

# Gerotor Selection *and Pump Design*





*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 that can best be addressed by one of our Product Engineers. Please don't hesitate to contact us.*

---

## Gerotor Selection and Pump Design, Version 1.1

---

### Introduction

---

Nichols Portland

The Gerotor Principle

The Gerotor Advantage

Gerotor Identification Code

---

### Pump Design

---

1. Determining application requirements

2. Dimensional constraints

3. Standard or custom gerotor

4. Selecting a standard gerotor

5. Pump design

Pump porting

Pump construction

Reversing pumps

Clearances and tolerances

Driving methods

Other design considerations

6. Pump performance

Pump overall efficiency

Pump volumetric efficiency

Pump mechanical efficiency

Cavitation

Noise

Summary

Glossary of Terms and Abbreviations

Standard gerotors and Specification Tables

---

## 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.

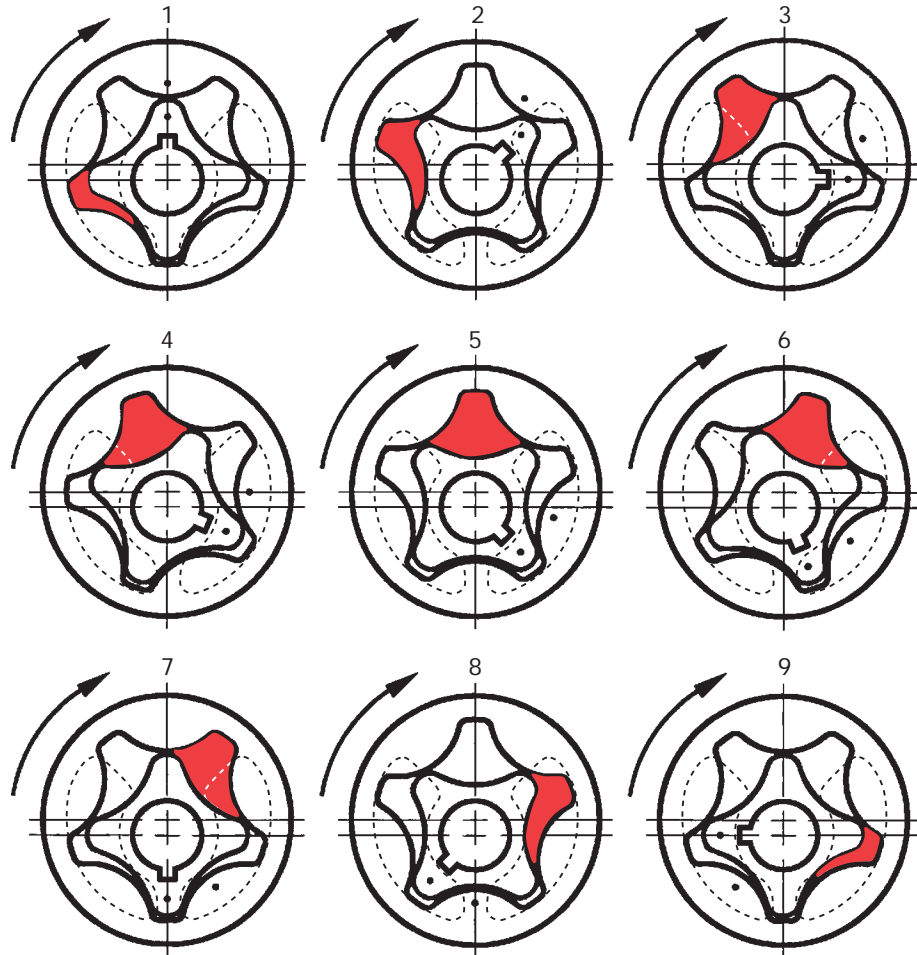


# Introduction

# 3



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 1500 psi (10,000 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<sup>3</sup>/rev/in). To determine the gerotor unit displacement, simply take the last three digits of the code and divide by 100 to get in<sup>3</sup>/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<sup>3</sup>/rev/in. and 6 teeth on the inner rotor. The 10010 gerotor has a unit displacement of 0.10 in<sup>3</sup>/rev/in and 10 teeth on the inner rotor.

