ENGINEERING
TOMORROW

Applications Manual

Pressure and Speed Limits for Hydrostatic Units



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Revision history

Table of revisions

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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

Pressure and Speed Limits

Introduction

In order to properly size hydrostatic units for power transmission applications, it is necessary to select an appropriate **design pressure** and **design speed**. The selection of these design parameters are affected by required **unit life**. The purpose of this guideline is to assist system integrators in the selection of design pressure and design speed to optimize hydraulic unit utility and to provide satisfactory transmission life.

Danfoss Power Solutions publishes continuous (1) ratings and maximum ratings for both pressure and speed capabilities. Application decisions are made by considering both typical and extreme operating conditions. Predicted unit life, based on the design pressure and speed, is compared to the required unit life for the application. Satisfactory design criteria must be obtained for both normal rated and maximum values.

(1) The term "continuous" refers to the speed ratings of Danfoss Power System products, but not to their pressure ratings.

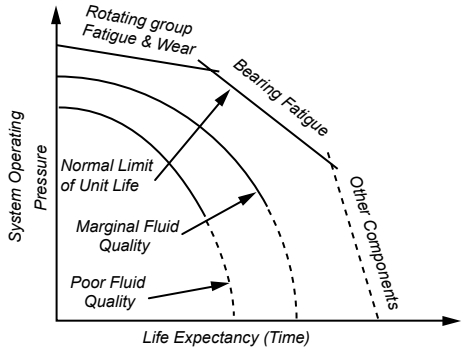
Load to Life Relationships

System pressure is the dominant operating variable affecting hydraulic unit life. High pressure, which results from high load, reduces life in a manner similar to many mechanical assemblies such as engines and gear boxes. There are load-life relationships for the rotating group and for the shaft bearings which must be considered separately.

The graph *Hydraulic Unit Life vs System Pressure* shows the general relationship between life expectancy and system pressure. The upper limit of hydraulic unit life is established by fatigue and wear of the rotating group, bearing fatigue, or wear and deterioration of other parts, each with their own load-life relationship. At high system pressures, **fatigue and wear of the rotating group** is the limiting factor, while **bearing failure** becomes the limiting factor at intermediate pressures. It is important to note the shallow slope of the life curve at high system pressures which indicates a significant decrease in life with a slight increase in pressure.

Actual life is also affected by a combination of factors relating to fluid quality as shown in *Hydraulic Unit Life vs System Pressure* by the curves corresponding to less-than-ideal fluid quality. Factors relating to fluid quality include temperature, contamination (heat, hard particle, water, chemical, and entrained air), viscosity, and lubrication properties. Most of these factors are also influenced by operating pressure. System design should take these factors into account. Technical Information Manuals (*Hydraulic Fluids and Lubricants* 520L0463, *Experience with Biodegradable Hydraulic Fluids* 520L0465, and *Design Guideline for Hydraulic Fluid Cleanliness* 520L0467) cover this subject in more detail.

Pressure and Speed Limits



Hydraulic Unit Life vs System Pressure

Pressure Limits for Hydraulic Units

The following definitions of hydraulic unit pressure capability and associated life requirements are provided to assist in sizing transmission components. Contact a Danfoss Power Solutions representative with your machine parameters if there is a question regarding typical load cycles.

In closed circuit hydrostatic units, **System Pressure** is the differential (delta) pressure between the high and low pressure work ports. It is the measure of the pressure available to accept or provide torque through the conversion of mechanical to/from hydraulic power. Zero System Pressure is the condition where both sides of the hydrostatic loop are at equal pressure, typically at Charge Pressure. System Pressure is the dominant operating variable affecting hydraulic unit life and performance. High system pressure, which results from high load, reduces expected life. Hydraulic unit life depends on the speed and normal operating, or weighted average, pressure which can only be determined from a duty cycle analysis.

In open circuit units, **System Pressure** is the gauge pressure of the working port since the low pressure reference is zero gage pressure at the reservoir.

In closed circuit hydrostatic units, it is common to provide a flow source at a moderate pressure to provide makeup flow (charge flow) for the hydrostatic loop. The pressure of this flow source is called **Charge Pressure** and is usually measured in terms of delta pressure above case pressure. The lower pressure side of the hydrostatic power loop will typically be at, or near, charge pressure. In open circuit systems, charge pressure may be considered to be the pressure available to the inlet of the pump or the outlet of the motor. It may be either below atmospheric (0 Gauge Pressure) or above. If the charge pressure is above atmospheric, it is often known as Boost Pressure for a pump or Back Pressure for a motor.

In closed circuit units, **Minimum Pressure** is the gauge pressure that must be maintained in the working ports under all operating conditions to avoid excessive cavitation and to provide lubrication for sliding components. Minimum pressure is typically less than Charge Pressure. Minimum Pressure limits are identified in the Technical Information Bulletin for each product. Unless otherwise specified, Minimum Pressure is defined as 10 bar [150 psig].

