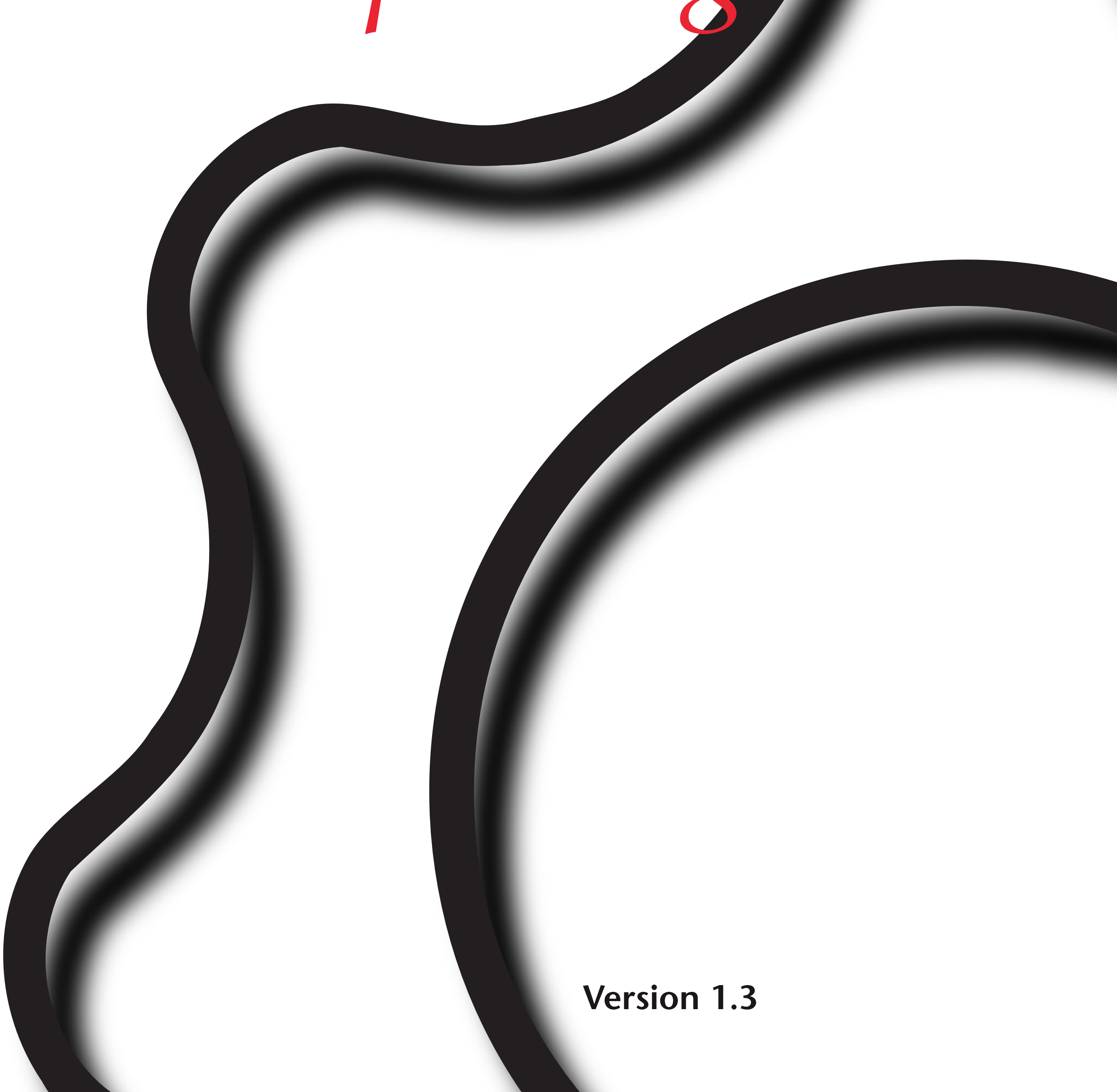
Nichols Portland

Gerotor Selection *and Pump Design*



Version 1.3

Gerotor Selection and Pump Design will guide you through the initial stages of your pump design. It will help you determine your application requirements, and decide whether a standard or custom gerotor best meets your needs.

It also reviews the most important considerations in designing a pump, including porting, construction, clearances and materials. Finally, pump performance is discussed in terms of efficiency, cavitation speed and noise.

However, as each application is unique, questions and issues will arise. When that occurs, please contact one of our Product Engineers directly.

Gerotor Selection and Pump Design, Version 1.3

Introduction

Pump Design

Nichols Portland

Nichols Portland is recognized as the world leader in designing and manufacturing precision gerotor elements. Whatever the application, we have available or can develop gerotors that provide your equipment with built-in cost and performance advantages along with increased design flexibility.

Using computer modeling and years of testing experience, Nichols Portland engineering teams can determine design and performance parameters - not only for the gerotor, but for the system surrounding it. We work in partnership with our customers, respecting confidentiality, while providing every level of technical service from gerotor selection to validation of system design.

Nichols Portland engineering services are as important as the gerotors we manufacture.

Introduction

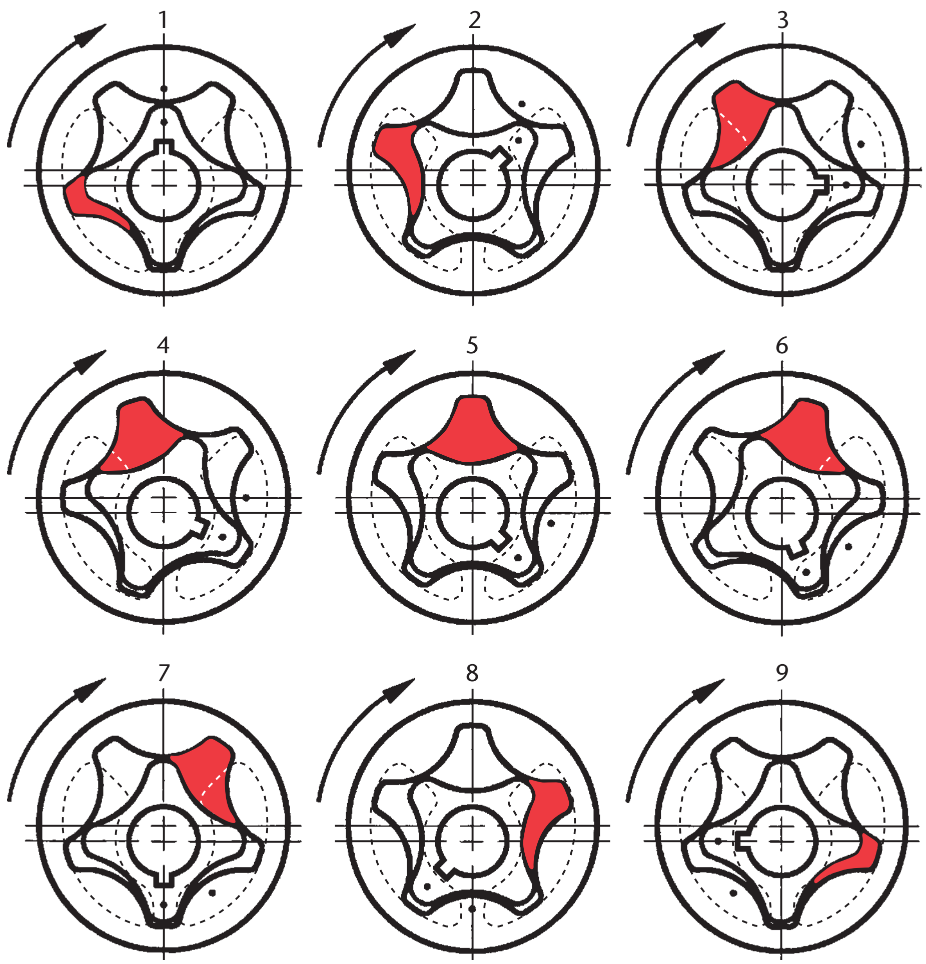


Figure 1

The Gerotor Principle

The gerotor is a positive displacement pumping unit consisting of just two elements: an inner rotor and an outer rotor. The outer rotor has one more tooth than the inner rotor and has its centerline positioned at a fixed eccentricity from the centerline of the inner rotor and shaft.