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

Using gerotor type 6170 (1.70 in<sup>3</sup>/rev/in) as an example:

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

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 (in<sup>3</sup>/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.)*

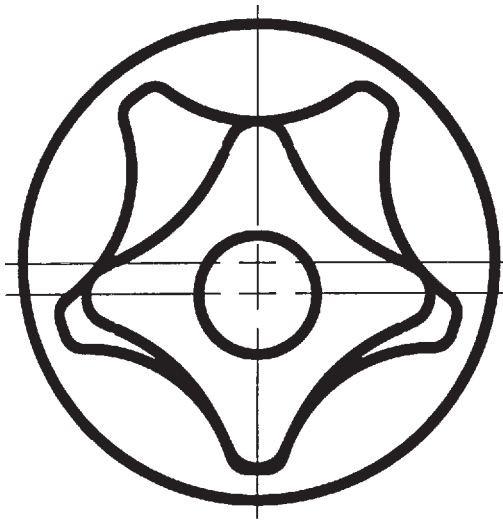
To calculate required displacement:

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



## Gerotors of Identical Theoretical Displacement

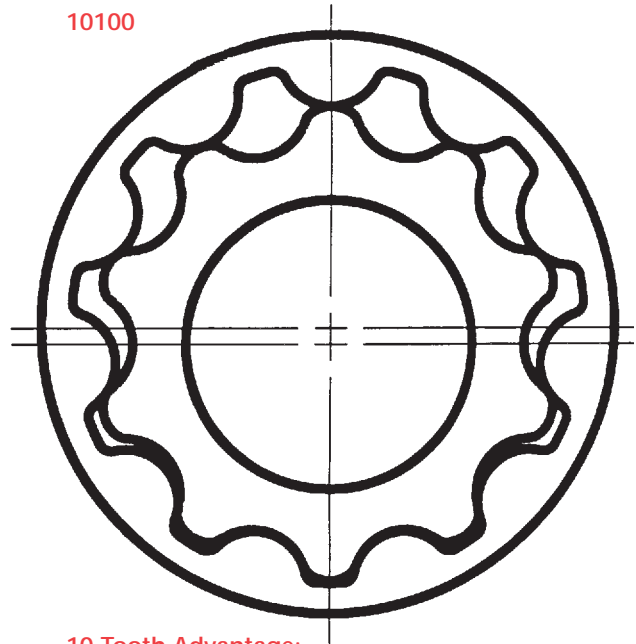
4100



4 Tooth Advantage:

- Smaller package
- Less power loss
- Higher operating speed

10100



10 Tooth Advantage:

- Less flow ripple
- Accommodates larger shaft

Figure 2

### 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

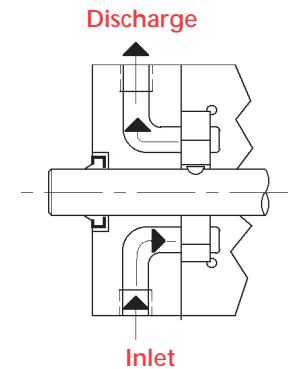
Each gerotor has a recommended maximum operating speed based on the onset of cavitation (see pages 22 and 23). Operating above this speed without corrective measures will degrade pump performance. For additional information, see the section on Cavitation.

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

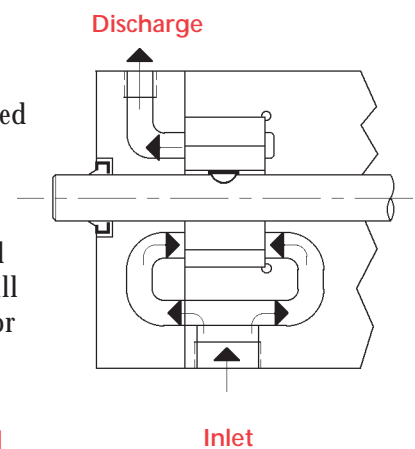
Theoretical flow ripple depends on the number of teeth and the specific geometry of the gerotor, but in general, more teeth means lower flow ripple. Also, a gerotor with an even number of inner teeth typically has a lower flow ripple than a gerotor with an odd toothed inner having one more tooth.

