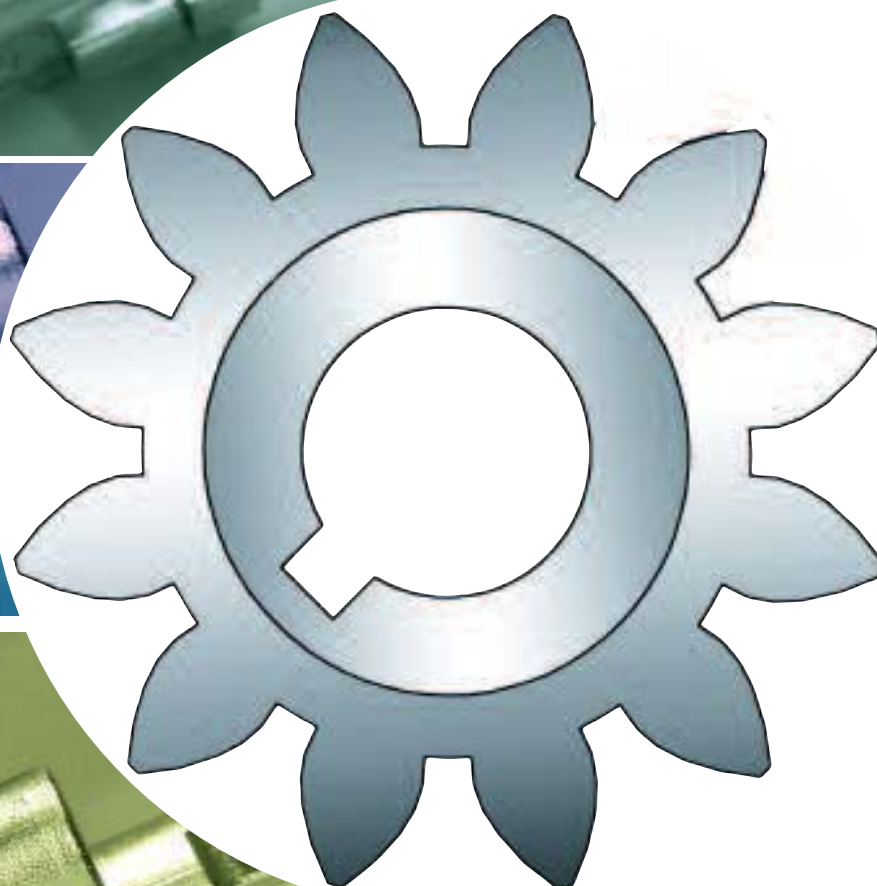
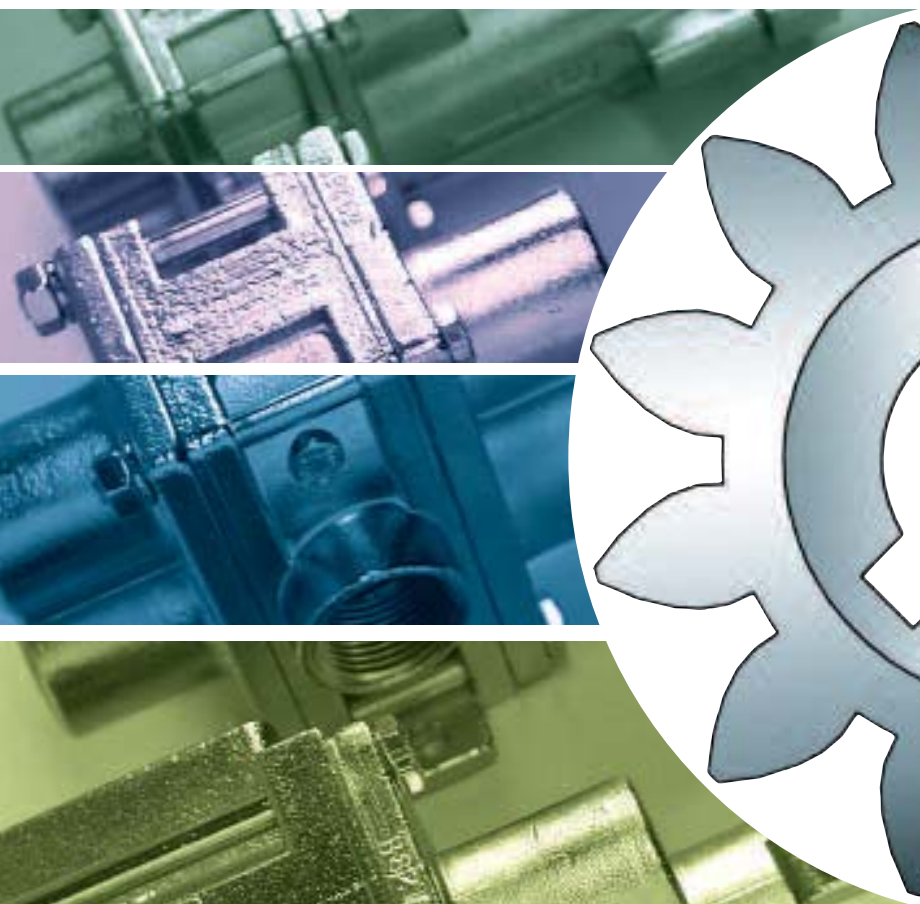




# ENGINEERING



LIQUIFLO CHEMICAL PROCESSING PUMPS



## Introduction

This section of the catalog contains valuable information about gear and centrifugal pumps, and their application to pumping fluids for the chemical processing industry. Please contact Liquiflo's Engineering Department if you have any questions about the material in this catalog, or to assist you with your chemical pumping applications.

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## EXTERNAL GEAR PUMP PRINCIPLE OF OPERATION &amp; ADVANTAGES

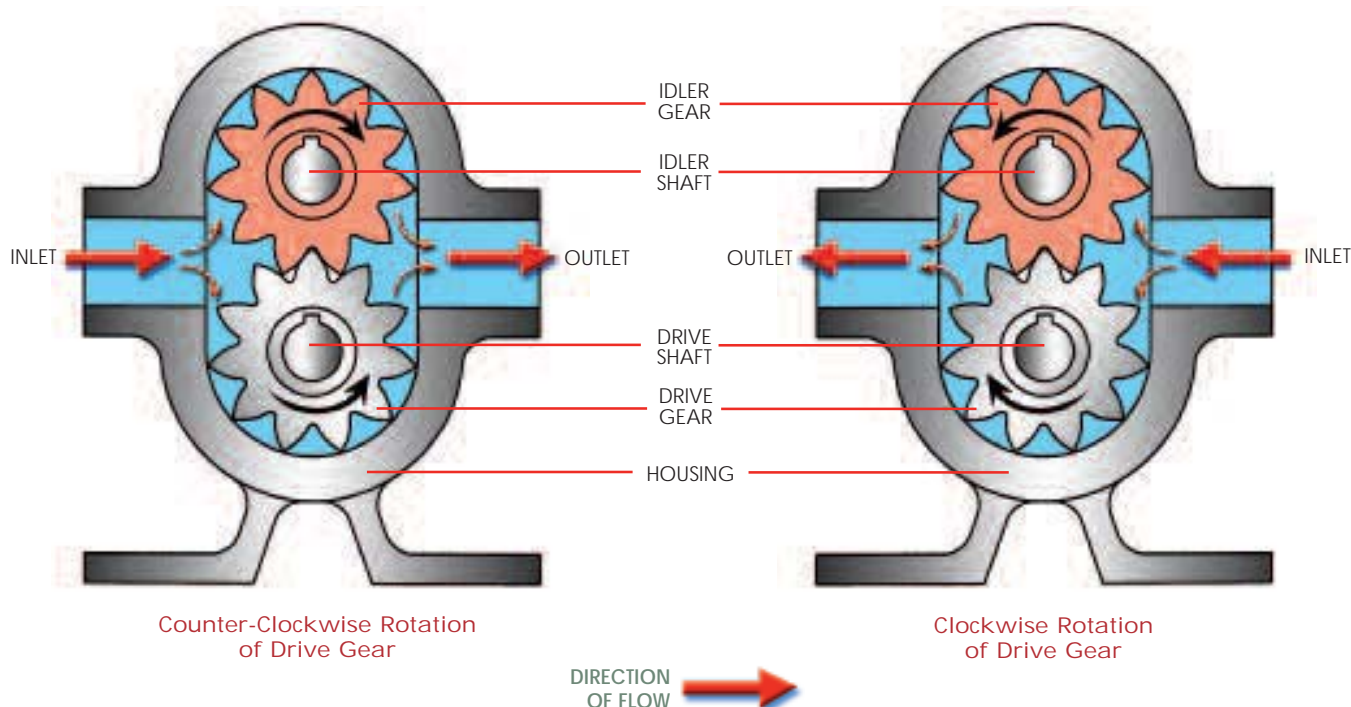


Figure 1: Cross-sectional views of external gear pump demonstrating operating principle.

The external gear pump is a positive displacement (PD) type of pump generally used for the transfer and metering of liquids. The pump is so named because it has two gears that are side-by-side or *external* to each other. (This nomenclature differentiates it from an *internal* gear pump, which has one gear positioned inside the other.) The gear pump is a precision machine with extremely tight fits and tolerances, and is capable of working against high differential pressures.

The working principle of the external gear pump is illustrated in Figure 1. A drive gear (that is driven by a motor) rotates an idler gear in the opposite direction. When the gears rotate, the liquid, which is trapped in the gear teeth spaces between the housing bore and the outside of the gears, is transferred from the inlet side of the pump to the outlet side. It is important to note that *the pumped liquid moves around the gears and not between the gears*. The rotating gears continue to deliver a fresh supply of liquid from the suction (inlet) side of the pump to the discharge (outlet) side of the pump, with *virtually no pulsations*. The meshing

of the gears on the discharge side of the pump forces the liquid out of the pump and into the discharge piping.

Figure 1 also shows that the direction of rotation of the drive gear determines the direction of flow thru the pump, and which side of the pump is the inlet and which side is the outlet. If the direction of rotation of the motor (and therefore the drive gear) is reversed, the direction of flow thru the pump will also reverse. This *bi-directional flow characteristic* is one of the many advantages inherent to gear pumps.

Another important advantage of the gear pump is its *self-priming capability*. Gear pumps are capable of self-priming because the rotating gears evacuate air in the suction line. This produces a partial vacuum that allows the atmospheric pressure to force the liquid into the inlet side of the pump. This ability of the gear pump makes it an ideal choice when the application requires that the pump be located above the liquid level, and the liquid must be lifted to the pump. Because a gear pump cannot create a perfect vacuum, the total lift (including pipe friction losses) should not exceed about 7.5 PSI, or about one-half of the atmospheric pressure.