In open circuit units, Minimum Inlet Pressure is the minimum allowable absolute pressure in the pump suction line. The minimum inlet pressure must be evaluated in both steady state, and transient conditions; note that when pump displacement increases, inlet pressure is reduced due to acceleration of fluid in the inlet line.

Application Pressure is the highest intermittent pressure defined for a given application; and is the high pressure relief, pressure limiter, or pressure compensator setting typically defined within the order code of the pump or motor. It is determined by the maximum machine load demand. Application Pressure is the "applied" System Pressure (delta pressure) at which the driveline generates the maximum calculated pull or torque, or at which maximum loads move in the application. Application Pressure is typically determined from the maximum design loads of the vehicle driveline or work function components. Application Pressure is assumed to occur for a small percentage of operating time, usually less than 2% of the total.

Pressure and Speed Limits

For systems with pressure limiter valves acting to avoid frequent overloads, it may be necessary to select a reduced maximum design pressure below that normally allowed because time at maximum Application Pressure may be excessive.

Maximum Working Pressure is the highest recommended Application Pressure. Maximum Working Pressure is the highest standard pressure setting allowed within the product code. Systems with Application Pressures at, or below, the Maximum Working Pressure should yield predicted (calculated) unit life given proper component sizing. Maximum Working Pressure limits are identified in the Technical Information manual for each product. Application Pressures above Maximum Working Pressure will only be considered with duty cycle analysis and factory approval. (Factory approval means Engineering approval is needed.)

Maximum Pressure is the highest allowable System Pressure under any circumstance. System Pressures above Maximum Pressure may result in damage to the hydrostatic unit which could result in reduction in useful life below what would be predicted by existing design/life analysis tools; and/or a permanent reduction in the performance or efficiency of the hydrostatic unit.

Root Mean Effective Pressure is the time weighted average for a pressure duty cycle using a specified weighting exponent relative to the internal components under consideration. RMEP is a calculated System Pressure equivalent for a specific duty cycle. It can be used to compare the relative severity of loading cycles and used for component life predictions.

The formula for calculating the RMEP value is shown below:

$$RMEP = \left[\sum_{i=1}^{T} (Pressure_i)^a * (\%\, time_i)\right]^{\left(\frac{1}{a}\right)}$$ ¶

Where:

RMEP = Root Mean Effective Pressure

Pressure = System Pressure average for the time step

% time = Total time for the time step expressed as a decimal percent

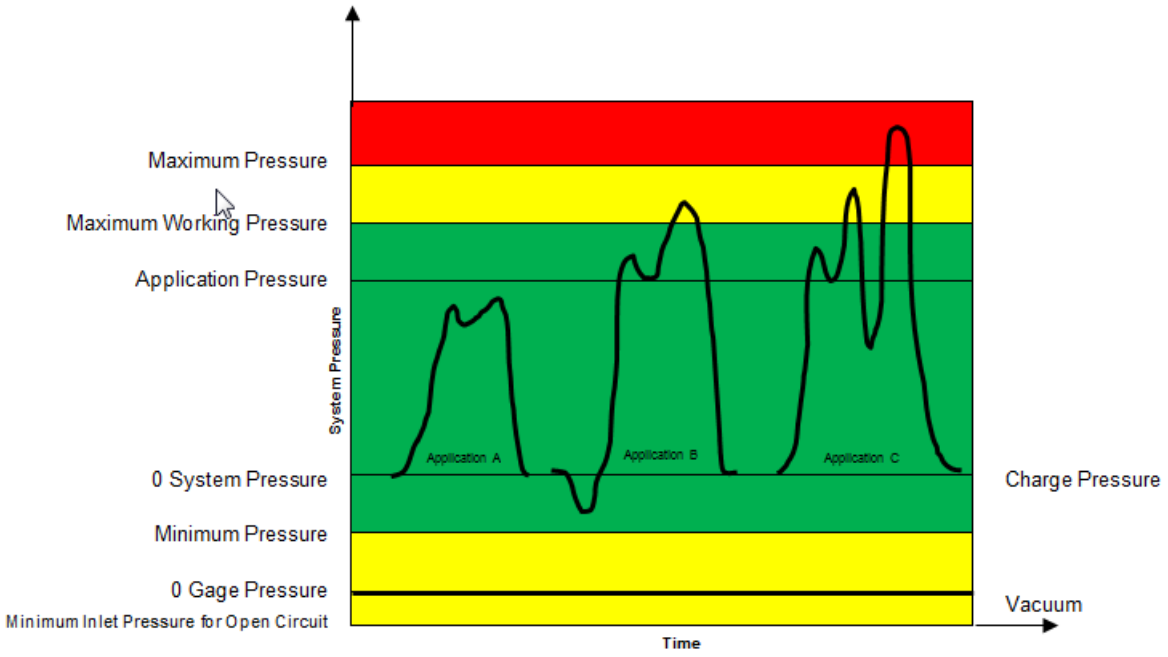
a = cumulative fatigue damage exponent for the unit or the component under consideration

- a = 8 for rotating groups and shafts in bending fatigue
- a = 3 for ball bearings
- a = 10/3 for tapered or cylindrical rolling element bearings

The exponent of 8 is derived from mechanical fatigue and is appropriate for the all rotating components except bearings. The exponents of 3 and 10/3 are derived from the bearing industry standard practice.