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 are 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 are 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

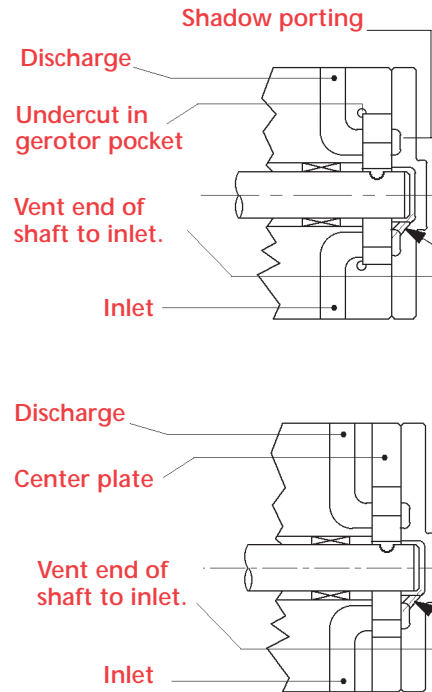




Figure 5

## 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's material 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.

The second 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.

### Reversibility Feature

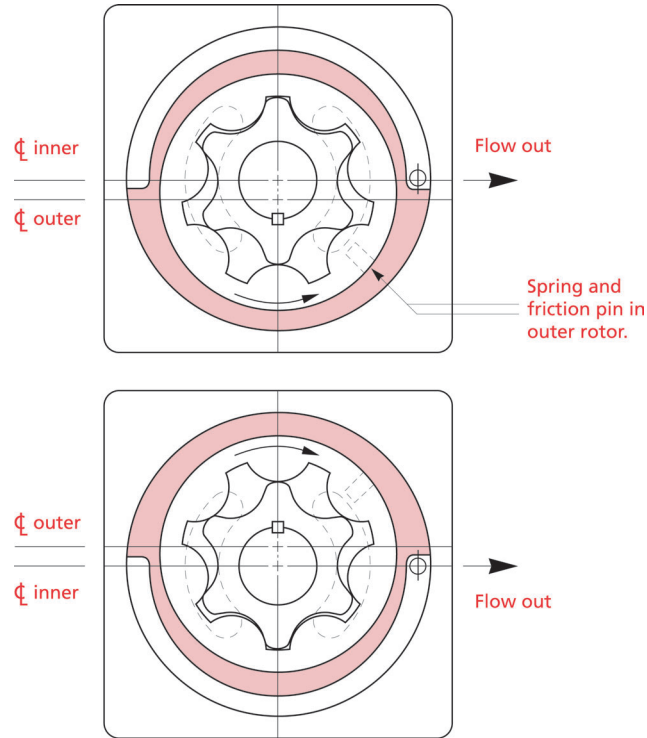


Figure 6

### Schematic: Reversing Pump With Check Valves

Figure 7

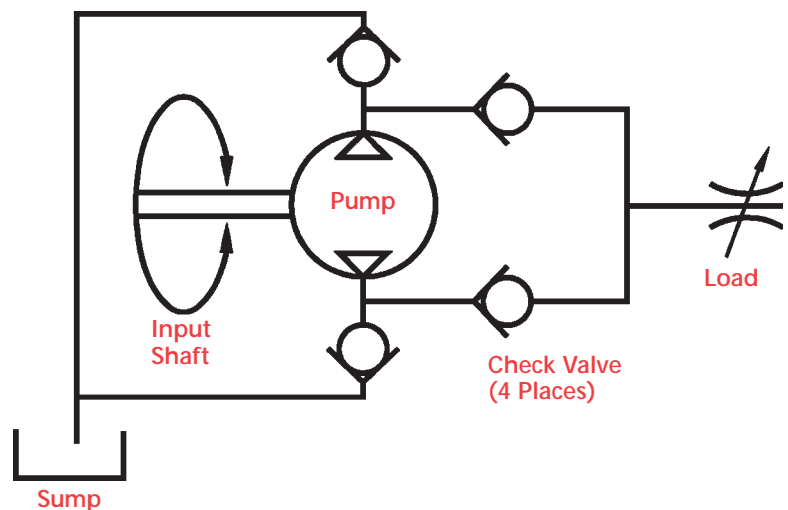




Figure 8

## Operating Conditions And Clearances For Some Typical Applications

	English	Metric
<i>Engine Lubrication Pump</i>		
Pressures:	60 to 80 psi	(400 to 550 kPa)
Speeds:	400 to 14,000 rpm	(400 to 14,000 rpm)
Displacements:	0.4 to 1.0 in <sup>3</sup> /rev	(6.5 to 16.5 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)
<i>Automatic Transmission Pump</i>		
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 <sup>3</sup> /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)
<i>High Pressure Pump</i>		
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 <sup>3</sup> /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)

Note: These clearances are application dependent and should be used only as guidelines.

## 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.

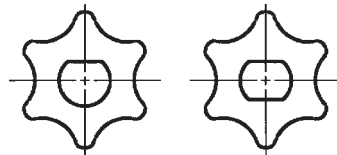
## F. Other Important Considerations

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.

