/H1T045	1.45	23.8
H1P053/H1T053	1.45	23.8
H1P060	1.95	31.8
H1P068	1.95	31.8
H1P078	2.20	36.5
H1P089	2.90	47.1
H1P100	2.90	47.1
H1P115	3.70	60.9
H1P130	3.70	60.9
H1P147	4.75	77.8
H1P165	4.75	77.8

Charge Pump Sizing

Series	Servo Volume (in^3)		Servo Volume (cm^3)	
H1 Bent Axis Motors	2-Position Control			
	$V_{max\ to\ min}$ (in^3)	$V_{min\ to\ max}$ (in^3)	$V_{max\ to\ min}$ (cm^3)	$V_{min\ to\ max}$ (cm^3)
H1B060	2.01	0.99	33.0	16.2
H1B080	2.89	1.52	47.4	25.0
H1B110	3.82	1.87	62.5	30.6
H1B160	5.73	2.92	93.9	47.9
H1B250	6.63	3.38	108.7	55.4
	Proportional Control			
	$V_{max\ to\ min}$ (in^3)	$V_{min\ to\ max}$ (in^3)	$V_{max\ to\ min}$ (cm^3)	$V_{min\ to\ max}$ (cm^3)
H1B060	1.72	0.84	28.3	13.8
H1B080	2.48	1.31	40.6	21.4
H1B110	3.30	1.62	54.1	26.5
H1B160	4.88	2.49	80.1	40.8
H1B250	5.49	2.81	90.0	46.0

Charge Pump Sizing

2 Position Control: Step Response, seconds

Step response has been tested with a 2-Position control (de-energized max. angle) at delta p = 30 bar. 210 bar and 400 bar and orifices for Servo M4 & M5
For Max. to Min., current went from zero to max, then back to zero current.

60cc		Orifice diameter			
		w/o	1.2 mm	0.8 mm	0.6 mm
30 bar	Max → Min (s) (100% → 20%)	0.22	0.47	1.13	1.70
	Min → Max (s) (20% → 100%)	0.21	0.52	1.00	1.56
210 bar	Max → Min (s) (100% → 20%)	0.14	0.26	0.58	0.85
	Min → Max (s) (20% → 100%)	0.15	0.25	0.45	0.71
400 bar	Max → Min (s) (100% → 20%)	0.13	0.21	0.46	0.65
	Min → Max (s) (20% → 100%)	0.15	0.20	0.33	0.51

110cc		Orifice diameter			
		w/o	1.2 mm	0.8 mm	0.6 mm
30 bar	Max → Min (s) (100% → 20%)	0.56	1.40	1.94	3.60
	Min → Max (s) (20% → 100%)	0.44	1.13	1.89	3.59
210 bar	Max → Min (s) (100% → 20%)	0.22	0.45	0.80	1.38
	Min → Max (s) (20% → 100%)	0.21	0.41	0.92	1.67
400 bar	Max → Min (s) (100% → 20%)	0.22	0.33	0.58	0.97
	Min → Max (s) (20% → 100%)	0.22	0.31	0.69	1.25

80cc		Orifice diameter			
		w/o	1.2 mm	0.8 mm	0.6 mm
30 bar	Max → Min (s) (100% → 20%)	0.30	0.61	1.50	2.80
	Min → Max (s) (20% → 100%)	0.40	0.71	1.50	2.70
210 bar	Max → Min (s) (100% → 20%)	0.14	0.30	0.74	1.32
	Min → Max (s) (20% → 100%)	0.19	0.34	0.73	1.16
400 bar	Max → Min (s) (100% → 20%)	0.11	0.23	0.57	1.00
	Min → Max (s) (20% → 100%)	0.14	0.27	0.54	0.90

160cc		Orifice diameter			
		w/o	1.2 mm	0.8 mm	0.6 mm
30 bar	Max → Min (s) (100% → 20%)	0.48	1.32	2.62	4.36
	Min → Max (s) (20% → 100%)	0.50	1.30	3.03	4.92
210 bar	Max → Min (s) (100% → 20%)	0.27	0.68	1.25	2.10
	Min → Max (s) (20% → 100%)	0.28	0.71	1.61	2.57
400 bar	Max → Min (s) (100% → 20%)	0.31	0.50	0.90	1.47
	Min → Max (s) (20% → 100%)	0.24	0.56	1.24	2.04