The tight clearances of the working parts inside a gear pump are what enable it to effectively pump liquids against high pressure. Low viscosity fluids such as alcohols and other solvents have more of a tendency to "slip" thru these tight spaces from the higher-pressure discharge side of the pump back to the lower-pressure suction side of the pump. The phenomenon of *slip* causes a reduction in flow rate and pump efficiency. Slip depends on the magnitude of the differential pressure (i.e., the difference between the discharge and suction pressures), the viscosity of the liquid pumped and the working clearances inside the particular pump that is used. Slip increases with decreasing viscosity, increasing differential pressure and increasing gear-housing clearances, and is usually measured as a percent decrease from ideal flow (i.e., flow with zero slip). For fluid viscosities greater than about 50-100 cP (depending on the particular pump), the slip is minor, but it still depends on the differential pressure. This behavior is shown in Figure 2, which compares a typical gear pump's performance curve for a thin fluid (such as water with a viscosity of about 1 cP at room temperature) with that of a moderately viscous fluid (such as a particular oil with a viscosity of 100 cP).

## EXTERNAL GEAR PUMP PRINCIPLE OF OPERATION & ADVANTAGES *(continued)*

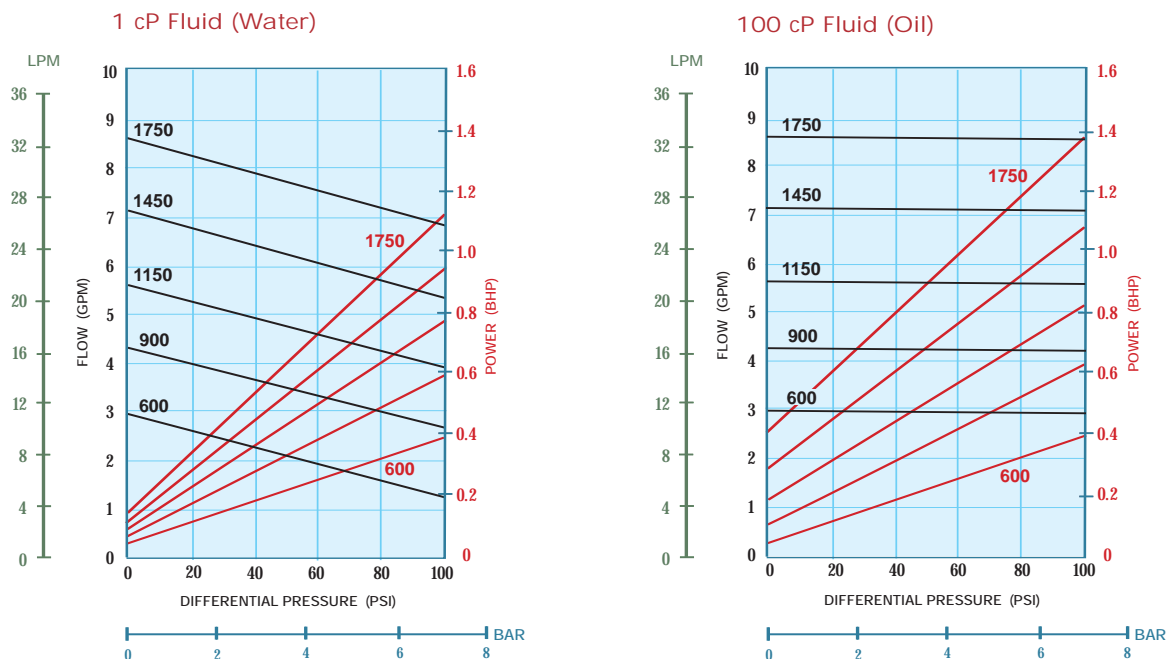


Figure 2: Performance curves for a typical external gear pump showing slip as a function of viscosity and differential pressure.

The Flow vs. Pressure curves for the thin fluid have high slopes, which indicate significant reductions in flow rate with increasing differential pressure (i.e., high slip). The curves for the 100 cP fluid are almost level, which indicate nearly constant flow rates with increasing differential pressure (i.e., almost zero slip).

The close tolerances and tight spaces inside the gear pump also limit liquids containing abrasives from being pumped. This is because the abrasive particles can work their way into these tight spaces and cause accelerated wear, and this can rapidly reduce pump performance. The resulting wear rate from pumping abrasives is dependent upon the hardness, size, and concentration of the particles, and the operating speed of the pump. The wear rate of a pump can also be adversely affected by pumping thin fluids because they have poor lubricating properties. For this reason, more care must be taken when making material selections for the internal components of the gear pump. Special materials are available to increase lubrication (such as carbon graphite) and resist wear when pumping extremely thin liquids or liquids containing abrasives. Please contact the factory to assist you with the material selection process.

Gear pumps – properly designed and engineered – can offer many advantages. These include compactness, simplicity of design, easy serviceability, bi-directional flow capability, ability to self-prime, pulseless flow, low NPSHR (net positive suction head required), high MTBM (mean time between maintenance), high-pressure and high-temperature capability, precise and accurate metering, and availability in multiple seal configurations or sealless mag-drives. **Liquiflo has over 35 years experience in designing and manufacturing quality high-alloy gear pumps for the chemical processing industry, and extensive experience in pumping acids, caustics, solvents, polymers and other types of chemicals.** Liquiflo's experienced engineers are available to assist you with your special chemical pumping applications.

## MAXIMUM TORQUE FOR VARIOUS GEAR COMBINATIONS (inch-pounds)

The following chart shows the maximum amount of torque that gears made of various materials can safely withstand. This becomes important when selecting appropriate gear materials for a specific application. The amount of torque required is a function of both pressure and viscosity of the liquid being pumped.

Drive/Idler Gear Materials	Pump Models											
	H1F 31F 41	H3F 33F 43	H5R 35R 44	H5F 35F 45	H7N	H7R 37R	H7F 37F	H9R 39R	H9F 39F	H12R 312R	H12F 312F	314F
Double Metal	23	23	74	74	134	189	189	189	189	1558	1558	1558
Idler Gear CARBON	3	9	15	21	28	45	56	79	113	210	280	560
Idler Gear CELCON	4	11	19	26	36	57	71	99	142	264	352	704
Idler Gear PEEK	4	12	20	28	38	61	76	107	153	285	380	760
Idler Gear RYTON	4	12	20	28	38	61	76	107	153	285	380	760
Idler Gear TEFLON	–	4	7	10	14	22	27	38	54	101	134	268

## THEORETICAL DISPLACEMENTS (zero slip)

The following chart shows the theoretical displacement (volume per revolution) for the various sizes of Liquiflo gear pumps. These theoretical values are based on a new pump operating with zero slip. In practice, the displacement will be less. This is due to the amount of slip in the pump which is a function of fluid viscosity, differential pressure, and the working clearances inside the pump.

H - Series	Pump Models											
	H1F 31F 41	H3F 33F 43	H5R 35R 44	H5F 35F 45	H7N	H7R 37R	H7F 37F	H9R 39R	H9F 39F	H12R 312R	H12F 312F	314F
3 - Series	31F	33F	35R	35F		37R	37F	39R	39F	312R	312F	314F
4 - Series	41	43	44	45								
Nominal Gear Length (in)	.125	.375	.625	.875	.625	1.000	1.250	1.750	2.500	1.500	2.000	4.000*
Displacement (cc/rev)	1.04	3.13	5.22	7.31	11.62	18.59	23.24	32.59	46.48	72.19	96.26	192.5
Displacement (in <sup>3</sup> /rev)	0.0638	0.1913	0.3185	0.4458	0.7092	1.1342	1.4183	1.9889	2.8367	4.4052	5.8743	11.749
Displacement (gal/rev)	.000276	.000828	.00138	.00193	.00307	.00491	.00614	.00861	.01228	.01907	.02543	.05086
Max Speed (RPM)	1750	1750	1750	1750	1750	1750	1750	1750	1750	1150	1150	1150
Max Flow Rate (GPM)	0.483	1.45	2.41	3.38	5.37	8.59	10.75	15.07	21.49	21.93	29.24	58.49

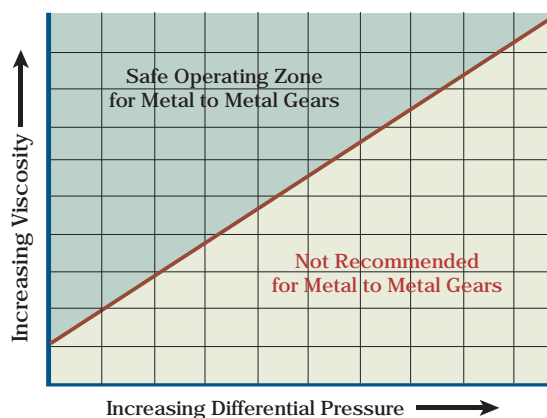
Notes: 1. Theoretical Displacements and Flow Rates @  $\Delta P = 0$  and Max Speed.  
2. Theoretical Displacements for Max-Series pumps are given on Page 66.

\* Two 312F Gears

## METAL vs. METAL GEARS: MINIMUM VISCOSITY vs. PRESSURE

The ability to operate a gear pump using both a metal drive gear and metal idler gear is strongly dependent on both the differential pressure and the viscosity of the fluid being pumped. When pumping fluids such as oil that have a sufficiently high viscosity and good lubricating properties, you can expect significant operating life when using metal to metal gears. Certain polymers can also be pumped when using metal to metal gears. When pumping low viscosity fluids that do not have lubricating properties, a non-metallic idler gear must be used.

The graph shows the effect that viscosity and pressure have on the operating zone for metal vs. metal gears. Contact factory for specific recommendations.





**PUMPING HOT LIQUIDS**

Hot liquids affect expansions in the pump and (most often) have lower viscosities as compared to cooler temperatures. Gear pumps with seals are limited to about 500 °F (260 °C) and the range of internal materials at this temperature level is limited to metallic or carbon gears, carbon or SiC wear plates, bearings and seal faces, and graphoil wedges or packing in the seals. At lower temperatures, a number of alternate materials can be used as shown in the Maximum Use Temperatures chart below.

Liquiflo's pump specification procedure analyzes the temperature conditions and determines what effects, if any, are relevant. Plastic gears are trimmed in length and diameter to accommodate thermal expansion. Under certain circumstances bearings will be trimmed as well.