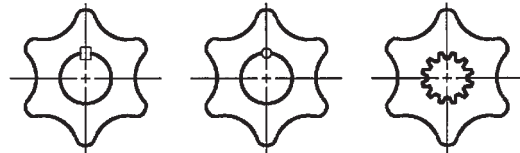
## Common Drive Configurations

Figure 9

Floating:



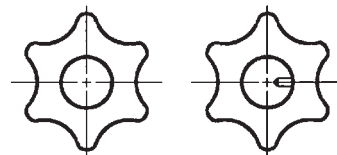
Round shaft with flat(s).



Round shaft with key or dowel.

Splined shaft.

Fixed:

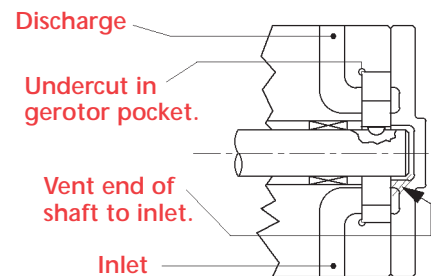


Press fit on round or serrated shaft.

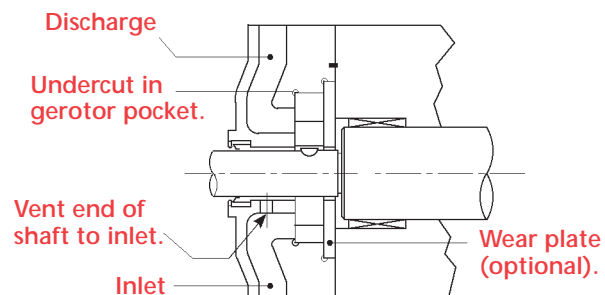
Press fit on shaft and pinned.

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 material 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.

Extreme operating conditions 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 ( $L_r$ ) on an inner rotor due to pressure is calculated by

$$L_r = p \times W \times (A + B)$$

$A$  = inside porting radius

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

$W$  = gerotor width

$p$  = pump outlet pressure

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



## Step 6: Pump performance

A hydraulic pump converts mechanical power ( $P_m = \text{speed} \times \text{torque}$ ) into hydraulic power ( $P_h = \text{flow} \times \text{pressure}$ ).

### A. Pump Overall Efficiency

The overall efficiency ( $E_{ovr}$ ) of the pump is a measure of the overall effectiveness of this conversion, and is defined as:

$$E_{ovr} = \frac{\text{Power Out}}{\text{Power In}} = \frac{P_h}{P_m}$$

The input mechanical power ( $P_m$ ) is calculated by

*English units:*

$$P_m = \frac{T \times N \times 2 \times \pi}{550 \times 12 \times 60} \text{ Horsepower}$$

where 550 converts ft.lbf/sec to h.p.