Although gerotors come in a variety of geometric configurations, materials, and sizes, all gerotor sets share the basic principle of having conjugately-generated tooth profiles which provide continuous fluid-tight sealing during operation.

As the rotors rotate about their respective axes, fluid enters the enlarging chamber to a maximum volume. As rotation continues, chamber volume decreases, forcing fluid out of the chamber. The process occurs constantly for each chamber, providing a smooth pumping action.

The Gerotor Advantage

The advantage of the gerotor lies in its simplicity, versatility, and performance. A gerotor pump requires the least amount of machining and fewest components of any type of pump. For example, there are no expensive crescents to machine into the pump pocket, and compared to spur gears, the gerotor can eliminate a shaft and two bearings. A gerotor pump can be used in most applications where a gear, vane, or crescent pump operates up to approximately 1200 psi (8300 kPa).

The versatility of the gerotor may be its greatest advantage. Gerotors can be mounted directly on an existing shaft or can be driven via the outer rotor. Gerotors can handle any flowing substance from air to hot melt glue. A single gerotor set can even accommodate multiple flow streams (inlet or outlet) operating at different pressures.

A gerotor's size and geometry can be optimized for performance. The outside diameter, inside diameter, and length can be adjusted to create the smallest possible overall size, saving weight and minimizing power loss. Gerotors have a long service life because the relative velocity between the inner and outer element is very low. For example, when a gerotor assembly consisting of an 8-tooth inner rotor and a 9-tooth outer rotor is operating at 1800 rpm, the relative speed between the inner and outer rotors is only 200 rpm.

Gerotor Identification Code

The Nichols numeric code system identifies gerotors by the number of teeth on the inner rotor and the gerotor displacement per unit thickness.

In English units, the last three digits of the code give the gerotor unit displacement (Du) in hundredths of a cubic inch per shaft revolution per inch of thickness (in^3/rev/in). To determine the gerotor unit displacement, simply take the last three digits of the code and divide by 100 to get in^3/rev/in. The first one or two digits of the code (the leading zero is typically left off) indicate the number of teeth on the inner rotor. For example, a 6170 has a unit displacement of 1.70 in^3/rev/in. and 6 teeth on the inner rotor. The 10010 gerotor has a unit displacement of 0.10 in^3/rev/in and 10 teeth on the inner rotor.

To convert the gerotor unit displacement (Du) from English units (in^3/rev/in) to Metric units (cm^3/rev/mm), the formula is as follows:

Using gerotor type 6170 (1.70 in^3/rev/in) as an example:

$$\frac{\mathbf{1.70\ in^3}}{\mathbf{inch}} \times \frac{\mathbf{16.387\ cm^3}}{\mathbf{1\ in^3}} \times \frac{\mathbf{1\ inch}}{\mathbf{25.4\ mm}} = \mathbf{1.10\ cm^3/rev/mm}$$

This conversion can be greatly simplified by multiplying the English units by .645.

Pump Design

Step 1: Determining application requirements

The most important piece of information needed to size a pump correctly is the relationship of pump flow to pump speed. There are three methods for determining this relationship:

A. Comparison to similar applications

This method is useful when modifying a design or revising the size of an existing design. It is important to compare the size and number of flow paths as well as leakage and any changes in material that may cause a thermal effect on the clearances.

B. Analysis of the system

This is often the only method available on a new design, and requires a thorough analysis of all design (and potential) flow paths, as well as the effects of tolerances, speed, pressure, and temperature. As with most analytical models, comparing predictions with measured values will increase the confidence in a new model.

C. Measurement of the system requirements

This is the most direct method, but it requires hardware to measure, and a knowledge of the clearances built into that hardware. For optimal accuracy, it is best to measure more than one system, preferably with one at maximum clearances. Nichols Portland has a portable test unit for measuring the flow demand in engine, transmission, and other lubrication systems. The test unit can be used in conjunction with a SuperFlow SF-901 engine dynamometer at Nichols Portland or it can be taken to a customer's facility for use there.

In many applications where the speed, pressure, and temperature vary, the pump will need to be sized based on the worst case operating conditions. To establish minimum pump flow, the worst case is typically a combination of low speed, high pressure, and high temperature. For maximum pump input power, the worst case will be some combination of high speed, high pressure, and low temperature.

Once the flow at a given speed has been determined, the gerotor's theoretical displacement (Dt) can be calculated.

The required (or actual) pump displacement (Dr) will depend on the volumetric efficiency (Ev) of the pump.

Volumetric efficiency varies as a function of operating conditions and pump construction and may range from 50-100%. Please contact Nichols Portland Product Engineering for estimates based on your application.

To calculate theoretical displacement:

English units:

$$Dt = \frac{231 \times Q}{N}$$

where 231 converts gallons to cubic inches and

Dt = theoretical displacement in cubic inches per revolution (in³/rev)

Q = flow in gallons per minute (gpm)

N = pump speed in revolutions per minute (rpm)

Metric units:

$$Dt = \frac{1000 \times Q}{N}$$

where 1000 converts liters to cubic centimeters and

Dt = theoretical displacement in cubic centimeters per revolution (cc/rev)

Q = flow in liters per minute (lpm)

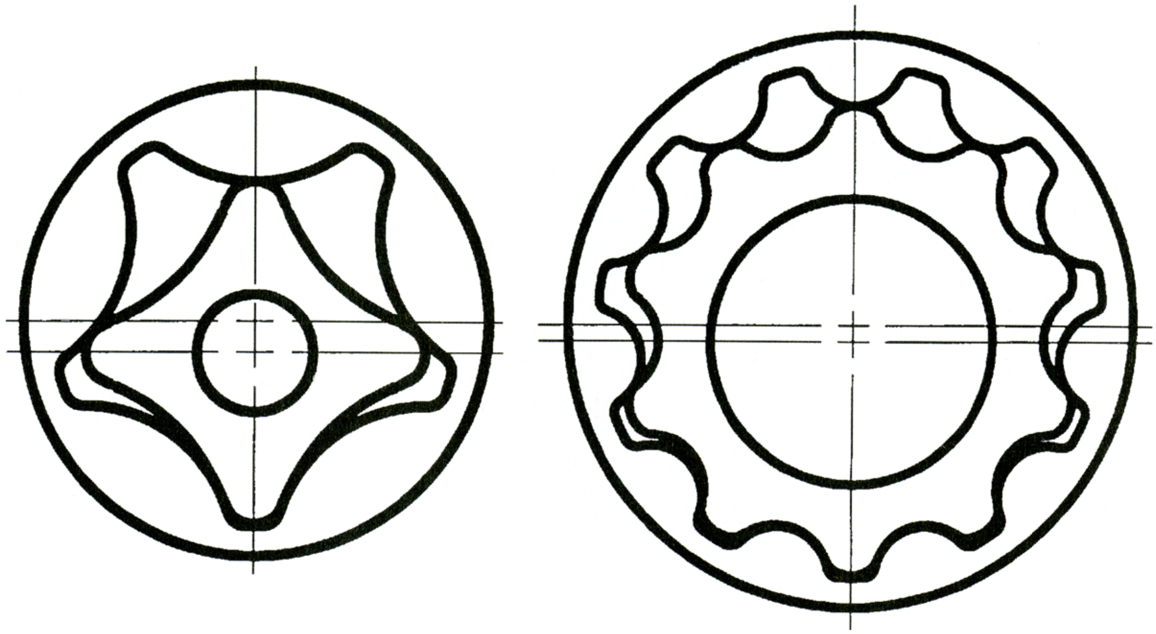
N = pump speed in revolutions per minute (rpm.)