**MAX USE TEMPERATURES (°F) FOR NON-METALLIC COMPONENTS IN GEAR PUMPS**

MATERIALS	GEAR	WEAR PLATE	BEARING	COATED SHAFT	O-RING/GASKET	SEAL FACE	SEAL WEDGE/PACKING
Teflon	230	260	180	NA	500	180	350
Ryton	300	350	250	NA	NA	NA	NA
PEEK	400	450	350	NA	NA	NA	NA
Carbon	500	500	500	NA	NA	500	NA
Silicon Carbide (SiC)	NA	500	500	NA	NA	500	NA
Graphoil	NA	NA	NA	NA	500	NA	500
Chrome Oxide (CO)	NA	NA	NA	250	NA	NA	NA
Tungsten Carbide (TC)	NA	NA	NA	500	NA	NA	NA

**PUMPING VISCOUS LIQUIDS**

Gear pumps excel at pumping moderately viscous and viscous liquids. Horsepower shown on Liquiflo's performance curves are based on the application's brake horsepower (BHP). The power requirements for viscous liquids can be significantly reduced (by as much as sixty percent) by trimming the gears (to increase clearances which reduce drag) and by lowering the RPM. By selecting a larger pump and running it slower, the savings in power consumption can outweigh the additional cost of the larger pump. In addition, operating a larger pump at a slower speed will increase the life span of the pump.

**PUMPING THIN LIQUIDS**

Frequently, gear pumps are the preferred solution to low-viscosity pumping applications because of their hydraulics (low flow, high pressure and pulseless flow), compactness, efficiency and low cost. However, thin fluids affect gear pump wear characteristics and slip. These considerations are addressed in Liquiflo's pump designs. Viscosities as low as 0.3 CPS have been satisfactorily pumped.

Thin liquids prevent hydrodynamic film formation. For gear pumps, this means that the bearing-to-shaft or gear-to-gear interfaces may have poor lubrication causing accelerated wear. Liquiflo has reduced the problems associated with thin fluids for many applications by providing parts specifically designed to minimize the types of wear associated with poor film lubrication. Tungsten Carbide shafts, Silicon Carbide bearings and specially designed front and rear housings, made to enhance journal bearing lubrication, are all significant aids.

Slip is also a factor when pumping thin fluids with gear pumps. Slip is a function of viscosity, differential pressure, and the specific pump design. In order to generate the specified flow at a given differential pressure, thin liquids require a higher RPM than viscous liquids.

## MAGNETIC COUPLINGS

Magnetically-coupled pumps can be used as an economical alternative to pumps with double mechanical seals. Mag-drive pumps should also be considered when pumping hazardous, volatile or toxic chemicals, or chemicals whose characteristics will change upon contact with water or air.

When selecting a magnetically-coupled pump, it is necessary to determine whether the pump will deliver the flow required and the magnetic coupling has sufficient torque transmission capability to accommodate the application. Magnet size requirements are a function of the application's brake horsepower, the RPM and temperature. If the coupling does not have sufficient strength, it will "break free or slip," and the pump will not rotate. Liquiflo has a large selection of magnetic couplings to accommodate applications requiring higher temperatures or greater torque transmissions.

Maximum Torque Transmission for Various Magnetic Couplings							
Magnetic Coupling	Torque (in-lbs)	Magnetic Material	Used for Pump Models	Magnetic Coupling	Torque (in-lbs)	Magnetic Material	Used for Pump Models
MCX	10	SmCo	2R/2F	MCD	33	Ceramic	620
MCN	20	NdFeB	41-45	MCF	120	SmCo	620-622
MCR	30	SmCo	41-45	MCW	200	SmCo	621/622
MCU	75	SmCo	H1-H9R/31-39/M0-M4	MCK	325	SmCo	H12R/H12F/312-314
MCB	120	SmCo	H1-H9F/31-39/M0-M8	MCJ	650	SmCo	H12R/H12F/312-314
MCV	200	SmCo	H5-H9/35F-39/M0-M8				

## Pumping Hot Liquids with Mag-drive Pumps

The magnetically-coupled pumps have a maximum temperature utility of 575 °F. Liquiflo uses 3 types of magnetic materials in its couplings: Ceramic, Rare Earth Samarium Cobalt (SmCo) and Rare Earth Neodymium Iron Boron (NdFeB). A number of different magnet strengths are available that will supply adequate torque transmission and minimize eddy current losses.

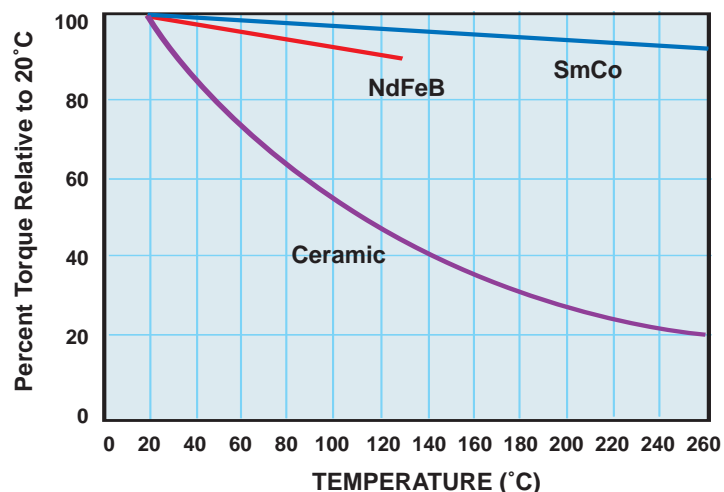
## Temperature Control Jacket (Clamp-on Design)

Liquiflo's Temperature Control Jacket can maintain the pump at either elevated or reduced temperatures. Temperature Control Jackets are commonly used when pumping liquids that solidify or become viscous when the temperature decreases. Either steam or a liquid at elevated temperature can be circulated through the heat jacket to maintain the fluid being pumped at a specified elevated temperature.

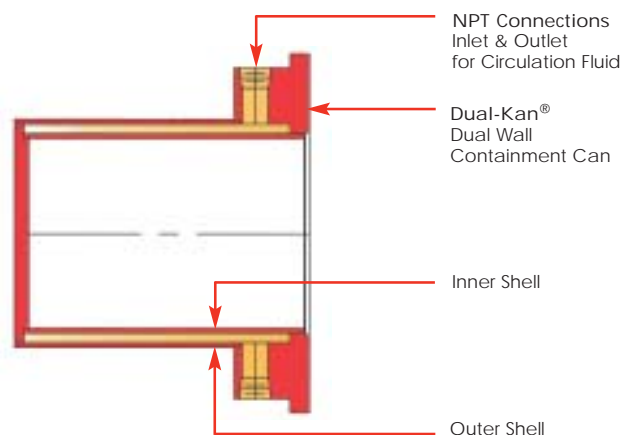
## Dual-Kan® (Double Wall Containment Can)

Liquiflo's patented Dual-Kan® controls the temperature of the back end of a magnetically-driven pump by circulating a heating or cooling fluid through the containment can. This is ideal for applications requiring precise and uniform temperature control. The Dual-Kan® will compensate for increases in temperature due to eddy current losses from the mag-drive coupling.

## TORQUE RETENTION vs. TEMPERATURE for MAGNETIC COUPLINGS



## DUAL-KAN® (Double Wall Containment Can)



## MAGNETIC DRIVES

The most important advantage of the permanent magnet drive is that it makes possible the transmission of torque through a barrier without the use of any mechanical connection. Because these drive couplings are entirely magnetic, this completely eliminates seals, along with their associated leakage, maintenance and contamination problems. The type of magnetic drive used in Liquiflo pumps are referred to as *synchronous* because the inner and outer magnets rotate at the exact same speed.

Radial type synchronous drives (which are the type used for Liquiflo Pumps) consist of an inner and outer magnet assembly magnetized with multiple poles about their respective inside and outside diameters. The radial type drive has no axial component of thrust but does require tight tolerance to minimize the radial gap between the inner and outer rotating members. When the maximum torque of a synchronous drive is exceeded, the follower, or driven member, will fall out of step and stop rotating. The load cannot be picked up again unless the driving member (the outer magnet which is connected to the motor), is first stopped and then restarted. This can be an important and useful feature if an over-pressure condition occurs in a system.

The containment shell for mag-drives pumps can be made from a conducting material; however, it is usually non-magnetic. Eddy current losses will occur if the containment shell is conductive and these losses are inversely proportional to the resistivity of the barrier shell.

Typical design requirements that are considered with permanent magnet drives are:

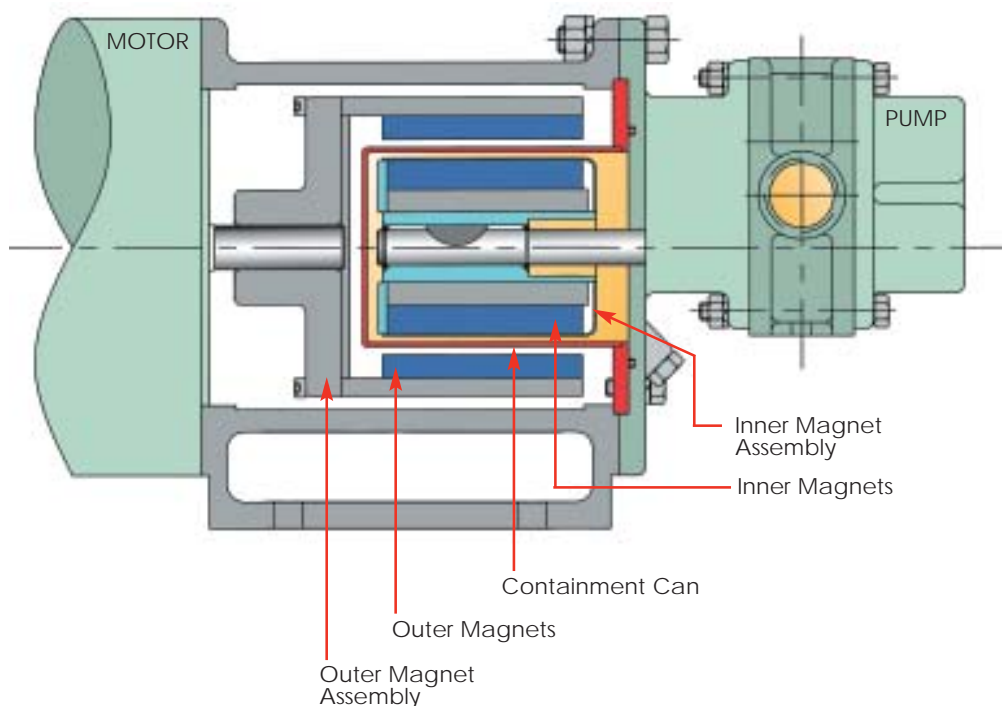
- (A) maximum torque to be transmitted
- (B) speed at which the torque is to be transmitted
- (C) the gap and material through which the torque is to be transmitted
- (D) the temperature at which the unit will operate
- (E) any physical size limitations of the coupling