$T$  = input torque in pound/inches

$N$  = input shaft speed in rpm

*Metric units:*

$$P_m = \frac{2 \times \pi \times T \times N}{60} \text{ Watts}$$

$T$  = input torque in Newton · meters

$N$  = input shaft speed in rpm

The output hydraulic power ( $P_h$ ) is calculated by

*English units:*

$$P_h = \frac{p \times Q \times 231}{550 \times 12 \times 60} \text{ Horsepower}$$

$p$  = the pressure rise across the pump in psi

$Q$  = the pump flow in gallons per minute

*Metric units:*

$$P_h = \frac{p \times Q}{60} \text{ Watts}$$

$p$  = pressure rise across the pump in kPa

$Q$  = pump flow in liters per minute

Overall efficiency is also a product of volumetric ( $E_v$ ) and mechanical ( $E_m$ ) efficiency.

### B. Pump Volumetric Efficiency

The volumetric efficiency is a measure of the actual flow ( $Q_a$ ) produced by the pump at a given speed compared to the theoretical flow ( $Q_t$ ) 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:*

$$E_v = \frac{Q_a}{Q_t} = \frac{Q_a \times 231}{D \times N} \times 100\%$$

$D$  = pump displacement in in<sup>3</sup>/rev

$N$  = pump speed in rpm

$Q_a$  = actual flow in gpm

*Metric units:*

$$E_v = \frac{Q \times 100}{D \times N} \times 100\%$$

$D$  = pump displacement in cc/rev

$N$  = pump speed in rpm

$Q_a$  = actual flow in lpm



### C. Pump Mechanical Efficiency

Mechanical efficiency is a ratio of theoretical torque ( $T_t$ ) the pump would require compared to the actual input torque ( $T_a$ ). 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:*

$$E_m = \frac{T_t}{T_a} = \frac{D \times p}{2 \times \pi \times T_a}$$

$(EM \leq 1.0)$

$D$  = pump displacement in  $\text{in}^3/\text{rev}$

$p$  = pressure rise across the pump in  $\text{psi}$

$T_t$  = theoretical torque in  $\text{in} \cdot \text{lbf}$

$T_a$  = actual torque in  $\text{in} \cdot \text{lbf}$

#### *Metric units:*

$$E_m = \frac{D \times p}{2 \times \pi \times 1000 \times T_a} \times 100\%$$

$D$  = pump displacement in  $\text{in}^3/\text{rev}$

$p$  = pressure rise across the pump in  $\text{psi}$

$T_t$  = theoretical torque in  $\text{Newton} \cdot \text{meters}$

$T_a$  = actual torque in  $\text{Newton} \cdot \text{meters}$

## D. Cavitation

Pump cavitation occurs whenever the local inlet pressure is below the vapor pressure of the fluid. When cavitation occurs, vapor bubbles form in the inlet circuit, displacing fluid at the pump inlet and reducing the output flow of the pump. When these bubbles are carried over to the high-pressure side of the pump, they implode and can cause noise and vibration. Cavitation bubbles can also damage the pump through erosion of the housing and gerotor material.

The three major contributors to pump cavitation are: gerotor tip speed, fluid inlet velocity, and suction line losses. For optimum performance with petroleum-based fluids having a viscosity of 5 to 100cP or mPa's, tip speed of the inner rotor should be kept below 50 ft/sec (15.3 m/sec), the inlet velocity of the fluid entering a pump should be kept below 6 ft/sec (2 m/sec), and inlet lines should be designed to minimize losses.

Cavitation should not be confused with aeration, which is the presence of air bubbles.