To calculate required displacement:

$$Dr = \frac{Dt}{Ev}$$

Gerotors of Identical Theoretical Displacement

Figure 2



4 Tooth Advantage:

- Smaller package
- Less power loss
- Higher operating speed

10 Tooth Advantage:

- Accommodates larger shaft
- Less flow ripple

Step 2: Dimensional constraints

Determine the dimensional constraints before choosing a gerotor for your application. These constraints are application-dependent and include the drive, eccentricity of the gerotor set, the diameter of the outer rotor, and the thickness of the gerotor set.

As illustrated in Figure 2, the number of teeth in a gerotor set directly affects size and performance.

Step 3: Standard or custom gerotor

Nichols Portland has tooling for a variety of gerotors referred to as Nichols Portland standard gerotors, but a custom gerotor can be designed specifically for your application. A third option is to use a standard gerotor with a custom drive and/or O.D.

If the production volume for your application does not justify custom tooling, consider using a standard gerotor. If your application is for high volume production, or other application constraints preclude the use of a standard gerotor, then a custom gerotor may be the most cost effective solution. Custom design is an area where we specialize. Nichols Portland has designed and manufactured hundreds of different gerotors to precisely match specific applications.

If you have any questions about whether to choose a standard or custom gerotor, please contact Nichols Portland Product Engineering.

Step 4: Selecting a standard gerotor

A standard gerotor can be selected using the following criteria.

A. Gerotor width

Calculate the actual width (W) of the gerotor using

English units:

$$W = \frac{Dr}{Du} \text{ Inches}$$

Dr = required displacement of the gerotor

Du = gerotor unit displacement

For metric conversion, see page 5.

B. Inlet port constraints

In most applications, pump inlet velocity through a line, fitting, or port should be below 6 feet per second (2 meters per second). (This assumes petroleum-based fluids with a viscosity of 5 to 100 cP and a properly sized inlet at standard atmospheric pressure. Other fluids may require different recommendations.)

To calculate the inlet velocity per port use

English units:

$$V = \frac{231 \times Q}{60 \times 12 \times Ap}$$

V = inlet velocity per port in ft/sec

Q = flow in gpm

Ap = port area in square inches

(see pages 22 and 23)

Metric units:

$$V = \frac{1000 \times Q}{60 \times Ap}$$

V = inlet velocity per port in m/sec

Q = flow in lpm

Ap = port area in square millimeters

(see pages 22 and 23)

Inlet velocities can be cut in half by using double fill (see Figure 3).

C. Speed constraints

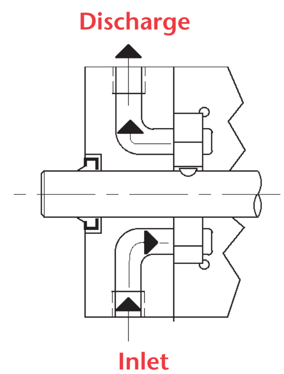
Each gerotor has a recommended maximum operating speed based on the onset of cavitation (see pages 22 and 23). Operating above this speed without corrective measures will degrade pump performance. For additional information, see the section on Cavitation.

D. Flow ripple is inherent in all positive displacement pumping devices and can contribute to system pressure ripple.

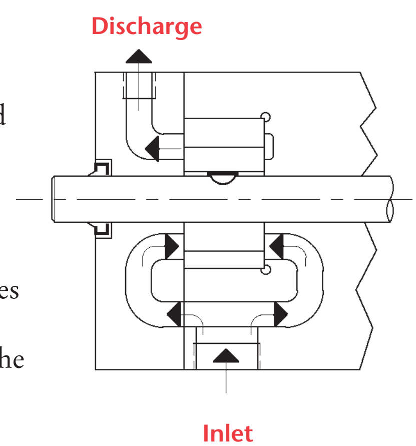
Theoretical flow ripple depends on the number of teeth and the specific geometry of the gerotor, but in general, more teeth means lower flow ripple. Also, a gerotor with an even number of inner teeth typically has a lower flow ripple than a gerotor with an odd toothed inner having one more tooth.

Figure 3

Single Fill



Double Fill



E. Power draw for a pump comes from two sources: hydraulic power and mechanical losses.

Hydraulic power is a function of flow and pressure. The mechanical losses in a pump are caused by viscous drag on the gerotor O.D. and faces, as well as bearing and seal drag. In applications where low power draw is critical, the gerotor with the smallest O.D. will have the lowest power loss due to viscous drag.

Step 5: Pump design

The basic elements to consider are: porting, pump construction, clearances, and materials.

A. Active Porting

There are several options for pump porting, depending on the application and manufacturing considerations.

The first step in designing a pump is to determine whether the pump needs to be single or double filled (see step 4B).

The next step is to determine which type of porting is best for your application. A summary of advantages and disadvantages is given below, and sketches of each type of porting are shown in Figure 4. Variations to each of these port designs is possible, based on the application. For more information, please contact Nichols Portland Product Engineering.

Type 1 Porting

The advantage of type 1 porting is simplicity of machining. A disadvantage is that the outside radius of the port reduces the face sealing for part of the gerotor by violating the major diameter of the outer rotor. This might be unacceptable in applications requiring high pressure and/or using low viscosity fluids.

Type 2 Porting

The advantages of type 2 porting include better filling and sealing than type 1 porting, but the disadvantage is the increased complexity to machine. If the porting is cast, or if the ports are put in by an NC machine, this is less of an issue.

Metered Porting

In applications where noise control is essential and rotation is unidirectional, metered porting can help. The metering groove can reduce the rate of pressure rise and the resulting noise and vibration. The metered port design is application specific, so please contact us.

Porting For Reversing Pumps

This specialized porting is required when using a reversing eccentric ring (see Figure 7) and the gerotor porting needs to be symmetrical from top to bottom.

Additional modifications to each of these ports is possible, based upon the application.

Once the porting has been chosen, further considerations include inlet line design, port depth and shadow porting.

Inlet Lines

The size and design of the inlet lines and fittings should be consistent with good pump design practice. To optimize the inlet flow, the length of the inlet line should be minimized, along with the number of bends and transitions. The inlet line area should be sized