### The 4 Golden Rules of Magnetic Drives

- 1 Power losses are directly proportional to the *square* of the speed (n).
- 2 Power losses are directly proportional to the *square* of the drive radius (r).
- 3 Power losses are directly proportional to the *square* of the flux density (B).
- 4 Power losses are inversely proportional to the resistivity ( $\rho$ ) of the barrier material.

$$P_{\text{Losses}} \propto \frac{n^2 r^2 B^2}{\rho}$$

## MAG-DRIVE PUMP, CLOSE-COUPLED



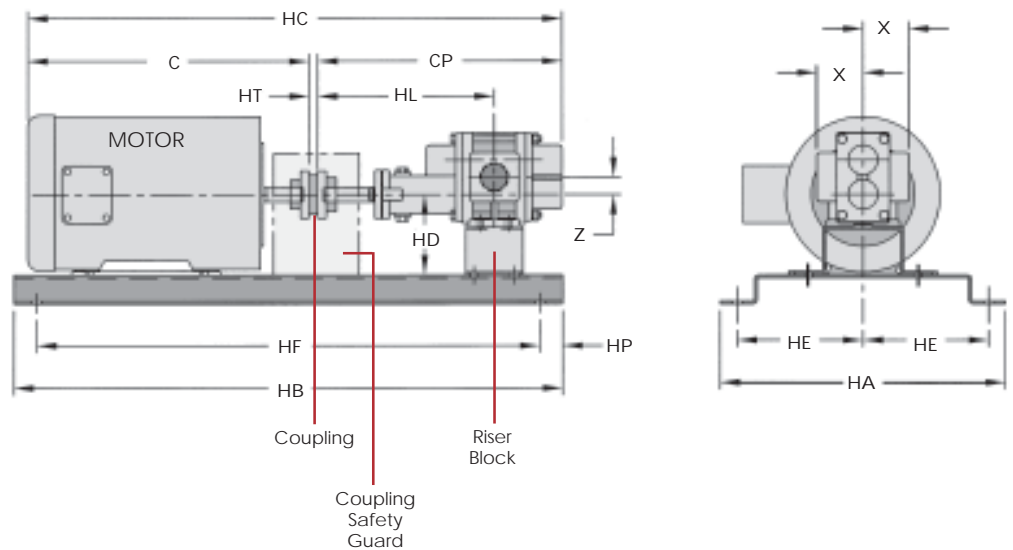


## PUMP-MOTOR COUPLING & MOUNTING OPTIONS

### Base Mounted Pump

Dimensions and nomenclature conform to Hydraulic Institute (HI) standards. Bases are made from stainless steel or from steel channel painted for corrosion resistance. These units can be assembled at the factory. This eliminates having to mount units in the field which may be difficult or inconvenient without proper alignment instrumentation.

Coupling safety guards are available to help protect personnel from the rotating elements of the pump and motor.

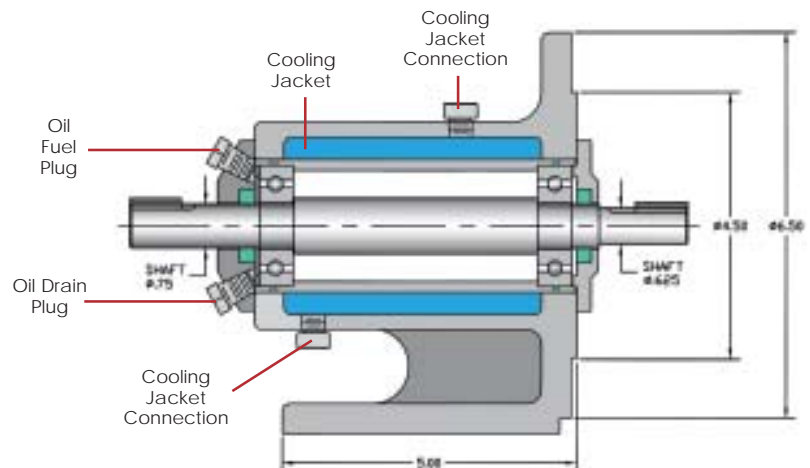


### Pedestal/Power Frame

This is the standard pedestal used for **long-coupling** the Liquiflo Centry-Series Centrifugal Pumps. The Liquiflo Power Frame is also used for long-coupling the Liquiflo H-Series and 3-Series Gear Pumps.

#### High-Temperature Applications:

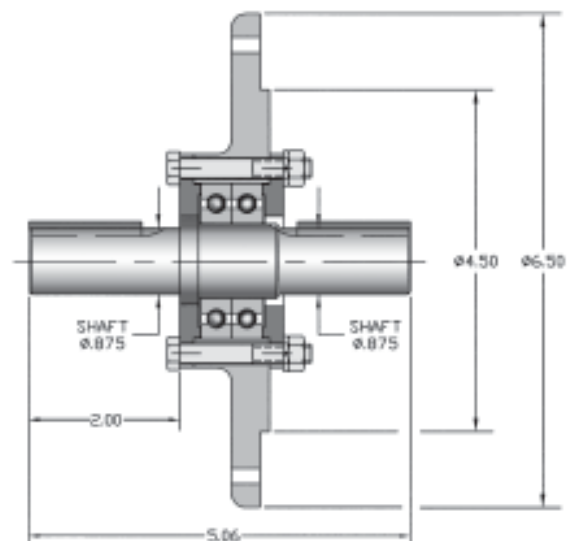
A key feature of the Liquiflo Power Frame is its **integral cooling jacket** which keeps the bearing system of the pedestal cool even when it is coupled to a pump operating at extremely high temperature.



### S-Adapter

The Liquiflo S-Adapter is used in conjunction with the MC-Pedestal Bracket for **long-coupling mag-drive pumps**. It is also used when long-coupling pumps to motors or drives that do not have a C-face mounting. Typical examples are air motors, gear reducers, or specialty motors that are not available with C-face mounting.

The temperature rating of the S-Adapter is 250°F (~120°C).



## RELIEF VALVES

### Protection of Pump from Over-Pressure Condition

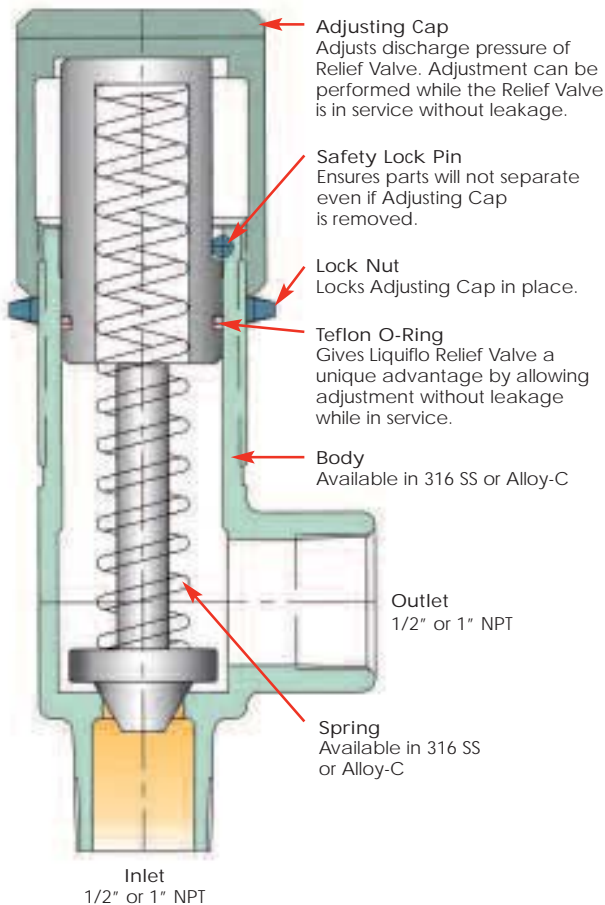
Positive Displacement Rotary Pumps should always be installed with a **Relief Valve** in the discharge line, to protect the pump and piping against any type of line blockage, including the inadvertent closing of an isolation valve.

Liquiflo manufactures two sizes of Relief Valves in **316 SS** and **Alloy-C**. Ten standard Relief Valve models are available depending on the body material, the port size and the discharge pressure range (see table below).

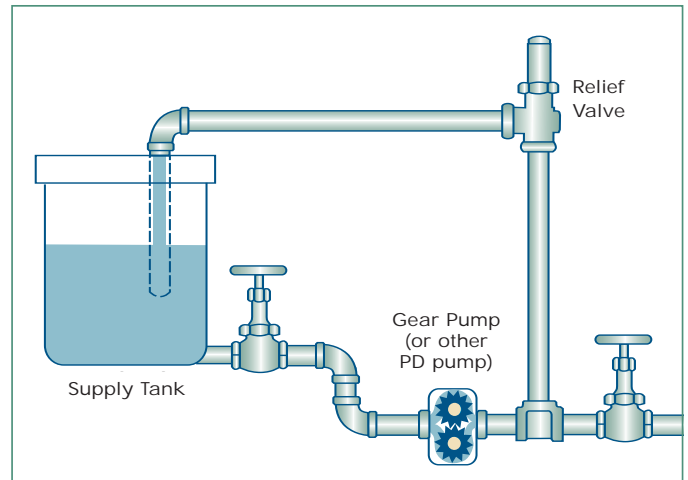
### Liquiflo Standard Relief Valves

Model #	Material	Port Size & Type	Pressure Setting (PSI)		Approximate Flow at 25% above Pressure Setting	
			Min	Max	GPM	LPM
RV1000-LP	316 SS	1/2" NPT	25	65	7.5	28
RV1000	316 SS	1/2" NPT	50	135	7.5	28
RV1000-HP	316 SS	1/2" NPT	75	200	7.5	28
RV1001-LP	Alloy-C	1/2" NPT	25	65	7.5	28
RV1001	Alloy-C	1/2" NPT	50	135	7.5	28
RV1001-HP	Alloy-C	1/2" NPT	75	200	7.5	28
RV2000-LP	316 SS	1" NPT	25	75	25	95
RV2000	316 SS	1" NPT	50	175	25	95
RV2001-LP	Alloy-C	1" NPT	25	75	25	95
RV2001	Alloy-C	1" NPT	50	175	25	95

### RELIEF VALVE CROSS-SECTION



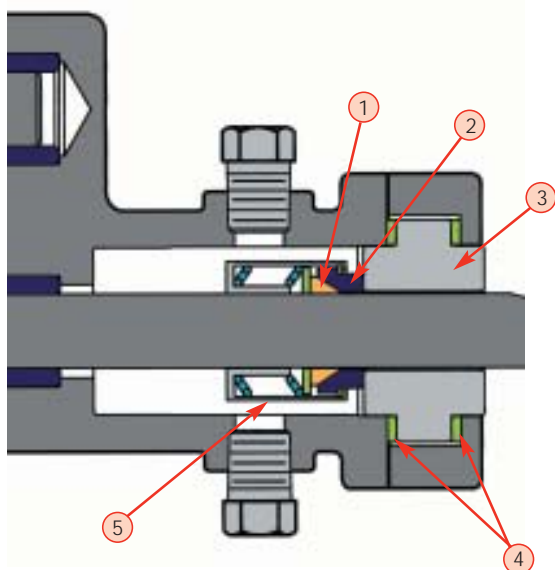
### PROPER HOOK-UP OF PUMP WITH SEPARATE RELIEF VALVE



Proper configuration with Relief Valve directly bypassing the discharge line back to the supply tank.