Pressure and Speed Limits

Pressure versus Time chart showing key pressure terms



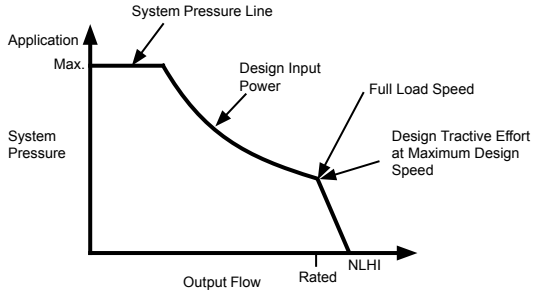
Application A - The System Pressure duty cycle stays under the Application Pressure. The unit in this application should have satisfactory predicted life provided the fluid viscosity, quality and cleanliness are maintained.

Application B - The delta system pressure starts out within the maximum working pressure but has a spike above Maximum Working Pressure. The trace also shows a spike below charge pressure, but still above the minimum pressure. The unit in this application would perform as expected, but has the potential of having a somewhat shorter predicted life, dependent upon the total amount of time spent near Maximum Pressure.

Application C - The system delta pressure trace shows a spike significantly above Maximum Pressure. The unit in this application may suffer damage and have its expected life significantly shortened; and/or no longer provide expected performance.

Normal working pressure is the pressure expected when performing “normal” work functions. It is the average pressure across the normal work load range.

The normal working pressure occurs at a regularly occurring load condition. Its design value can be established by calculating system pressure at the design input power and maximum pump displacement (see the following graph). For machines with variable load cycles, the design input power may be estimated by deducting the average power to the other functions from the maximum available engine power.



Pressure and Speed Limits

The preferred method of selecting an Application Pressure requires duty cycle information for the application. A duty cycle is a means of quantifying the pressure and speed demands of a particular system on a percent time basis. The duty cycle data is weighted to reflect the preponderant influence of pressure on unit life. Using the rotating group and shaft bearing load-life tools developed by Danfoss Power Solutions for the various product families, the optimum Application Pressure can be selected based on the required life expectancy for the application.

If an external side load is expected on the shaft, its magnitude and direction with respect to the swashplate must be known when calculating expected bearing life.

It is very important that the duty cycle information be as accurate as possible due to the exponential effect that pressure has on the life of the rotating group.

In the absence of duty cycle data, allowable pressure limits for design can usually be established based on the maximum and normal working load in the drive. This assumes that the duty cycle falls within a typical band width. By using this approach, two “pressure limit” values can be used to predict a successful application when machine life requirements are known.

The following table is provided to aid the engineer in understanding and selecting appropriate design pressure for a given component life requirement. These values are derived from the load-life relationship for the rotating groups of the various Danfoss Power Solutions product families at 1800 rpm shaft speed and no external shaft loads. Unit life is given in terms of L20 hours (80% survival) for the rotating group based on published recommendations for fluid cleanliness and viscosity. Using these guidelines, the system must meet both the maximum and continuous pressure limits to be considered acceptable.

Prototype test programs to verify operating pressures and predicted unit life are strongly recommended prior to finalizing any hydrostatic drive design.