Porting

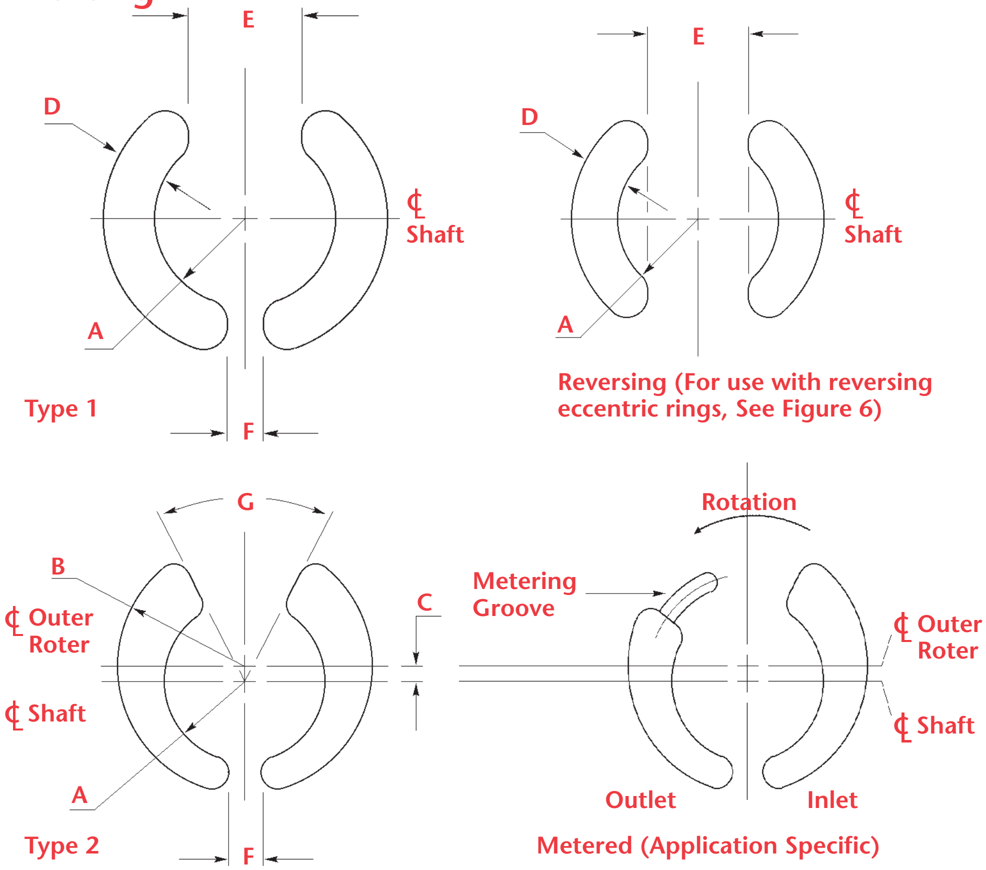


Figure 4

to meet the pump inlet velocity guidelines. Another critical consideration in the pump design is the location of the pump relative to the fluid sump. Ideally the pump inlet would be below the sump level, but if this is not feasible, the height of the pump above the sump should be minimal. Pump flow can enter or exit from either or both sides.

Port Depth

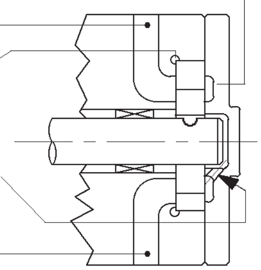
The depth of the active porting should be such that it does not restrict the pump flow, especially on the inlet side. In most single fill pumps, a port depth equal to the thickness of the gerotor is adequate. For a double fill pump, each port can be half of that depth.

Shadow porting

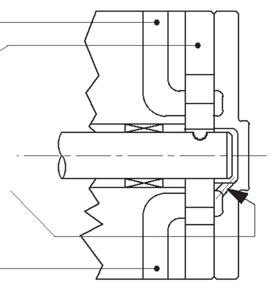
A shallow version of the active porting (on the opposite side of the gerotor), is recommended to help pressure balance the pump axially and reduce viscous losses. Shadow porting will have the same porting radii and widths as the active ports, but should be increased across sealing lands E, F and G to prevent any accidental crossport leakage that could result from angular misalignment. When metered porting is employed, a shadow outlet port does not usually require a metering groove. The depth of the shadow ports can be .02 to .10 inches (0.5 to 2.5 mm).

Figure 5

Typical Two Piece Pump Construction



Typical Three Piece Pump Construction



B. Pump Construction

The most common and cost effective method of construction employs using an eccentrically machined pocket for the gerotor and a cover plate (Figure 5).

In such a design, add an undercut to the bottom of the pump pocket, or specify chamfers on the O.D. of the gerotor to avoid any interference in that area.

In applications where a tight axial clearance is required, an eccentric ring or centerplate can be used. The centerplate establishes the eccentricity of the pump and must be precisely located relative to the shaft. The centerplate can be ground or lapped to thickness for better control of the axial clearance between the gerotor and the plate.

The centerplate material's coefficient of thermal expansion can also be closely matched to the gerotor, thereby ensuring constant axial clearance, regardless of temperature.

C. Reversing Pumps

In applications where the shaft rotation is bi-directional but the flow direction is fixed, there are two ways to ensure the gerotor pump will always pump in the same direction. One way is to use a reversing eccentric ring design (Figure 6). The reversing eccentric ring will reverse the gerotor position of eccentricity when the shaft rotation reverses, which maintains the direction of the flow. The spring and pin shown in Figure 6 ensure that there is enough friction between the outer rotor and the eccentric ring to reverse the ring on demand.

However, there are a number of disadvantages with this configuration. First, the addition of the eccentric ring adds another layer of clearance, which must be accommodated and can make the pump unsuitable for higher pressure applications. Second, if the assembly is fitted with a spring and friction pin, then over time, the latter can wear a groove in the cast iron eccentric ring, thereby lessening its effectiveness and to a lesser extent contamination the pumping medium. Last and most important is the fact that the reversing action is not repeatable within a set number of inner rotor revolutions. This variability is greatly affected by the clearances, inner rotor speed, eccentric ring OD configuration and fluid viscosity.

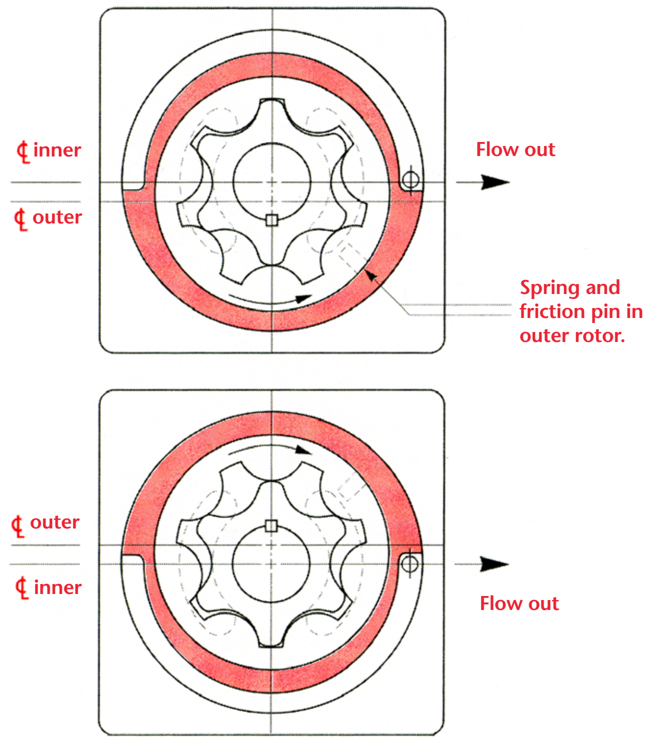
The second and preferred way to accommodate shaft reversals is to use four check valves in the circuit (Figure 7). The reversing eccentric ring does not add width to the gerotor package, but it does add to the packaging diameter, so there are some applications where the check valve system is preferred.

D. Clearances and Tolerances

The tolerances required in pump and gerotor designs are dependent upon each application. Figure 8 provides guidelines for some applications. The most important factors affecting pump performance are the pump clearances. The tolerance stackups for a pump, both radial and axial, can also affect pump performance. For recommendations on the clearances and tolerances for your application, please contact Nichols Portland Product Engineering.

Reversibilty Feature

Figure 6



Schematic: Reversing Pump With Check Valves

Figure 7

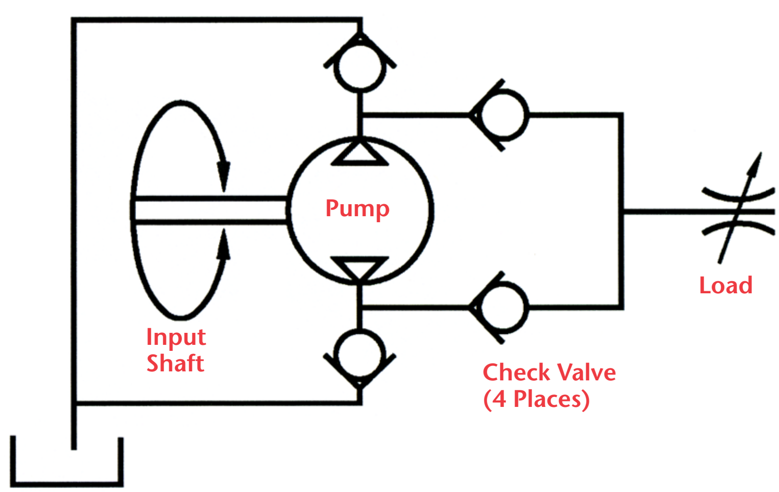


Figure 8

Operating Conditions And Clearances For Some Typical Applications

NOTE: Gerotor-to-housing clearances are appplication specific and dependent upon a variety of factors. The information provided below is to be used for comparison purposes only. Please contact Nichols Portland Product Engineering for more information about interface dimensions.