## SEAL ARRANGEMENTS FOR GEAR PUMPS

Single Mechanical Seal - Internally Mounted

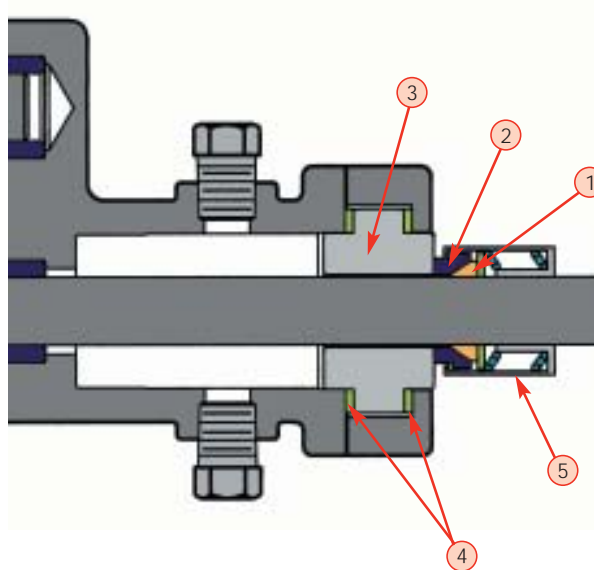


- ① Seal Wedge
- ② Rotating Seal Face
- ③ Stationary Seal Seat
- ④ Gaskets for Seal Seat
- ⑤ Mechanical Seal Body

## Single Internal Mechanical Seal

The Single Mechanical Seal arrangement is the standard and is by far the most commonly used when pumping any type of chemical where leakage needs to be kept to a minimum. This seal type can tolerate limited amounts of abrasive particles. The maximum fluid viscosity recommended is 5000 CPS.

Single Mechanical Seal - Externally Mounted



- ① Seal Wedge
- ② Rotating Seal Face
- ③ Stationary Seal Seat
- ④ Gaskets for Seal Seat
- ⑤ Mechanical Seal Body

## Single External Mechanical Seal

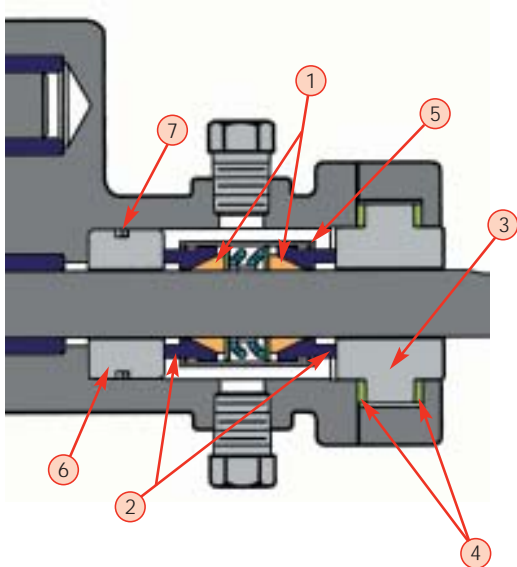
The External Mechanical Seal arrangement is used when system pressures are below atmospheric pressure (vacuum). This is because the atmospheric pressure will assist in holding the seal faces together. This seal arrangement isolates the mechanical seal from the fluid being pumped which can be an advantage when pumping chemicals that are not compatible with the seal body. The maximum fluid viscosity recommended is 5000 CPS.

## Seal Materials Available

PACKING	SEAL WEDGES	SEAL FACES	SEAL SEATS
<b>Braided Teflon</b> <ul style="list-style-type: none"> <li>Extremely chemically resistant – primary choice</li> </ul> <b>Graphoil</b> <ul style="list-style-type: none"> <li>For high temperature use – over 350 °F up to 500 °F</li> </ul>	<b>Teflon</b> <ul style="list-style-type: none"> <li>Extremely chemically resistant – primary choice</li> </ul> <b>Graphoil</b> <ul style="list-style-type: none"> <li>For high temperature use – over 350 °F up to 500 °F</li> </ul>	<b>Carbon</b> <ul style="list-style-type: none"> <li>Resistant to majority of chemicals – primary choice</li> </ul> <b>Teflon</b> <ul style="list-style-type: none"> <li>Extremely chemically resistant</li> <li>Pressure limited to 50 PSI</li> <li>Not abrasive resistant</li> </ul> <b>Silicon Carbide</b> <ul style="list-style-type: none"> <li>Extremely chemically resistant</li> <li>High abrasion resistance</li> </ul>	<b>Silicon Carbide</b> <ul style="list-style-type: none"> <li>Extremely chemically resistant – primary choice</li> <li>High abrasion resistance</li> </ul>

## SEAL ARRANGEMENTS FOR GEAR PUMPS *(continued)*

### Double Mechanical Seal

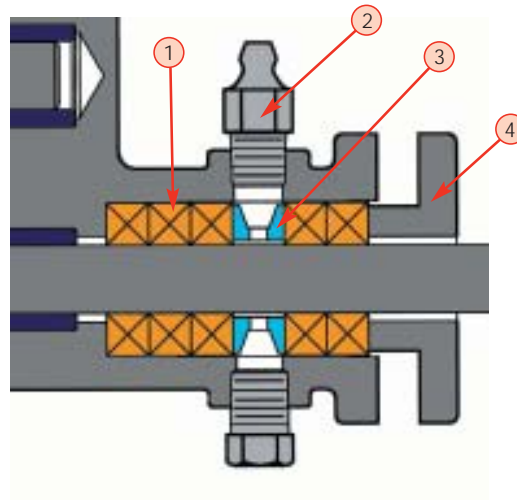


- ① Seal Wedges
- ② Rotating Seal Face
- ③ Outboard Seal Seat
- ④ Gaskets for Seal Seat
- ⑤ Mechanical Seal Body
- ⑥ Inboard Seal Seat
- ⑦ O-Ring

### Double Mechanical Seal

The Double Mechanical Seal arrangement relies on a lubricating system to cool the seal faces. The lubricating barrier liquid washes the seal faces and must be safe and compatible with the pumpage. Accordingly, double seals are preferred for crystallizing or abrasive liquids as the seal faces are only exposed to the flush liquid. Double seals must be pressurized so that the seal chamber has about 15 PSI higher pressure than the pump discharge pressure.

### Packing



- ① Packing
- ② Grease Fitting
- ③ Lantern Ring
- ④ Gland Plate

### Packing

Although Packing is still used, it is not very common in the chemical processing industry. It is still considered to be an acceptable solution when pumping safe liquids or where the seal drainage can be captured. Flocculants, water, and caustics are common examples of fluids which use this type of seal. Teflon is the standard packing material. Graphoil packing is used for high-temperature applications. A leakage rate of 8-10 drops per minute is required to lubricate the packing-shaft interface.

### High Temperature Fluids

When pumping liquids over 200 °F, seal materials that can withstand elevated temperatures must be considered. (See the charts on pages 91, 96 & 98 for recommended materials.) The maximum recommended operating temperature for mechanical seals is 500 °F.

### Abrasive Fluids

In general, it is recommended to use a double mechanical seal arrangement with a compatible fluid on this type of application. Mildly abrasive fluids can often be handled with internal mechanical seals using the appropriate material selections. When pumping fluids that contain abrasive materials in suspension, accelerated wear of seal faces and seal seats will occur. In these instances, seal face and seal seat combinations designed to resist abrasion must be used. The proper selection of material combination will depend upon the nature and hardness of the abrasive particles.

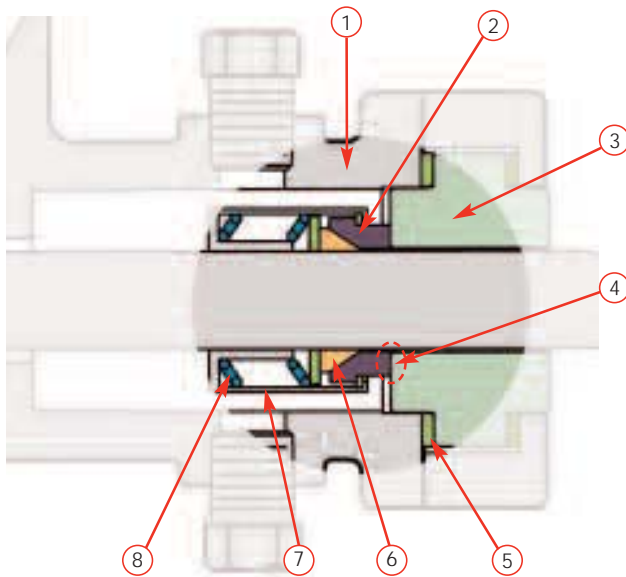
### Crystallizing Fluids

When pumping fluids which crystallize on contact with air, or water vapor that is often present in air, a pump with a double mechanical seal with a compatible barrier fluid or a Mag-drive type pump should be used. Crystallizing liquids have a tendency to build up around the edges of single mechanical seals and cause premature seal failure.

### Viscous Fluids

Single seals have a viscosity limit of approximately 5000 CPS. Double mechanical seals can pump liquids with a substantially higher viscosity when using thinner barrier liquids. Packing can also be used when pumping higher viscosity liquids.