Pressure and Speed Limits

Design Pressure Recommendations as a Function of Required Life

Required Life (L20 Hours)	Series 15		Series 40		KV/LV		Series 42		Series 90		H1P		Series 51		H1B	
	Displacement (cc/rev) & Root Mean Effective Pressure, at Full Displacement, Psid [dbar]															
1500	15	3200[220]	25	3675[253]	L25	4150[286]	28	4150[286]	42	4900[338]	45	4500[310]	60	4900[338]	60	>4150[286]
			35	4150[286]	L30	3650[252]	32	3650[252]	55	4900[338]	53	3900[269]	80	4900[338]	80	>4150[286]
			44	4025[277]	L35	2875[198]	41	4150[286]	75	4900[338]	60	4500[310]	110	4900[338]	110	>4150[286]
			PV046	3675[253]	K38	3650[252]	51	3650[252]	100	4900[338]	68	3900[269]	160	4900[338]	160	>4150[286]
			MV046	3675[253]	K45	2875[198]			130	4900[338]	69	4900[338]	250	4900[338]	250	>4150[286]
			MF046	3450[238]					180	4900[338]	78	4900[338]				
									250	4900[338]	89	4900[338]				
											100	4900[338]				
											115	4900[338]				
											130	4900[338]				
											147	4900[338]				
											165	4900[338]				
											210	4900[338]				
											250	4900[338]				
5000	15	2750[190]	25	3175[219]	L25	3600[248]	28	3600[248]	42	4225[291]	45	3850[265]	60	4225[291]	60	>3600[248]
			35	3600[248]	L30	3125[215]	32	3125[215]	55	4225[291]	53	3375[233]	80	4225[291]	80	>3600[248]
			44	3450[238]	L35	2475[171]	41	3600[248]	75	4225[291]	60	3850[265]	110	4225[291]	110	>3600[248]
			PV046	3175[219]	K38	3125[215]	51	3125[215]	100	4225[291]	68	3375[233]	160	4225[291]	160	>3600[248]
			MV046	3175[219]	K45	2475[171]			130	4225[291]	69	4225[291]	250	4225[291]	250	>3600[248]
			MF046	2975[205]					180	4225[291]	78	4225[291]				
									250	4225[291]	89	4225[291]				
											100	4225[291]				
											115	4225[291]				
											130	4225[291]				
											147	4225[291]				
											165	4225[291]				
											210	4225[291]				
											250	4225[291]				
15000	15	2400[165]	25	2775[191]	L25	3125[215]	28	3125[215]	42	3650[252]	45	3350[231]	60	3650[252]	60	>3125[215]
			35	3125[215]	L30	2725[188]	32	2725[188]	55	3650[252]	53	2925[202]	80	3650[252]	80	>3125[215]
			44	3000[207]	L35	2150[148]	41	3125[215]	75	3650[252]	60	3350[231]	110	3650[252]	110	>3125[215]
			PV046	2750[190]	K38	2725[188]	51	2725[188]	100	3650[252]	68	2925[202]	160	3650[252]	160	>3125[215]
			MV046	2750[190]	K45	2150[148]			130	3650[252]	69	3650[252]	250	3650[252]	250	>3125[215]
			MF046	2600[179]					180	3650[252]	78	3650[252]				
									250	3650[252]	89	3650[252]				
											100	3650[252]				
											115	3650[252]				
											130	3650[252]				
											147	3650[252]				
											165	3650[252]				
											210	3650[252]				
											250	3650[252]				

Speed Ratings for Hydraulic Units

The applied speed of hydraulic units is limited by several design and application factors which have varying effects on life. In general, hydraulic unit life varies inversely proportional to speed at, or below, the (continuous) speed limit. Operation at speeds in excess of the continuous speed limit causes less than normal life but is permissible at less than full power. The maximum speed limit cannot be exceeded without significant reduction in life and without risking immediate failure resulting in a loss of hydrostatic power.

Warning

Danfoss Power Solutions strongly recommends that a secondary means of braking be provided to stop a runaway load caused by block lift or any other condition..

Speed capability is affected by flow rate and mechanical loading of the rotating group, and therefore, is a function of hydraulic unit displacement (swashplate angle). The graph *Speed Ratings for Piston Pumps and Motors* shows the typical speed rating versus displacement relationship for axial piston and bent axis

Pressure and Speed Limits

hydraulic units. In general, speed capability increases as displacement decreases, up to a peak value at intermediate displacements.

The **Rated Continuous Speed Limit** is the highest speed recommended at full power conditions. This rating defines the highest speed at which normal life can be expected. Continuous full power operation applies at the conditions defined by continuous pressure.

Operation above the rated speed limit is expected at reduced power, as engine speed increase (for pumps) and reduced volumetric loss (for motors) will produce increased motor/vehicle speed at low, or negative, power. Some reduction of life would be expected at full power when operating above the rated speed limit. For normal systems, working power level, and resulting system pressures will be low when applied properly, and time above the rated speed limit will be low.

The rated speed limit varies with swashplate angle up to the reduced angle continuous maximum value. The upward sloping speed characteristic is derived from the kinematic design parameters of axial piston pumps and motors. They define the conditions at which normal operation is expected, based on sufficient charge pressure. Normally, pumps are not applied at reduced angle speed ratings because they frequently must operate at the highest speed and maximum angle simultaneously.

The **maximum speed rating** is the highest operating speed recommended and cannot be exceeded without severe reduction of unit life, or without risking premature failure and loss of drive. Operating conditions between continuous and maximum speed limits should be restricted to less than full power and to limited periods of time. For most drive systems, maximum motor speed occurs during downhill braking or negative power conditions.

The maximum speed rating varies with swashplate angle up to the reduced angle maximum value. At this speed, conditions of marginal lubrication may exist when system pressure is low. Also, rotational forces are high and may cause abnormal mechanical loading and stress on the rotating group.

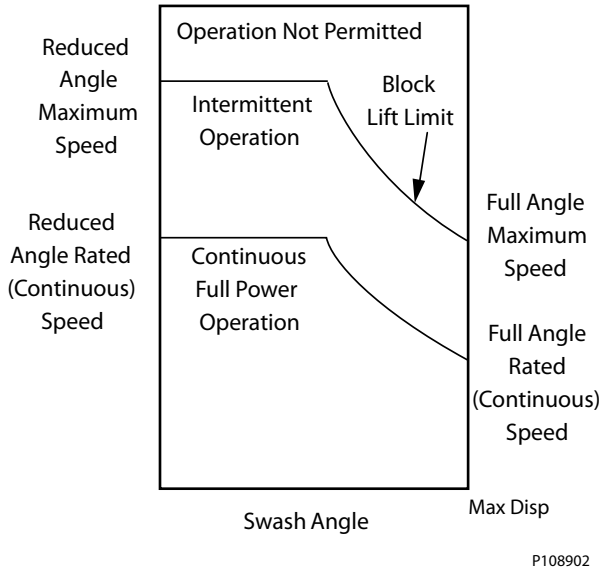
For all axial and bent axis units, exceeding the maximum speed limit is a safety concern when the transmission is providing retarding power. At high swashplate angles, the maximum speed is limited by block lift which causes a loss of drive power. This could allow dangerously high, and uncontrollable, output speed. Operation above the maximum speed limit must be avoided under all modes of operation including engine speed-up and reduced volumetric loss during downhill braking.