## 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 <sup>2</sup>	mm <sup>2</sup>
A,B,C,D, E, F, G, H, J, L	Dimensions for porting	inch	mm
Dr	Required displacement	in <sup>3</sup> /rev	cc/rev
Dt	Theoretical displacement	in <sup>3</sup> /rev	cc/rev
Du	Unit displacement	in <sup>3</sup> /rev/inch	cc/rev /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** (rpm)	Thickness Min. (inch)	Nominal Range Max. (inch)	Nominal O.D.		Nominal I.D.	
				Standard (inch)	Minimum Recommended*** (inch)	Standard (inch)	Maximum Recommended*** (inch)
6010	21556	0.100	0.360	0.805	0.805	0.202	0.202
10010	17250	0.125	0.500	0.998	0.873	0.313	0.375
6020	16000	0.125	0.750	1.123	1.123	0.313	0.313
6022	14500	0.125	0.625	1.123	1.123	0.313	0.313
8030	11250	0.125	0.625	1.498	1.373	0.500	0.625
10060	7250	0.188	1.250	2.248	2.123	0.625	1.000
6063	9000	0.188	1.250	1.998	1.748	0.625	0.688
4065	10000	0.188	1.250	1.748	1.624	0.500	0.500
4086	7850	0.188	1.250	1.969	1.969	0.500	0.500
6095	7250	0.188	1.500	2.248	2.248	0.750	0.750
12131	4500	0.250	2.000	3.248	3.123	1.000	1.750
4158	6250	0.250	1.375	2.473	2.473	0.543	0.875
14162	3750	0.250	1.500	3.998	3.748	1.625	2.125
6170	5500	0.250	2.000	2.998	2.873	1.000	1.125
6166	5000	0.250	1.375	2.998	2.998	1.000	1.125
4180	6000	0.250	1.375	2.627	2.627	0.563	0.875
6280	4250	0.250	3.000	3.998	3.750	1.250	1.500
8369	3250	0.375	3.000	4.498	4.498	1.750	2.125
8384	3250	0.375	1.750	4.498	4.498	1.750	2.125
10397	2800	0.375	1.000	5.000	5.000	2.000	2.400

## Standard Gerotor Porting Information (See Figure 4)

Gerotor Type	Nominal Porting Dimensions							
	Radius A (inch)	Radius B (inch)	Ecc. C (inch)	Width D (inch)	Land E (inch)	Land F (inch)	Angle G (degrees)	Port Area (in <sup>2</sup> )
6010	0.197	0.310	0.035	0.103	0.198	0.109	52.4	0.068
10010	0.273	0.361	0.026	0.078	0.174	0.078	34.5	0.076
6020	0.258	0.424	0.052	0.156	0.268	0.113	54.0	0.137
6022	0.287	0.453	0.052	0.156	0.297	0.134	54.0	0.147
8030	0.417	0.583	0.052	0.156	0.330	0.142	42.0	0.214
10060	0.682	0.888	0.065	0.195	0.435	0.220	34.5	0.430
6063	0.470	0.750	0.090	0.270	0.495	0.184	55.0	0.408
4065	0.366	0.706	0.110	0.330	0.551	0.254	79.5	0.366
4086	0.506	0.850	0.111	0.333	0.738	0.411	79.5	0.320
6095	0.580	0.920	0.110	0.330	0.610	0.235	55.0	0.607
12131	1.070	1.350	0.090	0.270	0.561	0.234	28.5	0.932
4158	0.595	1.100	0.165	0.495	0.875	0.469	77.5	0.855
14162	1.344	1.624	0.090	0.270	0.600	0.311	24.5	1.160
6170	0.750	1.210	0.150	0.450	0.778	0.320	54.0	1.073
6166	0.875	1.289	0.135	0.404	0.880	0.575	52.0	1.007
4180	0.575	1.149	0.188	0.564	0.870	0.384	79.0	0.990
6280	1.000	1.572	0.188	0.564	1.050	0.402	55.0	1.745
8369	1.375	1.948	0.188	0.564	1.106	0.335	43.0	2.437
8384	1.442	2.015	0.188	0.564	1.168	0.371	43.0	2.540
10397	1.690	2.220	0.171	0.500	1.080	0.410	34.5	2.697

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

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

\*\*\* These minimum and maximum recommended feature sizes are for reference only. This does NOT imply that tooling is available to produce these sizes, and an additional tooling charge may be required in order to produce non-standard O.D.s or I.D.s. Please contact Nichols Portland Product Engineering prior to considering a non-standard I.D. or O.D.