## MECHANICAL SEAL CONFIGURATIONS



- ① **Universal Seal Housing** will accommodate single, double or external mechanical seals, as well as packing.
- ② **Rotating Seal Face** made from Carbon, Teflon, or Silicon Carbide, with flat lapped surface.
- ③ **Stationary Seal Seat** made from Silicon Carbide.
- ④ **Mating area** for rotating Seal Face and stationary Seal Seat.
- ⑤ **Gasket** which seals stationary Seal Seat to the housing.
- ⑥ Teflon or Graphoil **Wedge** forms a tight seal around the shaft and up against the back of the Seal Face. Spring pressure maintains constant force to tighten the Wedge around the Shaft and up against the Seal Face.
- ⑦ **Mechanical Seal Retainer** normally made from the same material as the pump housing (316 SS or Alloy-C).
- ⑧ **Springs** made from same material as the mechanical seal retainer. Springs apply constant pressure to ensure proper sealing between the static Seal Seat and rotating Seal Face.

## INTRODUCTION

The choice of seal arrangement (single, double or external mechanical seal) in a chemical application is perhaps one of the most important decisions that need to be made. Seals come in a variety of forms and types and Liquiflo's [Universal Seal Housings](#) are designed to accommodate all of them. The wide choice of seal materials and seal arrangements will allow these pumps to operate in a variety of difficult applications, including high temperature, abrasive, or viscous service.

## Material Considerations for Mechanical Seals

Mechanical seals are used in a wide variety of applications. Therefore, care needs to be taken to select materials compatible with the fluid being pumped. The corrosive properties, the lubricity, and the temperature of the fluid must be considered. The selection of the mating seal seat and seal face is one of the most important steps to ensure a properly operating and cost-effective mechanical seal arrangement.

The ideal material should be hard (an exception to this is Teflon), corrosion resistant and capable of accepting a very flat finish. Most face materials exhibit poor wear characteristics when in frictional contact with another surface of the same material. Therefore, two dissimilar materials, one softer than the other, are usually selected. In extreme abrasive or corrosive environments, it is sometimes preferable to mate surfaces of identical materials, but this is practical only with extremely hard materials such as Tungsten Carbide or Silicon Carbide. In these cases, the liquid being pumped or the liquid being circulated through the seal chamber must have adequate lubricity to prevent heat-checking of the faces.

## Material Characteristics

Silicon Carbide	Silicon Carbide retains its strength at elevated temperatures, has a low thermal expansion rate, a high thermal conductivity, and excellent corrosion resistance. These properties make it an ideal material for mechanical seal faces, especially for applications in abrasive or corrosive environments such as nitric acid, hydrofluoric acid, or sodium hypochlorite.
Carbon	Carbon is most frequently used as a rotating seal face material. Carbon has excellent anti-friction qualities and is compatible with a wide range of temperatures and corrosive environments.
Teflon	Teflon is extremely chemically resistant to a wide range of corrosive chemicals. It is most often the choice for the seal wedge material unless temperature limitations prevent its use. For certain chemical applications where Carbon is not compatible, it is used for the rotating seal face. Due to physical limitations when used in sealing applications, the maximum pressure must be limited to 50 PSI.
Graphoil	For high temperature applications in excess of 350 °F, Graphoil should be used instead of Teflon for the seal wedge. The maximum operating temperature when used as a seal wedge is 500 °F.

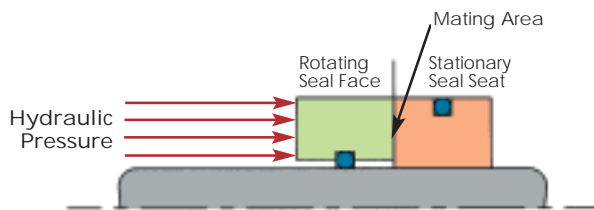


## UNBALANCED, SEMI-BALANCED & BALANCED SEALS

### Unbalanced, Semi-Balanced & Balanced Seals

Liquiflo normally uses unbalanced seals which are considered acceptable up to pressures of 350 PSI. When higher pressures or difficult applications that contain abrasives are encountered, it may be required to use semi-balanced or balanced seal configurations. These serve to reduce frictional forces in the mating area of the seal.

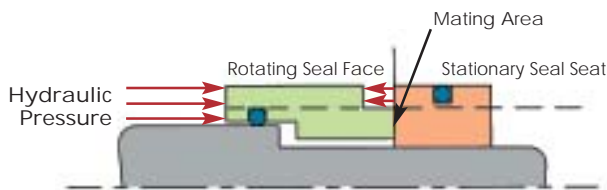
#### UNBALANCED SEAL



##### Unbalanced Seal

When unbalanced seals are mounted internally to the pump, the System Pressure assists the seal springs in keeping the seal faces closed. This is the most common seal configuration for Liquiflo gear and centrifugal pumps. Unbalanced seals can be used for pressures as high as 350 PSI; however, this depends on the type and nature of the fluid being pumped. When unbalanced seals are mounted externally to the pump, the System Pressure will have a tendency to open the seal faces.

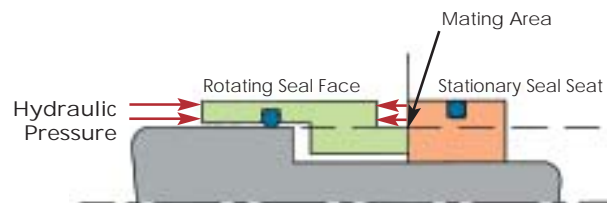
#### SEMI-BALANCED SEAL



##### Semi-Balanced Seal

Semi-balanced seals require a step shaft. They reduce the loading on the seal face/seal seat mating area, allowing higher System Pressure without exceeding the PV limit of the seal face combination. Cartridge style seals are available that incorporate a semi-balanced design.

#### BALANCED SEAL



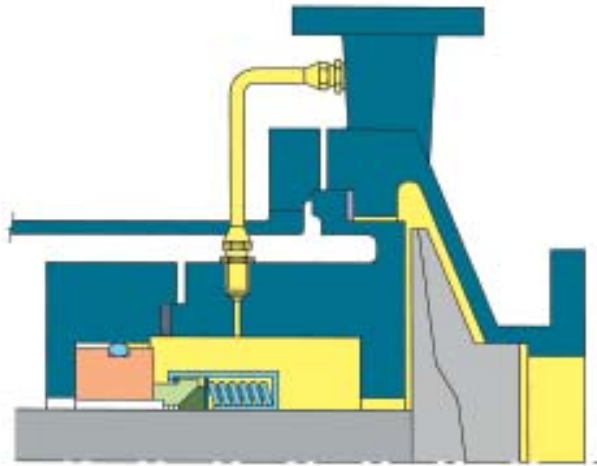
##### Balanced Seal

With fully-balanced seals, the spring pressure of the seal is the only force that holds the seal faces together. Fully-balanced seals are used in situations with extremely high System Pressure.



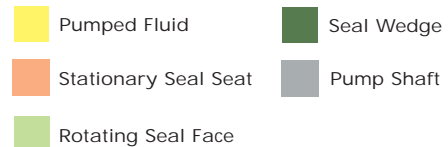
## SEAL ARRANGEMENTS FOR CENTRIFUGAL PUMPS

## Single Seal – Internally Mounted

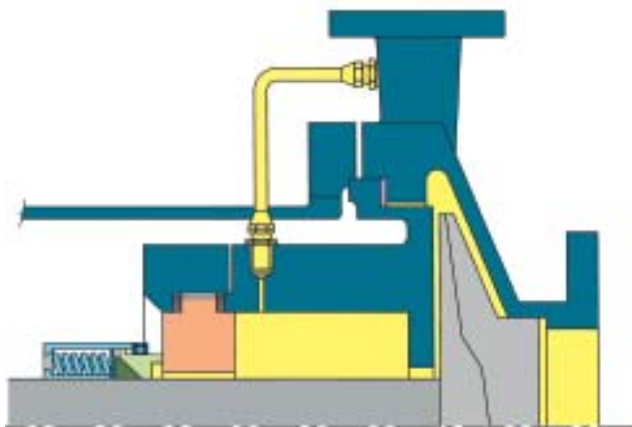


## Single Seal, Internal

Cooling and lubrication of a single inside-mounted mechanical seal is provided by piping from the discharge or high-pressure side of the pump to the seal chamber. This arrangement is an excellent choice because the liquid passing through the equipment can provide proper cooling and lubrication for the mechanical seal. A seal in this position automatically utilizes the pumping pressure to help it accomplish its sealing job. Consequently, it has the best reliability factor because the seal is cooled and lubricated when the equipment is in operation.

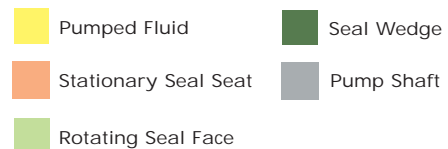


## Single Seal – Externally Mounted

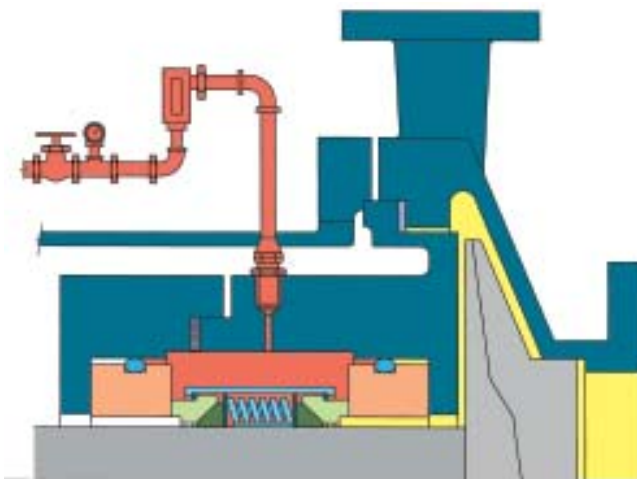


## Single Seal, External

An externally mounted mechanical seal requires cooling and lubrication flow through the seal chamber. This is accomplished with a seal flush line, generally from the high-pressure side to the low-pressure side of the pump, introduced to the seal chamber as close to the sealing faces as practical. An externally mounted seal does not have the same reliability of a single internally mounted seal because the pump pressure is working against it.

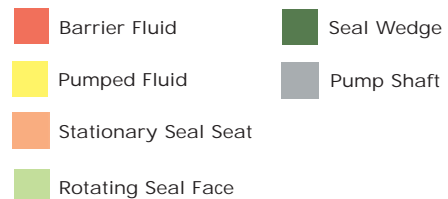


## Double Seal Arrangement



## Double Seal Arrangement

This is the most popular type of multiple seal arrangement. Two seals oriented back-to-back provide a closed area in which a proper seal lubricant and coolant is circulated at a pressure generally 5 to 20 psi above the pressure at the throat of the seal chamber. The barrier liquid is cooled by an external source.