Warning

Danfoss Power Solutions strongly recommends that a secondary means of braking be provided to stop a runaway load caused by block lift or any other condition.

Pressure and Speed Limits

Speed Ratings for Piston Pumps and Motors



The highest permissible rated (continuous) and maximum speed limits for Danfoss Power Solutions closed circuit piston pumps and motors are provided in the table *Figure : Speed Ratings for Closed Circuit Pumps and Motors* on page 13. Variable motors have speed ratings for both “maximum” and “minimum” displacements. The “minimum” displacement speed limit applies at approximately half of full displacement, but this varies with product family and frame size. Minimum displacement speed ratings are not provided for pumps, because pumps are typically expected to operate at fixed input speeds at full displacement.

Pressure and Speed Limits

Speed Ratings for Closed Circuit Pumps and Motors

	Pumps		Fixed Motors		Variable Motors			
					Full Disp.		Reduced Disp.	
Product	Rated	Max	Rated	Max	Rated	Max	Rated	Max
Series 90								
042	4200	4600	4200	4600	---	---	---	---
055	3900	4250	3900	4250	3900	4250	4600	5100
075	3600	3950	3600	3950	3600	3950	4250	4700
100	3300	3650	3300	3650	---	---	---	---
130	3100	3400	3100	3400	---	---	---	---
180	2600	2850	---	---	---	---	---	---
250	2300	2500	---	---	---	---	---	---
Series 20								
21	3500	3800	3500	3800	3500	3800	4400	5250
22	3200	3900	3200	3900	3200	3900	4000	4850
23	2900	3350	2900	3350	2900	3350	3700	4550
24	2700	3100	2700	3100	2700	3100	3450	4100
25	2400	2950	2400	2950	2400	2950	3100	3750
26	2100	2500	2100	2500	2100	2500	2800	3400
27	1900	2050	1900	2050	1900	2050	2450	2950
Series 51								
060	---	---	---	---	3600	4400	5600	7000
080	---	---	---	---	3100	4000	5000	6250
110	---	---	---	---	2800	3600	4500	5600
160	---	---	---	---	2500	3200	4000	5000
250	---	---	---	---	2200	2700	3400	4250
Series 42								
28	3400	3900	---	---	---	---	---	---
41	3400	3900*	---	---	---	---	---	---
Series 40								
M25	4000	5000	4000	5000	4000	5000		5700
M35	3600	4500	3600	4500	3600	4500		5300
M44	3300	4100	3300	4100	3300	4100		4850
M46	4000	4100	3600	3600	4000	4100		5000
Series 15								
15	4000	4200	4000	4200	---	---	---	---
Series 70								
BDP-10L	3600	3600	---	---	---	---	---	---
BDP-21L	3600	3600	---	---	---	---	---	---

* Any operation above rated speed requires application approval

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Pressure and Speed Limits

The following equations are provided to calculate motor speed limits for intermediated displacements. Although the calculated values will exceed the tabulated values at lower swashplate angles, speed limits from the table should never be exceeded. The equations are valid for “maximum” as well as “rated” speed limits and are provided for convenience of calculation.

Axial Piston Motors

$NV = NM (TM / TV)^{1/2}$

or

$NV = NM (DM / DV)^{1/2}$

where:

NV = Motor speed limit at reduced angle (RPM)

NM= Motor speed limit at maximum swashplate angle from Table 2 (RPM)

TV = Tangent of reduced swashplate angle

TM = Tangent of maximum swashplate angle

- Tan 15° = 0.268 (Series 15 and Series 40-M25)
- Tan 16° = 0.287 (Series 40-M35 and Series 40-M44)
- Tan 17° = 0.306 (Series 90 and Series 40-M46)
- Tan 18° = 0.325 (Series-20, Series 42, and H1P)

DM= Motor displacement at maximum angle (cc/rev or in^3/rev)

DV = Motor displacement at reduced angle (cc/rev or in^3/rev)

Bent Axis Motors

NV = NM (SM / SV)

or

NV = NM (DM / DV)

where:

NV = Motor speed limit at reduced angle (RPM)

NM= Motor speed limit at maximum angle from Table 2 (RPM)

SV = Sine of reduced angle

SM = Sine of maximum angle, Sin 32° = 0.530 (Series 51 and H1B)

DM= Motor displacement at maximum angle (cc/rev or in^3/rev)

DV = Motor displacement at reduced angle (cc/rev or in^3/rev)

Motor Performance at Low Speed

Motor Performance at Low Speed

Some applications require controlled motor operation at low speeds. Generally, speeds less than 5% of rated are considered “slow” for axial piston motors. For bent axis motors, speeds less than 2-3% of rated are considered “slow”. Like high speed applications, a common requirement for low speed operation is continuous smooth output shaft rotation. However, axial piston type motors have inherent limitations which may provide unacceptable performance at low speeds.