## Nichols Portland Standard Gerotors and Specifications\*

(Metric Units Table)

Gerotor Type	Maximum Operating Speed**	Thickness Min.	Nominal Range Max.	Nominal O.D.		Nominal I.D.	
				Standard	Minimum Recommended***	Standard	Maximum Recommended***
	(rpm)	(mm)	(mm)	(mm)	(mm)	(mm)	(mm)
6010	21556	2.5	9.1	20.40	20.40	5.10	5.10
10010	17250	3.175	12.7	25.35	22.17	7.95	9.53
6020	16000	3.175	19.1	28.52	28.52	7.95	7.95
6022	14500	3.175	15.9	28.52	28.52	7.95	7.95
8030	11250	3.175	15.9	38.05	34.87	12.70	15.88
10060	7250	4.8	31.8	57.10	53.92	15.88	25.40
6063	9000	4.8	31.8	50.75	44.40	15.88	17.48
4065	10000	4.8	31.8	44.40	41.25	12.70	12.70
4086	7850	4.8	31.8	50.01	50.01	12.70	12.70
6095	7250	4.8	38.1	57.10	57.10	19.05	19.05
12131	4500	6.4	50.8	82.50	79.32	25.40	44.45
4158	6250	6.4	34.9	62.81	62.81	13.79	22.23
14162	3750	6.4	38.1	101.55	95.20	41.28	53.98
6170	5500	6.4	50.8	76.15	72.97	25.40	28.58
6166	5000	6.35	34.9	76.15	76.15	25.40	28.58
4180	6000	6.4	34.9	66.73	66.73	14.30	22.23
6280	4250	6.4	76.2	101.55	95.25	31.75	38.10
8369	3250	6.4	76.2	114.25	114.25	44.45	53.98
8384	3250	6.4	44.45	114.25	114.25	44.45	53.98
10397	2800	9.5	25.4	127.00	127.00	50.80	60.96

## Standard Gerotor Porting Information (See Figure 4)

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)	(mm <sup>2</sup> )
6010	5.0	7.9	0.9	2.6	5.0	2.8	52.4	44.0
10010	6.9	9.2	0.660	2.0	4.4	2.0	34.5	49.0
6020	6.6	10.8	1.321	4.0	6.8	2.9	54.0	88.4
6022	7.3	11.5	1.321	4.0	7.5	3.4	54.0	94.8
8030	10.6	14.8	1.321	4.0	8.4	3.6	42.0	138.1
10060	17.3	22.6	1.651	5.0	11.0	5.6	34.5	277.4
6063	11.9	19.1	2.286	6.9	12.6	4.7	55.0	263.2
4065	9.3	17.9	2.794	8.4	14.0	6.5	79.5	236.1
4086	12.9	21.6	2.819	8.5	18.7	10.4	79.5	206.5
6095	14.7	23.4	2.794	8.4	15.5	6.0	55.0	391.6
12131	27.2	34.3	2.286	6.9	14.2	5.9	28.5	601.3
4158	15.1	27.9	4.191	12.6	22.2	11.9	77.5	551.6
14162	34.1	41.2	2.286	6.9	15.2	7.9	24.5	748.4
6170	19.1	30.7	3.810	11.4	19.8	8.1	54.0	692.3
6166	22.2	32.7	3.429	10.3	22.3	14.6	52	649.7
4180	14.6	29.2	4.775	14.3	22.1	9.8	79.0	638.7
6280	25.4	39.9	4.775	14.3	26.7	10.2	55.0	1125.8
8369	34.9	49.5	4.775	14.3	28.1	8.5	43.0	1572.3
8384	36.6	51.2	4.775	14.3	29.7	9.4	43.0	1638.7
10397	42.9	57.4	4.343	12.7	27.4	10.4	34.5	1740.0

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

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

\*\*\* These minimum and maximum recommended feature sizes are for reference only. This does NOT imply that tooling is available to produce these sizes, and an additional tooling charge may be required in order to produce non-standard O.D.s or I.D.s. Please contact Nichols Portland Product Engineering prior to considering a non-standard I.D. or O.D.