	English	Metric
Engine Lubrication Pump		
Pressures:	30 to 120 psi	(200 to 800 kPa)
Speeds:	400 to 16,000 rpm	(400 to 16,000 rpm)
Displacements:	0.4 to 4.0 in^3/rev	(6.5 to 65.6 cc/rev)
Temperature range:	-40 to 300°F	(-40 to 150°C)
Axial clearance:	.002 to .004 inches	(0.05 to 0.10 mm)
O.D. clearance:	.005 to .009 inches	(0.10 to 0.20 mm
Gerotor tip clearance:	.003 to .008 inches	(0.08 to 0.20 mm)
Eccentricity tolerance:	± .002 inches	(± .05 mm)
Automatic Transmission Pump		
Pressures:	60 to 300 psi	(400 to 2100 kPa)
Speeds:	500 to 7,000 rpm	(500 to 7,000 rpm)
Displacements:	1.0 to 2.5 in^3/rev	(16.5 to 41.0 cc/rev)
Temperature range:	-40 to 300°F	(-40 to 150°C)
Axial clearance:	.0008 to .0020 inches	(0.020 to 0.051 mm)
O.D. clearance:	.004 to .007 inches	(0.1 to 0.18 mm)
Gerotor tip clearance:	.006 inches max.	(0.15 mm max.)
Eccentricity tolerance:	± .0015 inches	(± .038 mm)
High Pressure Pump		
Pressures:	800 to 1200 psi	(5500 to 8300 kPa)
Speeds:	1800 to 3600 rpm	(1800 to 3600 rpm)
Displacements:	0.5 to 1.5 in^3/rev	(8.2 to 24.5 cc/rev)
Temperature range:	100 to 250°F	(40 to 120°C)
Axial clearance:	.0002 to .0012 inches	(0.005 to 0.030 mm)
O.D. clearance:	.003 to .005 inches	(0.08 to 0.13 mm)
Gerotor tip clearance:	.003 inches max.	(0.08 mm max.)
Eccentricity tolerance:	± .0008 inches	(± .020 mm)

E. Driving Methods

There are many ways to drive the inner rotor, as shown in Figure 9. The configuration for Nichols Portland standard gerotors is a round bore with keyways.

One advantage is that the gerotor can be designed with a large I.D. which accommodates thru-shaft applications. Figure 10 shows a comparison of a thru-shaft and shaft end constructed pump.

F. Other Important Considerations

One of the most important details of a pump design is to make sure there is good axial pressure balance on the gerotor to prevent unwanted thrust loads. Pressure can develop at the gerotor I.D. when leakage flow across the face of the gerotor is not adequately drained and a back pressure develops (see Figure 10). If this back pressure differs from one side of the gerotor to the other, a thrust load can develop that will force the gerotor against one face and may cause high torques or even a seizure.

Common Drive Configurations

Figure 9

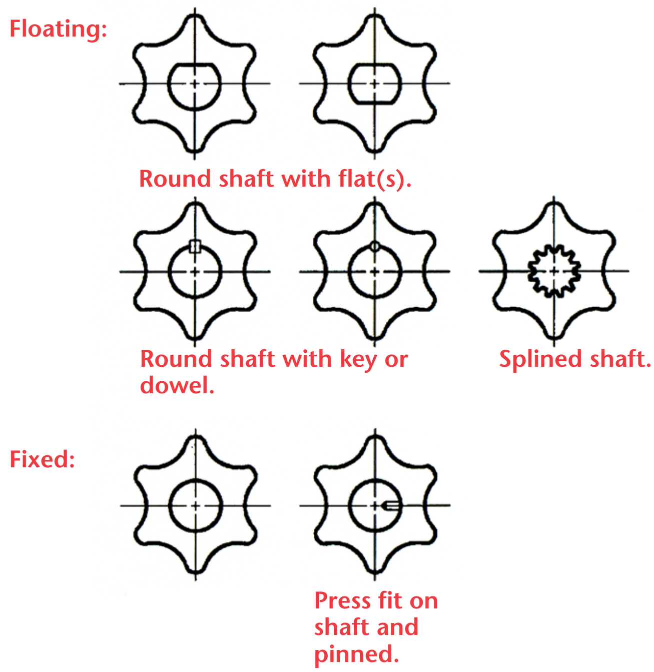
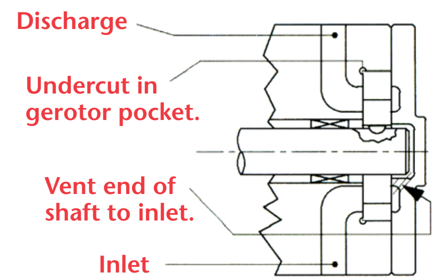
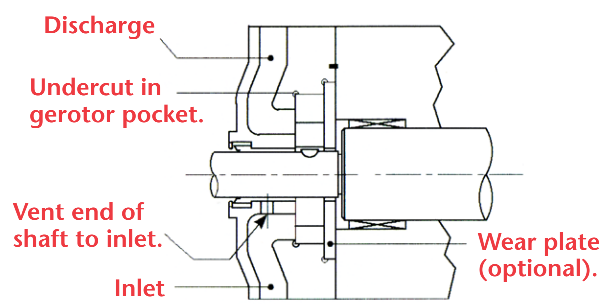


Figure 10

Typical Shaft End Installation



Typical Through-Shaft Installation



The gerotor is a positive displacement device. A properly designed gerotor pump operating at recommended conditions will have a rate of output flow that is proportional to its operating speed. Bearing in mind that pressure is created by a resistance to flow, there should be a means of preventing overpressurization of the system. Most pumping systems rely on a pressure relief valve or pressure regulating valve to accomplish this task.

If the valve is part of the pump and excess fluid is routed directly to the inlet port, two secondary benefits can be realized: first, the inlet circuit can be sized to accommodate the regulated output flow rather than the pump's total rated flow. (This can result in a significantly smaller inlet line.) Second, pressure at the inlet port is increased, thereby allowing the pump to operate at speeds greater than normally recommended. (Please note that this is an "application specific" enhancement. Contact Nichols Portland Product Engineering before incorporation this feature into your design.)

The <u>material</u> for the pump housing and cover should be chosen with the following properties in mind: wear, thermal expansion, and stiffness. A gerotor normally operates with hydrodynamic film supporting the O.D. and faces. However, there are conditions of high temperature, high pressure, low speed, and low viscosity when there could be metal-to-metal contact. Thus wear properties of the material are important. The thermal expansion and stiffness properties can increase the axial clearance of the pump during operation, which will lead to increased leakage at high temperatures and pressures. Also, in very cold conditions, differences in thermal expansion can reduce axial clearances to zero and cause pump seizure at startup.

<u>Extreme operating conditions</u> require special design approaches. In high-pressure pumps (typically over 500 psi or 3500 kPa), the shaft and the radial load on the inner rotor must be considered. Excessive shaft deflection can cause increased wear or mechanical lockup during operation. The radial load (Lr) on an inner rotor due to pressure is calculated by

Lr = p x W x (A + B + C)

*A = inside porting radius**

*B = outside porting radius (type 2 port)**

*C = eccentricity**

W = gerotor width

p = pump outlet pressure

<u>Housing deflections</u> also become critical. For example, end plate deflection on some pumps could be similar to the allowable axial clearance.

* *See Porting Dimension pages 21 & 22*

Step 6: Pump performance

A hydraulic pump converts mechanical power (Pm = speed x torque) into hydraulic power (Ph = flow x pressure).

A. Pump Overall Efficiency

The overall efficiency (Eovr) of the pump is a measure of the overall effectiveness of this conversion, and is defined as:

$$\text{Eovr} = \frac{\text{Power Out}}{\text{Power In}} = \frac{\text{Ph}}{\text{Pm}}$$

The input mechanical power (Pm) is calculated by

English units:

$$\text{Pm} = \frac{\text{T x N x2 x } \varpi}{\text{550 x 12 x 60}} \quad \text{Horsepower}$$

where 550 converts ft·lbf/sec to h.p.