The main limitation is related to the interaction of the pistons as they transition from high to low pressure across the valve plate. As each piston makes the transition, there is a variation in the motor’s total displacement. Even if system pressure remains constant, the variation in displacement results in a “torque ripple” at the motor output shaft, which is generally described as shaft cogging. As system pressure increases, the speed at which cogging occurs also increases. Other system parameters affecting this inherent phenomenon include inertia of the load, load fluctuations, and bulk modulus of the system (line diameter and length, line type, and fluid compressibility).

Another inherent limitation to low speed operation, common to all types of driveline components, is simply mechanical friction. For piston motors, this is largely a function of the design of the unit and the manufacturing tolerances of the rotating group parts.

As a rule of thumb, most Danfoss Power Solutions axial piston motors will operate smoothly down to ~150 rpm. Some axial piston motors may run smoothly at less than 150 rpm, but performance must be verified for the specific application. Danfoss Power Solutions does not recommend that any axial piston motor be size to operate continuously at speeds less than 100 rpm.

The H1B and Series 51 variable displacement bent axis motors have demonstrated smooth performance at speeds down to 50 rpm in most applications, but acceptability must be judged based on specific application requirements.

Because the limitation to low speed operation is inherent to all piston type motors, Danfoss Power Solutions cannot offer “special hardware” to improve low motor speed performance. Low speed applications should be critiqued carefully at the design phase and thoroughly tested for acceptable performance at the prototype phase.

Bypass Valve Speed Limitations

Bypass Valve Speed Limitations

Most Danfoss Power Solutions axial piston pumps have a bypass valve which allows system fluid to be cross-ported between the main “A” and “B” ports when pump input shaft rotation is not possible. This valve requires manual engagement and disengagement via an external device on the pump, such as an adjustment screw or plunger. The intent of this valve is to allow a vehicle using a closed loop propel circuit to be moved a short distance at slow speeds. Typically, this translates to loading a vehicle onto a trailer, getting a malfunctioning machine away from the work area, or out of a traffic lane.

Cross-port bypassing is required to prevent hydraulic lockup as the motor becomes a flow source while the vehicle is being moved (assuming the motor remains coupled to the ground via tires or tracks). The bypass valve should not be used or referred to as a “towing valve” which implies high speed and long duration usage.

Vehicle speed limitations associated with cross-port bypassing are a function of flow rate and duration of time. Exceeding one, or both, of these criteria can lead to failure of the transmission. Flow rate is critical because the bypass valve may become saturated due to its pressure rise rate creating excessive heat. The temperature rise of the fluid as it passes thru the bypass valve is proportional to the square of the flow rate. (i.e., if you double the flow rate, you will increase the temperature rise rate by 4-times).

Duration is critical because there will be no charge fluid to makeup leakage in the circuit, and the system lines will become evacuated. Because the flow rate being bypassed is a function of the motor displacement, final drive ratio, and rolling radius; it is impossible to publish strict speed ratings for the bypass mode. Also, the circuit design will affect how quickly system fluid is lost to leakage. However, the following rule of thumb for flow rate and duration has been used successfully to quantify the limitations during a bypass mode.

Do not operate the transmission in bypass mode at more than 10~15% of rated motor speed and do not exceed a duration of 3~5 minutes, depending on the product. Refer to the product specific Technical Information Manuals for details and limits.

During normal transmission operation, the bypass valve must remain fully closed. Partial opening of the valve will cause sluggish performance and will create excessive heat in the system.

The bypass function is standard on all Series 90 pumps, Series 40 pumps and H1P pumps. It is an optional feature on Series 42 pumps and Series 15 pumps. Please refer to the technical information bulletins for the specific pump families for details on the use of the bypass function.

The bypass function is not available for Tandem H1 Pumps.

Inlet Vacuum Limits

Inlet Vacuum Limits

Flow enters the charge pump through the charge inlet. Unless this inlet is pressurized, a vacuum will exist due to the resistance to flow through the inlet line leading to the inlet port of the charge pump.

The degree of inlet vacuum affects the ability of the charge pump to fill properly. Starving the charge pump can cause the inlet fluid to reach its “air release” pressure level which results in cavitation. **Cavitation** is the formation and collapse of vapor and/or air bubbles in the fluid. When the bubbles collapse, they pit or erode the metal surfaces on which they are in contact resulting in reduced Volumetric Efficiency. Charge pumps are usually less susceptible to erosion damage because of the reduced pressure in the charge cavity. Typically, damage occurs at the valve plate, cylinder block and piston slippers due to the high pressures in the system circuit. Volumetric losses can become so extreme that the charge pump will be unable to meet the demands of the system. Loss of charge pressure can cause serious degradation of system performance as well as component failure.