250cc		Orifice diameter			
		w/o	1.2 mm	0.8 mm	0.6 mm
30 bar	Max → Min (s) (100% → 20%)	0.55	1.56	3.28	5.59
	Min → Max (s) (20% → 100%)	0.62	1.46	3.52	5.94
210 bar	Max → Min (s) (100% → 20%)	0.32	0.72	1.38	2.21
	Min → Max (s) (20% → 100%)	0.31	0.72	1.64	2.73
400 bar	Max → Min (s) (100% → 20%)	0.33	0.53	0.97	1.50
	Min → Max (s) (20% → 100%)	0.25	0.55	1.24	2.03

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Charge Pump Sizing

Proportional Control: Step Response, seconds

Step response has been tested with a Proportional control (de-energized max. angle) at delta p = 30 bar. 210 bar and 400 bar and orifices for Servo M4 & M5
For Max. to Min., current went from zero to max, then back to zero current.

060cc		Orifice diameter			
		w/o	1.2 mm	0.8 mm	0.6 mm
30 bar	Max → Min (s) (100% → 20%)	0.30	0.47	0.96	1.59
	Min → Max (s) (20% → 100%)	0.04	0.50	0.95	1.37
	50% → 20% (s)	0.20	0.25	0.43	0.68
	20% → 50% (s)	0.21	0.27	0.42	0.62
210 bar	Max → Min (s) (100% → 20%)	0.24	0.31	0.53	0.84
	Min → Max (s) (20% → 100%)	0.23	0.32	0.46	0.65
	50% → 20% (s)	0.20	0.19	0.27	0.39
	20% → 50% (s)	0.20	0.20	0.24	0.32
400 bar	Max → Min (s) (100% → 20%)	0.28	0.27	0.43	0.65
	Min → Max (s) (20% → 100%)	0.24	0.26	0.38	0.49
	50% → 20% (s)	0.23	0.21	0.24	0.33
	20% → 50% (s)	0.21	0.20	0.22	0.27

110cc		Orifice diameter			
		w/o	1.2 mm	0.8 mm	0.6 mm
30 bar	Max → Min (s) (100% → 20%)	0.51	1.27	1.76	3.27
	Min → Max (s) (20% → 100%)	0.40	1.03	1.72	3.26
	50% → 20% (s)	0.20	0.34	0.69	1.11
	20% → 50% (s)	0.18	0.34	0.68	1.16
210 bar	Max → Min (s) (100% → 20%)	0.20	0.41	0.73	1.25
	Min → Max (s) (20% → 100%)	0.19	0.37	0.84	1.52
	50% → 20% (s)	0.18	0.16	0.29	0.45
	20% → 50% (s)	0.19	0.20	0.37	0.61
400 bar	Max → Min (s) (100% → 20%)	0.20	0.30	0.53	0.88
	Min → Max (s) (20% → 100%)	0.20	0.28	0.63	1.14
	50% → 20% (s)	0.33	0.31	0.21	0.33
	20% → 50% (s)	0.24	0.18	0.33	0.56

080cc		Orifice diameter			
		w/o	1.2 mm	0.8 mm	0.6 mm
30 bar	Max → Min (s) (100% → 20%)	0.36	0.78	1.39	2.28
	Min → Max (s) (20% → 100%)	0.30	0.59	1.45	2.35
	50% → 20% (s)	0.17	0.30	0.50	0.89
	20% → 50% (s)	0.16	0.37	0.53	0.95
210 bar	Max → Min (s) (100% → 20%)	0.16	0.40	0.56	0.97
	Min → Max (s) (20% → 100%)	0.27	0.31	0.67	1.16
	50% → 20% (s)	0.18	0.18	0.20	0.34
	20% → 50% (s)	0.17	0.20	0.25	0.43
400 bar	Max → Min (s) (100% → 20%)	0.19	0.34	0.40	0.69
	Min → Max (s) (20% → 100%)	0.32	0.25	0.50	0.86
	50% → 20% (s)	0.17	0.20	0.15	0.24
	20% → 50% (s)	0.13	0.20	0.19	0.33

160cc		Orifice diameter			
		w/o	1.2 mm	0.8 mm	0.6 mm
30 bar	Max → Min (s) (100% → 20%)	0.52	1.15	2.08	3.58
	Min → Max (s) (20% → 100%)	0.42	1.03	2.12	3.74
	50% → 20% (s)	0.27	0.48	0.84	1.45
	20% → 50% (s)	0.35	0.48	0.90	1.52
210 bar	Max → Min (s) (100% → 20%)	0.40	0.58	1.05	1.64
	Min → Max (s) (20% → 100%)	0.28	0.58	1.15	1.91
	50% → 20% (s)	0.43	0.31	0.45	0.67
	20% → 50% (s)	0.49	0.38	0.53	0.81
400 bar	Max → Min (s) (100% → 20%)	0.53	0.49	0.75	1.18
	Min → Max (s) (20% → 100%)	0.24	0.46	0.87	1.43
	50% → 20% (s)	0.56	0.39	0.35	0.50
	20% → 50% (s)	0.57	0.43	0.45	0.68