T = input torque in pound/inches

N = input shaft speed in rpm

Metric units:

$$\text{Pm} = \frac{\text{2 x } \varpi \text{ x T x N}}{60} \quad \text{Watts}$$

T = input torque in Newton ·meters

N = input shaft speed in rpm

The output hydraulic power (Ph) is calculated by

English units:

$$\text{Ph} = \frac{\text{p x Q 231}}{\text{550 x 12 x 60}} \quad \text{Horsepower}$$

p = the pressure rise across the pump in psi

Q = the pump flow in gallons per minute

Metric units:

$$\text{Ph} = \frac{\text{p x Q}}{60} \quad \text{Watts}$$

p = pressure rise across the pump in kPa

Q = pump flow in liters per minute

Overall efficiency is also a product of volumetric (Ev) and mechanical (Em) efficiency.

B. Pump Volumetric Efficiency

The volumetric efficiency is a measure of the actual flow (Qa) produced by the pump at a given speed compared to the theoretical flow (Qt) at that speed.

Sources of flow loss are leakage and cavitation. Major internal pump leakages are across the face and through the tips. Both are proportional to the pressure drop and the cube of the axial clearance. For example, if the clearances are doubled, the leakage will increase by a factor of eight. Leakage is also inversely proportional to the length of the leak path and the fluid viscosity.

English units:

$$\text{Ev} = \frac{\text{Qa}}{\text{Qt}} = \frac{\text{Qa x 231}}{\text{D x N}} \text{ x } 100\%$$

D = pump displacement in in³/rev

N = pump speed in rpm

Qa = actual flow in gpm

Metric units:

$$\text{Ev} = \frac{\text{Q x 100}}{\text{D x N}} \text{ x } 100\%$$

D = pump displacement in cc/rev

N = pump speed in rpm

Qa = actual flow in lpm

C. Pump Mechanical Efficiency

Mechanical efficiency is a ratio of theoretical torque (Tt) the pump would require compared to the actual input torque (Ta). Actual input torque will always be greater than the theoretical torque.

Mechanical efficiency is a measure of mechanical losses in a pump. These losses are caused by viscous drag increased by large diameter, high speed, tight clearances, and high viscosity. The gerotor O.D. is the dominant factor, so in applications where it is important to keep mechanical losses to a minimum, try to choose a gerotor with the smallest O.D. The best design strikes a balance between manufacturability, volumetric efficiency, and mechanical efficiency.

Nichols Portland has developed computer models to help optimize your design.

English units:

$$\mathbf{Em} = \frac{\mathbf{Tt}}{\mathbf{Ta}} = \frac{\mathbf{D \times p}}{(\mathbf{2 \times \varpi \times Ta})}$$

(EM <= 1.0)

D = pump displacement in in³/rev

p = pressure rise across the pump in psi

Tt = theoretical torque in in·lbf

Ta = actual torque in in·lbf

Metric units:

$$\mathbf{Em} = \frac{\mathbf{D \times p}}{(\mathbf{2 \times \varpi \times 1000 \times Ta})} \times \mathbf{100\%}$$

D = pump displacement in in³/rev

p = pressure rise across the pump in psi

Tt = theoretical torque in Newton ·meters

Ta = actual torque in Newton ·meters

D. Cavitation

A pump fills by creating a partial vacuum at the inlet and by atmospheric pressure pushing the liquid through the inlet circuit, through the inlet port(s) and into the passing pumping chambers. In most cases, that means only 14.7 PSI (100 kPa) is available to overcome the resistance that is provided by gravity, the inlet circuit and the viscosity of the oil. What energy is left must accelerate the liquid so that it can enter the pumping chambers.

Cavitation occurs when the total amount of resistance that is provided by the inlet circuit reaches the vapor pressure of the liquid, causing it to boil.

In the simplest of applications, the inlet is flooded and there is no inlet circuit. In that case, the pump can operate until the speed of the pump reaches the maximum recommended operating speed of the particular gerotor that is being used. At that point, the pump will begin to cavitate.

In the vast majority of applications, there is an inlet circuit and the speed at which the pump will cavitate is determined by the amount of resistance that is provided by the inlet circuit and the operating speed of the pump.

Ideally, one would design the pump's inlet circuit by performing calculations to ensure that there is not an excessive amount of resistance, thereby avoiding cavitation. However, not everyone has access to the software that is required to perform those calculations. When that is the case, a simple calculation of the inlet velocity will provide assistance. A good target is 6 ft/sec (2 m/sec).

E. Pump Noise

Mechanical noise can be caused by cavitation, driveline windup, torsional vibration, structural resonances, and gear noise. This mechanical noise can be minimized with careful attention to pump clearances and gerotor geometry.

Hydraulic noise is usually a result of pressure ripple in the pump, which can cause a hydraulic line, valve, or casting wall to vibrate and radiate sound. Pump pressure ripple is affected by the gerotor flow ripple, pump porting geometry, system compliance, pump cavitation, and pump speed and pressure. The frequency (f) of the pressure ripple is usually based at the pump fundamental frequency (tooth pass frequency), which can be calculated by

$$f = \frac{N \times n}{60}$$

f = frequency in Hz

N = pump speed in rpm

n = number of teeth on the inner rotor (assuming a shaft driven inner)

Harmonic frequencies can also be present in the pressure ripple. In cavitation mode, the pump pressure ripple and noise become broad banded with a higher frequency content.

Summary

The gerotor offers tremendous design flexibility.

Gerotors and gerotor pumps continue to evolve, so whether your application is well established or in the design stage, Nichols Portland can help optimize your product's performance.

Glossary Of Terms And Abbreviations

Variable	Description	English Units	Metric Units
Ap	Port area	in^2	mm^2
A,B,C,D, E, F, G, H, J, L	Dimensions for porting	inch	mm
Dr	Required displacement	in^3/rev	cc/rev
Dt	Theoretical displacement	in^3/rev	cc/rev
Du	Unit displacement	in^3/rev per inch	cc/rev per mm
Em	Mechanical efficiency		
Eovr	Overall efficiency		
Ev	Volumetric efficiency		
f	Frequency	Hz	Hz
I.D.	Inside diameter	inch	mm
Lr	Radial load on gerotor	lbs	N
M	Metric designator		
N	Pump shaft speed	rpm	rpm
n	Number of inner teeth		
O.D.	Outside diameter	inch	mm
p	pressure	psi	kPa
Ph	Hydraulic power	Horsepower	Watts
Pm	Mechanical power	Horsepower	Watts
Q	Pump flow	gpm	lpm
Qa	Actual pump flow	gpm	lpm
Qt	Theoretical pump flow (Ev=100%)	gpm	lpm
T	Torque	lb-in	Newton·meters
Ta	Actual torque	lb-in	Newton·meters
Tt	Theoretical torque	lb-in	Newton·meters
V	Velocity of fluid	ft/sec	m/sec
W	Gerotor width (thickness)	inches	mm

Nichols Portland Standard Gerotors and Specifications

(English units table)