# Gerotor Pump Design Checklist

- ☐ 1. Flow requirement:  
 $Q = \text{_____ gpm or _____ lpm at } N = \text{_____ rpm}$
- ☐ 2. Calculate theoretical displacement:  
English units:  $D_t = 231 \times Q / N \text{ in}^3/\text{rev}$   
Metric units:  $D_t = 1000 \times Q / N \text{ cc/rev}$
- ☐ 3. For an assumed volumetric efficiency ( $E_v$ ), calculate the required pump displacement.  
(For help in determining an appropriate  $E_v$ , contact Nichols Portland Product Engineering):  
 $D_r = D_t / E_v$
- ☐ 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:  $W = D_r / D_u \text{ inches}$   
Metric units:  $W = D_r / D_u \text{ mm}$
- ☐ 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 =  $A_p = \text{_____ in}^2 = \text{_____ mm}^2$   
English units:  $V = 231 \times Q / (60 \times 12 \times A_p) \text{ ft/sec}$   
Metric units:  $V = 1000 \times Q / (60 \times A_p) \text{ m/sec}$
- ☐ 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	$Q = \frac{E_v \times D \times N}{231} \text{ gpm}$	$Q = \frac{E_v \times D \times N}{1000} \text{ lpm}$
Torque In	$T = \frac{D \times P}{2 \times \pi \times E_M} \text{ in-lb}$	$T = \frac{D \times P}{2 \times \pi \times E_M \times 1000} \text{ N-m}$
Mechanical Power In	$P_{IN} = \frac{T \times N}{63025} \text{ horsepower}$	$P_{IN} = \frac{2 \times \pi \times T \times N}{60} \text{ Watts}$
Hydraulic Power Out	$P_{OUT} = \frac{P \times Q}{1714.3} \text{ horsepower}$	$P_{OUT} = \frac{P \times Q}{60} \text{ Watts}$
Overall Efficiency	$E_{OVR} = \frac{P_{OUT}}{P_{IN}} = \frac{36.77 \times P \times Q}{T \times N}$	$E_{OVR} = \frac{P \times Q}{2 \times \pi \times T \times N}$

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

### 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	cubic inches per revolution (in <sup>3</sup> /rev)	cubic centimeters per revolution (cc/rev)
E <sub>v</sub>	Volumetric Efficiency (0 - 1.0)		
E <sub>M</sub>	Mechanical Efficiency (0 - 1.0)		
E <sub>OVR</sub>	Overall Efficiency (0 - 1.0)		





**Nichols**  
Portland

Design

Engineering

Manufacturing

**Nichols Portland**

**A Division of Parker Hannifin Corporation**

2400 Congress Street

Portland, Maine 04102 USA

Voice: 207 774 6121

Fax: 207 774 3601

**Fax to: 207 774 3601, Attn: Product Engineering**

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

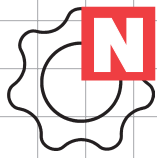


Complete motor design required



25







**Nichols Portland**

**A Division of Parker Hannifin Corporation**

2400 Congress Street

Portland, Maine 04102 USA

Voice: 207 774 6121

Fax: 207 774 3601

[www.nicholsportland.com](http://www.nicholsportland.com)

[www.gerotor.com](http://www.gerotor.com)

[www.parker.com](http://www.parker.com)

**Nichols Europe**

**Parker Hannifin Corporation**

Lange Strasse 29

D-26892 Doerpen

Germany

Phone: +49-(0) 4953-91-65-28

Fax: +49-(0) 4963-91-65-29

Mobile: +49-(0) 175-57-56-176

Email: [nrenemann@parker.com](mailto:nrenemann@parker.com)