## METERING WITH GEAR PUMPS

The numerous advantages of using gear pumps have made them a popular pump type when flows need to be metered or totaled. These advantages over other types of pumps are:

#### Virtually no pulsations

This allows flows to be easily and accurately measured with standard flow meters. Pipe diameters can be much smaller in relation to those used with pulsing diaphragm pumps whose pipe diameters are based on high instantaneous flow rates.

#### Lower NPSH requirements

Gear pumps require less NPSH because of their steady non-pulsating flow characteristics. Diaphragm or piston style pumps require higher NPSH because of their high instantaneous flow rates. NPSHA decreases further as the viscosity of the fluid increases.

#### Simplicity of design

Gear pumps are extremely simple to understand and maintain. There are no check valves that can become clogged or fail, and pulsation dampeners are not required.

#### Sealless design

Mag-drive gear pumps eliminate the leakage associated with mechanically sealed pumps. There are no diaphragms to rupture allowing process fluid to leak or destroy the pump, or gear box lubrication oil to enter the process.

#### High accuracy

With standard instrumentation, such as flow meters, pH sensors, and variable speed drives, accuracies of 0.5% are easily achievable.

#### Low cost

The cost to automate and maintain more efficient rotary gear pumps has decreased significantly. Metering using gear pumps is now considered to be one of the most cost-efficient methods in industry today.

Electronic instrumentation has made metering with gear pumps a very simple and extremely accurate solution. Three of the most common types of metering applications are:

- 1) Maintaining a constant flow rate regardless of differential pressure or fluid viscosity.
- 2) Delivering a variable flow of chemical to maintain a system parameter (e.g., pH value).
- 3) Delivering a given amount of fluid. (This is normally found in batch processing operations.)

#### Operation Without Feedback

When operating without feedback control, the metering capability with either a gear pump or diaphragm pump is a function of many variables. The differential pressure affects both slip in the pump and motor speed — both of which affect the flow rate. The differential pressure in turn is a function of viscosity and density, which are primarily functions of temperature. Viscosity, in addition to affecting the differential pressure, also independently affects slip.

The impact of these factors is much more pronounced in the gear pump when operating on thinner fluids (i.e., the gear pump is not as “pressure stiff” as diaphragm pumps in the lower range of viscosities). However, at a viscosity of around 50 cps or higher, gear pumps’ volumetric efficiency and pressure stiffness increases to rival that of diaphragm pumps. Diaphragm pumps, on the other hand, start to become less pressure stiff at the higher viscosities due to the delay in reseating of the check valves and the high instantaneous flow rates associated with pulsing type pumps.

#### Operation Using Feedback Control

Economical closed-loop feedback systems have given gear pumps the ability to perform as metering pumps at the lower end of the viscosity range. The burden for accuracy and repeatability has been moved from the pump to the instrumentation. “Pressure stiffness” is no longer the primary factor. Flow meters with analog outputs capable of down to 0.25% accuracy coupled with PID controls allow gear pumps to achieve accuracies in the range of 0.5% to 2.0%. In addition, tremendous adjustability is available with gear pumps when matched with DC motors coupled with SCR controllers capable of 20:1 turn downs or AC inverter duty motors coupled with VFD controllers capable of 1000:1 turn downs. See Figure 1 and Figure 2 for examples of gear pumps in metering applications using feedback control.

Figure 1

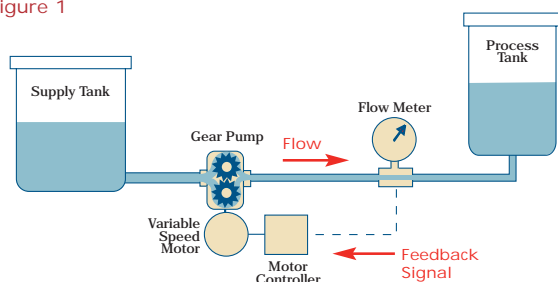


Figure 1 shows a schematic hook-up of a metering application precisely controlling the flow rate of a liquid into a process. A 4-20 mA signal coming from the flow meter controls the speed of the pump to accurately administer a designated flow rate into a process. The system is set up to maintain an exact flow rate that is continually being monitored by the flow meter. The signal from the flow meter will either speed up or slow down the motor speed to maintain this flow rate value. Either DC motors with SCR (Silicon Controlled Rectifier) controllers or AC duty motors with VFD (Variable Frequency Drive) controllers are commonly used.

Figure 2

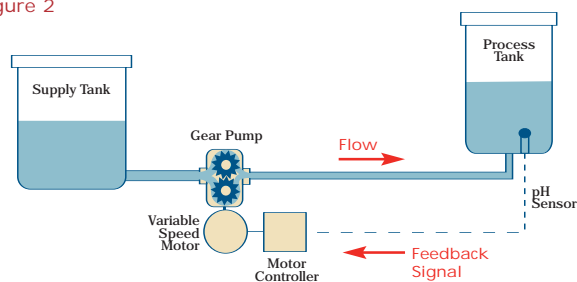


Figure 2 shows a schematic hook-up of a metering application precisely controlling the pH range of a process. Depending on the condition of the process a varying amount of solution needs to be added to maintain the process at the desired pH level. The system is set up to maintain this exact pH level and is continually being monitored by the pH sensor. The signal from the pH sensor will either speed up or slow down the motor speed to maintain the desired pH value.

## CHEMICAL CLASSIFICATIONS

Pump engineering requires a thorough knowledge of the fluid's chemical and physical properties, and hazards. Although there are literally millions of unique chemicals, most liquids can be classified as shown below. Some chemicals can be included in more than one class or group. For example, a chemical can be a solvent, a thin liquid, a volatile liquid, a flammable liquid and a Newtonian fluid. Typical examples of chemicals or fluid materials are included for each of the given groups.

## CHEMICAL CLASSES

**Acids** – Chemicals producing pH < 7 when in aqueous solution (e.g., acetic, benzoic, boric, carbonic, carboxylic, chlorosulfonic, chromic, citric, fluorosilicic, hydrobromic, hydrochloric, hydrofluoric, lactic, maleic, nitric, nitrous, oleic, oxalic, perchloric, phosphoric, phthalic, sulfuric, sulfurous, toluenesulphonic)

**Caustics** – Chemicals producing pH > 7 when in aqueous solution (e.g., ammonium hydroxide, barium hydroxide, calcium hydroxide, calcium hypochlorite, potassium hydroxide, sodium hydroxide, sodium hypochlorite)

**Solvents** – Organic or inorganic liquids capable of dissolving chemicals (e.g., acetaldehyde, acetone, acetonitrile, amyl acetate, aniline, benzene, benzyl alcohol, butyl acetate, butyl alcohol, carbon disulfide, carbon tetrachloride, chlorobenzene, chloroform, cyclohexane, cyclohexanol, diethylamine, diethyl ether, dioctyl phthalate, ethanol, ethyl acetate, ethylene glycol, heptane, hexane, isopropyl alcohol, methanol, methyl ethyl ketone, methylene chloride, octane, pentane, perchloroethylene, propyl alcohol, propylene glycol, pyridine, tetrahydrofuran, toluene, trichloroethane, trichloroethylene, water, xylene)

**Inorganic Salt Solutions** – Typically aqueous solutions with corrosive properties (e.g., aluminum chloride, aluminum potassium sulfate, aluminum sulfate, ammonium chloride, ammonium nitrate, ammonium sulfate, brine, calcium chloride, copper sulfate, ferric chloride, ferric sulfate, potassium nitrate, potassium permanganate, silver nitrate, sodium bicarbonate, sodium bisulfite, sodium chlorate, sodium chloride, sodium silicate, sodium sulfide)

**Organic Solutions** – Organic compounds dissolved in water or other solvents (e.g., alcohols, esters, ethylene glycol, flocculants, food dyes, formaldehyde, glucose, glycerin, phenol, propylene glycol, polymers, pyridine, surfactants, urea, vinegar)

**Homogeneous Mixtures** – Uniform fluids comprised of two or more chemicals (e.g., aqua regia, asphalt, bitumen, colloids, crude oil, diesel fuel, Dowtherm, emulsions, fatty acids, formulations, fuel oil, gasoline, hydraulic fluid, jet fuel, kerosene, mineral oil, naphtha, oleum, paraffin wax, petroleum, tall oil, tallow)

**Monomers** – Chemicals capable of being polymerized (e.g., ethylene, methylene diphenyl diisocyanate (MDI), phenylene sulfide, propylene, tetrafluoroethylene, toluene diisocyanate (TDI), vinyl chloride)

## PHYSICAL CLASSES

**Low-Viscosity Liquids** have viscosities of ~0.3 cP to < 10 cP (e.g., acetone, benzene, gasoline, hydrogen peroxide, mercury, water)

**Moderately Viscous Liquids** have viscosities of 10 to < 100 cP (e.g., 50% NaOH, conc. H<sub>2</sub>SO<sub>4</sub>, cyclohexanol, glycols, light oils)

**Viscous Liquids** have viscosities of 100 cP or higher (e.g., corn syrup, glycerin, heavy oils, lactic acid, soaps)

**Newtonian Fluids** maintain viscosity with changes in shear rate (e.g., most mineral oils, dilute solutions, solvents)

**Non-Newtonian Fluids** vary in viscosity with changes in shear rate (e.g., emulsions, ketchup, paints, printing inks)

**Abrasive Fluids** contain hard colloidal or suspended particles capable of causing wear (e.g., detergents, inks, toothpastes)

**Crystallizing Fluids** are solutions that can crystallize when exposed to air (e.g., sugar & salt solutions)

**Volatile Liquids** have low boiling points and high vapor pressures at room conditions (e.g., acetone, pentane, ether)

**Heavy Liquids** have specific gravities significantly greater than water (e.g., bromine, conc. H<sub>2</sub>SO<sub>4</sub>, mercury, lead/tin solder)

**Molten Liquids** are substances that exist as solids or extremely viscous fluids at room temperature, that have been heated to a liquid state (e.g., asphalt/bitumen, lead/tin solder, naphthalene, paraffin wax, phenol, stearic acid, sulfur, tallow)

**Liquefied Gases** require cooling below room temperature or pressurization above atmospheric (e.g., ammonia, butadiene, butane, chlorine, ethylene oxide, formaldehyde, vinyl chloride)

CHEMICAL CLASSIFICATIONS *(continued)*HAZARD  
CLASSES

**Note:** The following is a list of common hazard classes but it is not intended to be a complete list of all possible hazards. Consult the material safety data sheet (MSDS) and observe all safety precautions when working with hazardous chemicals.