Gerotor Type*	Maximum Operating Speed**	Thickness Min.	Nominal Thickness Max.	Nominal O.D. Standard	Nominal O.D. Minimum Recommended***	Nominal I.D. Standard	Nominal I.D. Maximum Recommended***
	(rpm)	(inch)	(inch)	(inch)	(inch)	(inch)	(inch)
6010	21556	0.1000	0.3000	0.805	0.805	0.202	0.202
10010	17250	0.125	0.500	0.998	.873	0.313	0.375
6020****	16000	0.1250	0.5000	1.123	1.123	0.313	0.313
6022	14500	0.1250	0.5000	1.123	1.123	0.313	0.313
8030	11250	0.1250	1.0000	1.498	1.373	0.500	0.625
10060	7250	0.1880	1.0000	2.248	2.123	0.625	1.000
6063	9000	0.1880	0.8750	1.998	1.748	0.625	0.688
4065	10000	0.1880	1.0000	1.748	1.624	0.500	0.500
4086	7850	0.1880	1.0000	1.969	1.969	0.500	0.500
6095	7250	0.1880	1.2500	2.248	2.248	0.750	0.750
12131	4500	0.2500	1.0000	3.248	3.123	1.000	1.750
4158	6250	0.2500	1.3750	2.473	2.473	0.543	0.875
14162****	3750	0.2500	0.8750	3.998	3.748	1.625	2.125
6166	5000	0.2500	1.5000	3.012	3.012	1.000	1.125
6170	5500	0.2500	1.5000	2.998	2.873	1.000	1.125
4180	6000	0.3750	1.2500	2.627	2.627	0.563	0.875
10239	3655	0.3750	1.0000	3.950	3.950	1.750	2.000
6280	4250	0.3750	1.5000	3.998	3.750	1.250	1.500
8369****	3250	0.3750	3.0000	4.498	4.498	1.750	2.125
8384	3250	0.3750	1.5000	4.498	4.498	1.750	2.125
10397****	2800	0.3750	1.0000	5.000	5.000	2.000	2.400

Standard Gerotor Porting Information† (See Figure 4, page 11)

Gerotor Type	Nominal Porting Dimensions Radius A	Radius B	Ecc. C	Width D	Land E	Land F	Angle G	Port Area
	(inch)	(inch)	(inch)	(inch)	(inch)	(inch)	(degrees)	(in^2)
6010	0.1970	0.3100	0.0343	0.1030	0.1980	0.1090	52.4	0.068
10010	0.2730	0.3610	0.0260	0.0780	0.1740	0.0780	34.5	0.076
6020****	0.2580	0.4240	0.0520	0.1560	0.2680	0.1130	54.0	0.137
6022	0.2870	0.4530	0.0520	0.1560	0.2970	0.1340	54.0	0.147
8030	0.4170	0.5830	0.0520	0.1560	0.3300	0.1420	42.0	0.214
10060	0.6820	0.8880	0.0650	0.1950	0.4350	0.2200	34.5	0.43
6063	0.4700	0.7500	0.0900	0.2700	0.4950	0.1840	55.0	0.408
4065	0.3660	0.7060	0.1100	0.3300	0.5510	0.2540	79.5	0.366
4086	0.5060	0.8500	0.1110	0.3330	0.7380	0.4110	79.5	0.470
6095	0.5800	0.9200	0.1100	0.3300	0.6100	0.2350	55.0	0.607
12131	1.0700	1.3500	0.0900	0.2700	0.5610	0.2340	28.5	0.932
4158	0.5950	1.1000	0.1650	0.4950	0.8750	0.4690	77.5	0.855
14162****	1.3440	1.6240	0.0900	0.2700	0.6000	0.3110	24.5	1.16
6166	0.8750	1.2890	0.1346	0.4040	0.8800	0.5750	52.0	1.007
6170	0.7500	1.2100	0.1500	0.4500	0.7780	0.3200	54.0	1.073
4180	0.5750	1.1490	0.1880	0.5640	0.8700	0.3840	79.0	0.990
10239	1.2999	1.7116	0.1339	0.4017	0.8150	0.3402	33.6	1.652
6280	1.0000	1.5720	0.1880	0.5640	1.0500	0.4020	55.0	1.745
8369****	1.3750	1.9480	0.1875	0.5640	1.1060	0.3350	43.0	2.437
8384	1.4420	2.0150	0.1875	0.5640	1.1680	0.3710	43.0	2.540

Note: Some gerotors are better suited to some applications than others. Please contact Nichols Portland Product Engineering prior to final selection.

* See Gerotor Identification Code, Page 5.

** Recommended maximum operating speed (see Section 6D).

*** These minimum and maximum recommended feature sizes are for reference only and does not imply that standard tooling is available to produce them. Also, utilizing parts made to these sizes may adversely affect performance and/or longevity of the device. Please contact Nichols Portland Product Engineering when considering a non-standard I.D. or O.D.

***** Limited tooling availability and restricted material choices – contact Nichols Portland prior to selecting this gerotor.

† Porting radii tolerances are generally not critical. The accuracy of sealing lands E, F, and G become more important as performance requirements increase, since errors will result in trapped fluids or excessive leakage. For most applications, E, F, and G will operate well with a +.01/-0” (+.25/0.0mm) or angular equivalent tolerance. IMPORTANT – porting tolerances are different than housing or bearing tolerances! Please contact Nichols Portland Product Engineering if you need additional advice related to a specific application.

Nichols Portland Standard Gerotors and Specifications

(Metric units table)

Gerotor Type*	Maximum Operating Speed**	Thickness Min.	Nominal Thickness Max.	Nominal O.D. Standard	Nominal O.D. Minimum Recommended***	Nominal I.D. Standard	Nominal I.D. Maximum Recommended***
	(rpm)	(mm)	(mm)	(mm)	(mm)	(mm	(mm)
6010	21556	2.540	7.620	20.45	20.45	5.13	5.13
10010	17250	3.175	10.160	25.35	22.17	7.95	9.53
6020****	16000	3.175	12.700	28.52	28.52	7.95	7.95
6022	14500	3.175	12.700	28.52	28.52	7.95	7.95
8030	11250	3.175	25.400	38.05	34.87	12.70	15.88
10060	7250	4.775	25.400	57.10	53.92	15.88	25.40
6063	9000	4.775	22.225	50.75	44.40	15.88	17.48
4065	10000	4.775	25.400	44.40	41.25	12.70	12.70
4086	7850	4.775	25.400	50.01	50.01	12.70	12.70
6095	7250	4.775	31.750	57.10	57.10	19.05	19.05
12131	4500	6.350	25.400	82.50	79.32	25.40	44.45
4158	6250	6.350	34.925	62.81	62.81	13.79	22.23
14162****	3750	6.350	22.225	101.55	95.20	41.28	53.98
6166	5000	6.350	38.100	76.50	76.50	25.40	28.58
6170	5500	6.350	8.100	76.15	72.97	25.40	28.58
4180	6000	9.525	31.750	66.73	66.73	14.30	22.23
10239	3655	9.525	25.400	100.33	100.33	44.45	50.80
6280	4250	9.525	38.100	101.55	95.25	31.75	38.10
8369****	3250	9.525	76.200	114.25	114.25	44.45	53.98
8384	3250	9.525	38.100	114.25	114.25	44.45	53.98
10397****	2800	9.525	25.400	127.00	127.00	50.80	60.96

Standard Gerotor Porting Information† (See Figure 4, page 11)

Gerotor Type	Nominal Porting Dimensions: Radius A	Radius B	Ecc. C	Width D	Land E	Land F	Angle G	Port Area
	(mm)	(mm)	(mm)	(mm)	(mm)	(mm)	(Degrees)	(mm²)
6010	5.004	7.874	0.871	2.616	5.029	2.769	52.4	43.87
10010	6.934	9.169	0.660	1.981	4.420	1.981	34.5	49.03
6020****	6.553	10.770	1.321	3.962	6.807	2.870	54.0	88.39
6022	7.290	11.506	1.321	3.962	7.544	3.404	54.0	94.84
8030	10.592	14.808	1.321	3.962	8.382	3.607	42.0	138.06
10060	17.323	22.555	1.651	4.953	11.049	5.588	34.5	277.42
6063	11.938	19.050	2.286	6.858	12.573	4.674	55.0	263.23
4065	9.296	17.932	2.794	8.382	13.995	6.452	79.5	236.13
4086	12.852	21.590	2.819	8.458	18.745	10.439	79.5	303.23
6095	14.732	23.368	2.794	8.382	15.494	5.969	55.0	391.61
12131	27.178	34.290	2.286	6.858	14.249	5.944	28.5	601.29
4158	15.113	27.940	4.191	12.573	22.225	11.913	77.5	551.61
14162****	34.138	41.250	2.286	6.858	15.240	7.899	24.5	748.39
6166	22.225	32.741	3.419	10.262	22.352	14.605	52.0	649.68
6170	19.050	30.734	3.810	11.430	19.761	8.128	54.0	692.26
4180	14.605	29.185	4.775	14.326	22.098	9.754	79.0	638.71
10239	33.017	43.475	3.401	10.203	20.701	8.641	33.6	1065.80
6280	25.400	39.929	4.775	14.326	26.670	10.211	55.0	1125.80
8369****	34.925	49.479	4.763	14.326	28.092	8.509	43.0	1572.25
8384	36.627	51.181	4.763	14.326	29.667	9.423	43.0	1638.71
10397****	42.926	56.388	4.343	12.700	27.432	10.414	34.5	1740.00

Note: Some gerotors are better suited to some applications than others. Please contact Nichols Portland Product Engineering prior to final selection.