As a general recommendation, continuous inlet vacuum should not exceed 6 inches of Hg. for any gerotor style charge pump during normal operation. Intermittent limits for cold start situations are more a function of the capability of the charge pump. Danfoss Power Solutions medium and high power pump families are generally capable of achieving upwards of 24 inches Hg. for short duration events. Series 15 products are capable of vacuums on the order of 20 inches Hg. for short duration events. **Refer to the product specific Technical Information Manuals for details and limits.**

Normal operating inlet vacuum in the range of 6 to 10 inches Hg is generally associated with a suction filter in need of replacement.

Inlet suction specifications are based on the use of mineral base fluids at sea level. Fluids with a higher specific gravity than mineral base fluids, such as fire resistant or synthetic fluids; are more resistant to flow and are more likely to cause cavitation. Some of these fluids may require inlet vacuum limits on the order of 3 inches Hg for continuous operation. Please refer to *Hydraulic Fluids and Lubricants* 520L0463, *Experience with Biodegradable Hydraulic Fluids* 520L0465, and *Design Guideline for Hydraulic Fluid Cleanliness* 520L0467 for more details on the subject.

Operation at altitudes above sea level also changes charge pump inlet vacuum limits. As a rule of thumb, for every one thousand feet above sea level, the inlet vacuum limit should be reduced by 1 inch Hg. Pressurized reservoirs can be used to compensate for lost atmospheric pressure in closed circuit hydraulic systems. **Refer to the product specific Technical Information Manuals for details and specific inlet vacuum limits.**

Dynamic Braking

Dynamic Braking

By definition, a closed circuit hydrostatic transmission is capable of transmitting power through both the "A" and "B" sides of the power loop. In a typical mobile vehicle system, there are two modes of power transmission: propel and braking. **Propel** implies that the power is transmitted from the engine (or prime mover), to the pump through to the motor, through the final drive gears and wheels, and finally to the ground. Power transmitted during the braking mode is opposite from the propel mode. **Braking** implies that the power from vehicle inertia is transmitted from the ground, through the wheels and final drive gears, to the motor, to the pump and back into the engine. All closed circuit hydrostatic systems with duty cycles that include accelerations and decelerations for a load are subjected to both the propel and braking modes of operation. On mobile equipment applications, the braking mode will typically be encountered when the vehicle is commanded to decelerate quickly or when it travels down an incline of sufficient grade to overcome the rolling resistance of the vehicle. These modes of operation are often referred to as "dynamic braking" and "downhill braking", respectively.

It is important to note that, although dynamic braking is an inherent feature of closed circuit hydrostatic systems, the pump and motor only act to transmit the power; and that the braking power of the vehicle is limited to the braking power of the engine. The ability of the vehicle to dynamically brake is only a function of the power absorption capability of the engine while remaining within the maximum speed limits of the engine, pump, and motor. Most engines have the capacity to absorb 10~15% of their rated power. Braking power transmission above the absorption capability of the engine may cause the engine to over-speed and risk premature failure. Consult the engine supplier for the braking power absorption capability.

Some high inertia applications may require the addition of a special dynamic braking valve to prevent engine over-speed. The **Integrated Speed Limitation Valve** improves vehicle deceleration while protecting the engine from failure due to over-speed. Please consult *H1 Axial Piston Pump 115/130 147/165 ISL Integrated Speed Limitation* 11053026 for additional information that applies to the H1 Series of pumps. It is recommended that a test program to verify safe, and controllable, performance during all modes of dynamic and downhill braking is completed.

Although the closed circuit hydrostatic system is capable of transmitting dynamic braking power back to the engine, the loss of hydrostatic driveline power in any mode of operation may cause the loss of hydrostatic braking capability. Therefore, a braking system which is independent of the hydrostatic transmission must be provided which is adequate to stop and hold the vehicle should the condition develop.

As implied above, the roles of the pump and motor are reversed during dynamic braking. The motor become a flow source, while the pump receives the flow and offers some resistance to it based on its displacement and engine frictional load. The resistance, or course, determines the system pressure in the braking loop. The pump transmits hydraulic power (flow at pressure) back to the engine where it is dissipated as frictional heat.

During a downhill operating mode, the vehicle will experience an increase in travel speed for two reasons. The first reason involves volumetric leakage of the pump and motor. In the propel mode (as defined above) normal pump and motor leakage reduce the output speed (i.e., vehicle speed) relative to the input speed. However, during downhill braking, when the pump and motor roles are reversed, the leakage acts to increase the vehicle speed.

The second, and most significant issue, involves the internal forces acting on the swashplate as a result of braking pressure in the power loop. In the propel mode, internal swashplate forces act to reduce the displacement of the pump. The magnitude of the resultant force or moment is a function of system pressure, input speed, and servo spring force, swashplate angle and valve plate design. In the braking mode, however, the swashplate forces act to increase the displacement of the pump. If the pump displacement increases, the motor can rotate faster since more flow is accepted by the pump, with the net result being increased vehicle speed.