250cc		Orifice diameter			
		w/o	1.2 mm	0.8 mm	0.6 mm
30 bar	Max → Min (s) (100% → 20%)	0.58	1.29	3.12	4.68
	Min → Max (s) (20% → 100%)	0.50	1.14	2.89	4.93
	50% → 20% (s)	0.28	0.53	0.98	1.63
	20% → 50% (s)	0.23	0.48	1.00	1.76
210 bar	Max → Min (s) (100% → 20%)	0.24	0.55	1.12	1.74
	Min → Max (s) (20% → 100%)	0.24	0.56	1.27	2.27
	50% → 20% (s)	0.42	0.43	0.46	0.66
	20% → 50% (s)			0.55	0.92
400 bar	Max → Min (s) (100% → 20%)	0.19	0.41	0.82	1.24
	Min → Max (s) (20% → 100%)	0.19	0.43	0.97	1.72
	50% → 20% (s)	0.55	0.50	0.35	0.49
	20% → 50% (s)			0.48	0.78

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Charge Pump Sizing

Loop Flushing

Loop Flushing Flow ___________ gpm (lpm)

The amount of loop flushing flow will normally vary between 2 – 4 gpm (8 – 16 lpm) depending on the charge pump displacement, input speed, and relative settings between the pump and motors charge relief valves.

Fluid Compressibility

Magnitude of pressure spike, ΔP	_______________ psi (bar)
Time Duration, Δt	_______________ sec
Bulk Modulus, BM	_______________ psi (bar)
Hose Length, L	_______________ ft (cm)
Hose I.D., D	_______________ in. (cm)
Hose Volume, V	_______________ in^3 (cm^3)
Charge flow required	_______________ gpm (lpm)

$$\text{Hose Volume} = V = \frac{\pi * (\text{ID, in})^2 * (\text{Length, ft}) * (12 \text{ in/ft})}{4}, \text{ in}^3$$

$$\text{Hose Volume} = V = \frac{\pi * (\text{ID, cm})^2 * (\text{Length, cm})}{4 * (1000 \text{ cm}^3/\text{liter})}, \text{ liters}$$

$$Q = \frac{(\Delta P) * (V)}{(BM) * \Delta t} * 0.26, \text{ gpm}$$

$$Q = \frac{(\Delta P) * (V)}{(BM) * \Delta t} * 60, \text{ lpm}$$

Auxiliary Functions

Hydraulically released brakes	_______________ gpm (lpm)
Two-speed motor shifting	_______________ gpm (lpm)
Cylinders	_______________ gpm (lpm)
Other components	_______________ gpm (lpm)
Total Auxiliary Flow	_______________ gpm (lpm)

Charge Pump Sizing

Total Charge Flow Required

Leakage + Control + Loop Flushing + Compressibility + Auxiliary = ________________ gpm (lpm)

Select a preliminary charge pump displacement:

Charge pump displacement ________________ cubic inch/rev (cc/rev)
Volumetric Efficiency ________________ %
Charge flow provided ________________ gpm (lpm)

$$\text{Charge Flow} = \frac{(\text{Charge Displacement, cir})*(\text{Input Speed, RPM})*(\text{Efficiency, } \%/100)}{231}, \text{ gpm}$$

$$\text{Charge Flow} = \frac{(\text{Charge Displacement, cc/rev})*(\text{Input Speed, RPM})*(\text{Efficiency, } \%/100)}{1000}, \text{ lpm}$$

Is the charge pump capable of providing adequate charge flow?

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

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Local address:

Danfoss
Power Solutions (US) Company
2800 East 13th Street
Ames, IA 50010, USA
Phone: +1 515 239 6000

Danfoss
Power Solutions GmbH & Co. OHG
Krokamp 35
D-24539 Neumünster, Germany
Phone: +49 4321 871 0

Danfoss
Power Solutions ApS
Nordborgvej 81
DK-6430 Nordborg, Denmark
Phone: +45 7488 2222

Danfoss
Power Solutions Trading (Shanghai) Co., Ltd.
Building #22, No. 1000 Jin Hai Rd
Jin Qiao, Pudong New District
Shanghai, China 201206
Phone: +86 21 3418 5200