* See Gerotor Identification Code, Page 5.

** Recommended maximum operating speed (see Section 6D).

*** These minimum and maximum recommended feature sizes are for reference only and does not imply that standard tooling is available to produce them. Also, utilizing parts made to these sizes may adversely affect performance and/or longevity of the device. Please contact Nichols Portland Product Engineering when considering a non-standard I.D. or O.D.

***** Limited tooling availability and restricted material choices – contact Nichols Portland prior to selecting this gerotor.

† Porting radii tolerances are generally not critical. The accuracy of sealing lands E, F, and G become more important as performance requirements increase, since errors will result in trapped fluids or excessive leakage. For most applications, E, F, and G will operate well with a +.01/-0" (+.25/0.0mm) or angular equivalent tolerance. IMPORTANT – porting tolerances are different than housing or bearing tolerances! Please contact Nichols Portland Product Engineering if you need additional advice related to a specific application.

Gerotor Pump Design Checklist

❑ **1. Flow requirement:**

Q = ______ gpm or ______ lpm at N = _______ rpm

❑ **2. Calculate theoretical displacement:**

English units: Dt = 231 x Q / N in^3/rev

Metric units: Dt = 1000 x Q / N cc/rev

❑ **3. For an assumed volumetric efficiency (Ev), calculate the required pump displacement.**

(For help in determining an appropriate Ev, contact Nichols Portland Product Engineering):

Dr = Dt / Ev

❑ **4. Custom or standard gerotor?**

a. Application specific gerotor?

b. Standard gerotor with application specific drive?

c. Standard gerotor?

Reference the Pump Design section, Step 3 for further details.

❑ **5. Calculate width of gerotor:**

English units: W = Dr / Du inches

Metric units: W = Dr / Du mm

❑ **6. Does the maximum speed in the application exceed the maximum recommended operating speed?**

If so, contact Nichols Portland Product Engineering for recommendations.

❑ **7. Inlet velocity checks:**

Is inlet velocity below 6 ft/sec (2 m/sec)?

Does the gerotor need to be dual filled?

Port area = Ap = ______ in^2 = ______ mm^2

English units: V = 231 x Q / (60 x 12 x Ap) ft/sec

Metric units: V = 1000 x Q / (60 x Ap) m/sec

❑ **8. Check the pump theoretical input power and torque against the prime mover power and torque.**

❑ **9. Choose type of porting, based on manufacturing methods and application.**

❑ **10. Lay out pump design, keeping in mind:**

a. Inlet losses should be minimized (length of line, bends, transitions, and lift)

b. Deflection of housing, cover plate, and shaft.

c. Pressure and area balance of the inner rotor.

d. Design and routing of the bypass valve and system.

❑ **11. Contact Nichols Portland Product Engineering** with any questions, or for a design review of your pump. We provide design reviews as a free service to all customers to ensure proper application of the gerotor. Our years of experience at pump designs and troubleshooting may save you weeks or months of development time.

Gerotor Pump and Motor Equations

Pump Equations	English Units	Metric Units
Flow Out	$Q=\frac{E_V \times D \times N}{231}$ gpm	$Q=\frac{E_V \times D \times N}{1000}$ lpm
Torque In	$T=\frac{D \times P}{2 \times \varpi \times E_M}$ in-lb	$T=\frac{D \times P}{2 \times \varpi \times E_M \times 1000}$ N-m
Mechanical Power In	$P_{IN}=\frac{T \times N}{63025}$ horsepower	$P_{IN}=\frac{2 \times \varpi \times T \times N}{60}$ Watts
Hydraulic Power Out	$P_{OUT}=\frac{P \times Q}{1714.3}$ horsepower	$P_{OUT}=\frac{P \times Q}{60}$ Watts
Overall Efficiency	$E_{OVR}=\frac{P_{OUT}}{P_{IN}}=\frac{36.77 \times P \times Q}{T \times N}$	$E_{OVR}=\frac{P \times Q}{2 \times \varpi \times T \times N}$

Motor Equations	English Units	Metric Units
Flow In	$Q=\frac{D \times N}{231 \times E_V}$ gpm	$Q=\frac{D \times N}{1000 \times E_V}$ lpm
Torque Out	$T=\frac{D \times P \times E_M}{2 \times \varpi}$ in-lb	$T=\frac{D \times P \times E_M}{2 \times \varpi \times 1000}$ N-m
Hydraulic Power In	$P_{IN}=\frac{P \times Q}{1714.3}$ horsepower	$P_{IN}=\frac{P \times Q}{60}$ Watts
Mechanical Power Out	$P_{OUT}=\frac{T \times N}{63025}$ horsepower	$P_{OUT}=\frac{2 \times \varpi \times T \times N}{60}$ Watts
Overall Efficiency	$E_{OVR}=\frac{P_{OUT}}{P_{IN}}=\frac{.0272 \times T \times N}{P \times Q}$	$E_{OVR}=\frac{2 \times \varpi \times T \times N}{P \times Q}$

Where:

Variable	Description	English Units	Metric Units
Q	Flow	gallons per minute (gpm)	liters per minute (lpm)
P	Pressure	pounds per square inch (psi)	kiloPascal (kPa)
T	Torque	inch-pounds (in-lb)	Newton-meters(N-m)
N	Speed	revolutions per minute (rpm)	revolutions per minute (rpm)
D	Displacement revolution (in3/rev)	cubic inches per revolution (cc/rev)	cubic centimeters per
E_V	Volumetric Efficiency (0 - 1.0)		
E_M	Mechanical Efficiency (0 - 1.0)		
E_{OVR}	Overall Efficiency (0 - 1.0)		

Nichols Portland

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Engineering

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From ____________________

Phone ____________________ Fax ____________________

Customer Information

Name ____________________ Company/Division ____________________

Type of business ____________________ Address ____________________

Engineering contact ____________________ Phone ____________________

Purchasing contact ____________________ Phone ____________________

Application Information

Application: ____________ Pump ____________ Motor ____________ Other ____________

Describe ____________________

____

New application ____________________ Existing application ____________________

If existing, change is due to: ____________ Performance ____________ Price ____________ Package size ____________

Estimated annual production quantities: ____________________

Timing: ____________ Experimental hardware Production ____________

Application requirements

Application constraints

Design flow: ____________ GPM/LPM at ____________ RPM

Flow range: Min ____________ to Max ____________ GPM/LPM

Speed range: Min ____________ to max ____________ RPM

Pressure range: Min ____________ to max ____________ PSI/kPa

Lift: ____________ cm/in

Fluid: ____________________

Viscosity: ____________ cP/SUS at ____________ degrees F/C

Temperature range: ____________ degrees F/C

Packaging constraints

Min I.D.: ____________________ Max O.D.: ____________________

Max thickness: ____________________

Action Requested

- ☐ Standard gerotor recommendation required
- ☐ New gerotor design required
- ☐ Design recommendation required (Porting and tolerances)
- ☐ Complete pump design required
- ☐ Complete motor design required

Parker

Notes

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