Dynamic Braking

Displacement Control:

The mechanical feedback mechanism in the displacement control ultimately limits the amount of pump swashplate angle increase, which is called over-travel. However, the pump displacement control, like most controls, has a dead-band within which no error signal will be received from the feedback mechanism. Thus, the control will regulate swashplate position only after the swashplate has traveled outside of the dead-band. In other words, the displacement control limits the amount of swashplate over-travel, but cannot eliminate it completely. While the pump is operating within the dead-band during a downhill condition, the resulting effect on vehicle performance is generally considered unfavorable.

There are two ways to improve the downhill speedup characteristics of a pump: reduce the deadband of the displacement control or increase the internal swashplate forces. Because downhill speedup tends to be more pronounced in larger, heavier vehicles, Danfoss Power Solutions has addressed these issues with the Series 90 and H1 pump families.

Danfoss Power Solutions has also created a special set of hardware identified as a “Downhill Kit” that improves the internal forces during the downhill condition for the Series 90 pump family. The “Downhill Kit” includes nested servo springs (one spring nested inside another) and a special valve plate. The “Downhill Kit” improves the ability of a vehicle to minimize downhill speedup, but does not eliminate the potential for speedup on severe grades. This situation can only be considered during the design phase of the vehicle powertrain.

Because the downhill kit increases the neutral return forces on the swashplate, the actual charge pressure must be at least 24 bar (350 psi) to allow the pump to reach full displacement.

Because it is difficult to quantify all of the parameters associated with downhill speedup, it is difficult for the designer to predict whether to be concerned for a given application. In general, however, applications involving large inertial loads on off-highway grades should be critiqued carefully for downhill performance.

Case Drain Pressure Limits

Case Drain Pressure Limits

Both steady state and transient case pressure limits can have an impact on the success of the application of the hydrostatic pump and/or motor in the system. It is rare that case pressure will affect the performance of the system in a measurable way; but, it can have an impact on the success of the shaft seals and various seals/gaskets to prevent external leakage.

Case pressure may be above, or below, ambient pressures; depending on where the internal case pressure is referenced. For example, case pressure above, or below, ambient atmospheric pressure may impact the seals/gaskets at the various joints between housings and end caps.

Likewise, case pressure above, or below, the pressure in the driveline coupling cavity can have an impact on the shaft seals' performance. In general, individual shaft seals are designed to seal against an internal pressure that is greater than the external pressure. If the external pressure is greater than the internal pressure, then the fluid in the external cavity, air, or other forms of contaminant can enter the case cavity.

Some Danfoss Power Solutions products have minimum case pressure limits to ensure that all of the internal components are lubricated over the entire operating spectrum of the system. In other products, only the maximum case pressure limits are considered.

Please consult the specific product Technical Information Bulletins for the maximum, and minimum, case pressure limits for all Danfoss Power Solutions products in the system.

In some applications, particularly those with small diameter and/or long case drain hoses; case pressure can be influenced significantly by both viscosity of the fluid and by rapid changes in flow velocity. Under cold start conditions, when the viscosity is highest; the pressure in the case cavity can be reduced by reducing the velocity of the fluid in the hose and/or the length of the hose.

Short duration, transient, pressures should also be considered by the system designer. If there is a sudden acceleration of the fluid in the case drain line, for any reason; then there will be a transient pressure change inside the case cavity of the pump, or motor. If there is a positive acceleration of the fluid in the case drain line, there will be a short duration, transient pressure rise above the normal operating pressure. These pressure pulses can have the effect of reducing the life of the shaft seal. If there is a negative acceleration of the fluid in the case drain line, there will be a short duration, transient pressure drop below the normal operating pressure. These pressure pulses can have the effect of drawing contaminant (fluid, air, water, or abrasive particles) into the case cavity across the sealing interface of the shaft seal.

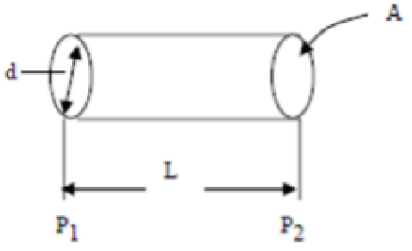
For an incompressible fluid, the peak value of the pressure required to accelerate the fluid in a circular passage if related to the mass density of the fluid, the length of the line, the diameter of the line and the fluid's acceleration. Reducing the line length will reduce the transient pressure linearly; increasing the diameter of the line will reduce the transient pressure by the square of the ratio of diameters considered.

$$F = \Delta PA = ma = \rho QV$$

$$\Delta PA = \rho QV = \rho \frac{LA}{t} V$$

$$\Delta PA = \rho \frac{LA}{t} \frac{Q}{A}$$

$$\Delta P = \rho \frac{LQ}{tA}$$



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Frequently, the line length is constrained by the location of the various components in the vehicle; but, it may be relatively easy to change the line diameter to keep case pressure within the limits defined in the product technical information manuals.

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