**Corrosive Liquids** are capable of chemically attacking both metal and nonmetal materials, and burning skin.

**Examples:** acids & caustics, bromine, hydrazine, inorganic salt solutions.



**Toxic or Noxious Liquids** are poisonous or biologically harmful.

**Examples:** acetaldehyde, ammonium & barium hydroxides, benzene, butadiene, chlorinated solvents, chlorine, chlorosulfonic acid, chromic acid, copper sulfate, ethylene glycol, ethylene oxide, formaldehyde, hydrazine, hydrofluoric acid, MDI & TDI, methanol, nitric acid, oleum, phenol, potassium & sodium hydroxides, propanol, silver nitrate, sulfuric acid, thionyl chloride, vinyl chloride.



**Flammable Liquids** emit vapors that can be ignited in air, causing fires.

**Examples:** acetates, acetonitrile, alcohols, aldehydes, butadiene, butane, carbon disulfide, cyclohexane, diethyl amine, ethers, ethylene oxide, gasoline, hydrazine, hydrocarbons, isocyanates, JP-4 jet fuel, ketones, molten sulfur, naphtha, paraffin wax, pyridine, some chlorinated solvents, tetrahydrofuran, vinyl chloride.



**Explosive Liquids** can be highly unstable or reactive under certain conditions.

**Examples:** ammonium nitrate, hydrazine, hydrogen peroxide, nitroglycerin, organic peroxides, perchloric acid, potassium nitrate.



**Hot or Molten Liquids** are pumped at high temperatures and can cause burns to skin.

**Examples:** asphalt/bitumen, lead/tin solder, naphthalene, paraffin wax, phenol, stearic acid, sulfur, tallow.



**Cold or Cryogenic Liquids** are pumped at low temperatures and are capable of causing frostbite.

**Examples:** ammonia, butadiene, butane, chlorine, ethylene oxide, Freons, nitrogen, perfluorocarbons, vinyl chloride.



**Radioactive Liquids** contain chemicals with unstable isotopes that emit ionizing radiation.

**Examples:** Fluorine-18, Gallium-67, Indium-111, Iodine-123, Thallium-201.



## CHEMICAL RESISTANCE CHART

Notice: The Chemical Resistance Chart given on the following pages is intended as a *general guide* for rating the resistance of typical engineering materials to common industrial chemicals. *It is not intended as a guarantee of material performance.* The ratings in the chart are based on data obtained from technical publications, material manufacturers and laboratory tests. The information given in the chart should be used as a first approximation for material selection, rather than the final answer. This is because chemical effects are dependent on many

factors, which can cause chemical ratings to change within a given application. Such factors include variations in temperature, pressure and concentration, chemical combinations, impurities or filler materials, aeration, agitation and exposure time. It is recommended that pumps and materials first be tested under simulated field service conditions. *Never test or operate a pump before acceptable chemical ratings have been obtained and the proper safety precautions have been taken.*

## Interpretation of Chemical Resistance Ratings

Rating	Meaning	Corrosion Rate (CR)	Units
A	Excellent – Virtually no effect; very low corrosion rate.	CR < 0.002	in/yr
		CR < 0.05	mm/yr
B	Good – Minor effect; low corrosion rate or slight discoloration observed.	0.002 ≤ CR < 0.02	in/yr
		0.05 ≤ CR < 0.5	mm/yr
C	Fair – Moderate effect; moderate to high corrosion rate. Softening, weakening or swelling may occur. <i>Not recommended for continuous use.</i>	0.02 ≤ CR < 0.05	in/yr
		0.5 ≤ CR < 1.3	mm/yr
D	Severe Effect – Immediate attack, explosive or very high corrosion rate. <i>Not recommended for any use.</i>	CR ≥ 0.05	in/yr
		CR ≥ 1.3	mm/yr
N	No Data Available	---	---

- Notes:**
- 1 Ratings for all chemicals apply at room temperature unless chemical is molten (e.g., paraffin wax, sulfur, etc.).
  - 2 Ratings for Carbon apply to Graphite-grade Carbon (i.e., Graphite or Carbon-60).
  - 3 Ratings for SiC apply to Self-Sintered Silicon Carbide.
  - 4 Ratings for Ceramic apply to Ceramic Aluminum Oxide (Al<sub>2</sub>O<sub>3</sub>).
  - 5 Ratings apply to chemicals at 100% concentration (except for salts which are based on aqueous solutions) unless stated otherwise.

## Definition of Terms:

SS = Stainless Steel  
 Alloy-20 = High-Nickel Stainless Steel  
 Alloy-C = Ni-Cr-Mo Alloy  
 Ti = Titanium  
 C = Carbon  
 SiC = Silicon Carbide  
 Cer. = Ceramic

Teflon® = Poly-tetra-fluoro-ethylene (PTFE)  
 Ryton® = Poly-phenylene Sulfide (PPS)  
 PEEK = Poly-ether-ether-ketone  
 Viton® = Fluorocarbon Rubber  
 EPDM = Ethylene-Propylene-Diene Monomer (Rubber)  
 NBR = Nitrile Buna Rubber  
 Kalrez® = Perfluorinated Elastomer

Chemical/Fluid	Metals					Plastics			Elastomers				Minerals		
	304 SS	316 SS	Alloy-20	Alloy-C	Ti	Teflon	Ryton	PEEK	Viton	EPDM	NBR	Kalrez	C	SiC	Cer.
<b>A</b>															
Acetaldehyde	A	A	A	A	A	A	A	A	D	B	C	A	A	A	A
Acetic Acid	B	A	A	A	A	A	A	A	B	A	B	A	A	A	A
Acetone	A	A	A	A	A	A	A	A	D	A	D	A	A	A	A
Acetonitrile	A	A	A	A	N	A	A	A	A	A	C	A	A	A	A
Aluminum Chloride, 20%	D	C	C	A	D	A	A	A	A	A	A	A	A	A	A
Alum. Potass. Sulfate, 10%	A	A	A	B	A	A	N	A	A	A	A	A	A	A	A
Aluminum Sulfate	A	A	A	A	A	A	A	A	A	A	A	A	A	A	A
Ammonia, Anhydrous	A	A	A	A	A	A	A	A	A	A	A	B	A	A	A
Ammonium Chloride	D	B	B	B	A	A	A	A	A	A	A	A	A	A	A
Ammonium Hydroxide	A	A	A	A	A	A	A	A	D	A	D	B	A	A	A
Ammonium Nitrate	A	A	A	B	A	A	A	A	A	A	A	A	A	A	A
Ammonium Sulfate	C	B	B	B	A	A	A	A	D	A	A	A	A	A	A
Amyl Acetate	A	A	A	A	A	A	A	A	D	C	D	A	A	A	A
Aniline	A	A	A	B	A	A	A	A	C	B	D	A	A	A	A
Aqua Regia <sup>1</sup>	D	D	D	C	A	A	D	D	A	B	D	A	D	A	A
Asphalt	A	A	A	N	N	A	A	A	A	D	B	A	A	A	A
<b>B</b>															
Barium Hydroxide	B	A	A	B	A	A	A	N	A	A	A	A	A	A	A
Benzene	A	A	A	B	A	A	A <sub>2</sub>	A <sub>2</sub>	A	D	D	A	A	A	A
Benzoic Acid	B	B	B	A	A	A	A	A	A	D	D	A	A	A	A
Benzyl Alcohol	A	A	A	A	A	A	A <sub>2</sub>	N	A	D	D	A	A	A	A
Bitumen	A	A	A	A	A	A	A	A	A	D	B	A	A	A	A
Boric Acid	B	A	B	A	A	A	A	A	A	A	A	A	A	A	A
Brine (NaCl & Water)	B	B	B	A	A	A	A	A	A	C	A	A	A	A	A
Butadiene	A	A	A	A	A	A	A	N	A	D	D	A	A	A	A
Butane	A	A	A	A	A	A	A	A	A	D	A	A	A	A	A
Butyl Acetate	B	B	B	A	A	A	A	A	D	B	D	A	A	A	A
Butyl Alcohol	A	A	A	A	A	A	A	A	A	B	B	A	A	A	A
<b>C</b>															
Calcium Chloride	D	B	B	B	A	A	A	A	A	A	A	A	A	A	A
Calcium Hydroxide	B	B	B	A	A	A	A	A	A	A	A	A	A	A	A
Calcium Hypochlorite	D	B	C	B	A	A	A	A	A	A	B	A	A	A	A
Carbon Disulfide	B	B	B	B	A	A	A	D	A	D	D	A	A	A	A
Carbon Tetrachloride	A	A	A	A	A	A <sub>2</sub>	B	A	A	D	B	A	C	A	A
Carbonic Acid	A	A	A	A	A	A	A	A	A	A	B	A	A	A	A
Chlorine, anhydrous liquid	A	A	A	A	D	A	D	D	A	D	B	A	A	A	A
Chlorobenzene	B	B	B	B	A	A <sub>2</sub>	A <sub>2</sub>	A <sub>2</sub>	A	D	D	A	A	A	A
Chloroform	A	A	A	B	A	A <sub>2</sub>	B	A	A	D	D	A	C	A	A
Chlorosulfonic Acid	B	B	B	A	A	A	D	A	D	D	D	A	A	A	A
Chromic Acid, 30%	D	D	D	B	A	A	B	A	A	B	D	A	A	A	A
Citric Acid	B	A	A	A	A	A	A	A	A	A	A	A	A	A	A
Copper Sulfate	B	B	B	A	A	B	A	A	A	A	A	A	A	A	A
Crude Oil	A	A	A	A	A	A	A	A	A	D	B	A	A	A	A
Cyclohexane	B	B	B	B	A	A	A	A	A	D	A	A	A	A	A
Cyclohexanol	A	A	A	A	A	A	A	A	A	D	A	A	A	A	A
<b>D</b>															
Diesel Fuel	A	A	A	A	A	A	A	A	A	D	A	A	A	A	A
Diethylamine (DEA)	A	A	A	A	A	A	N	A	C	A	C	A	A	A	A
Diethylether	A	A	A	A	A	A	A	A	D	D	D	A	A	A	A
Diethyl Phthalate (DOP)	A	A	A	A	A	A	A	N	B	B	D	A	A	A	A
Dowtherm	A	A	A	A	A	A	A	N	A	D	D	A	A	A	A

## Footnotes:

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- 3) Depends on specific type of Freon.
- 4) For MDI or TDI service, pump must be completely dry. Carbon must not come into contact with humid environments or any compound containing a hydroxyl (-OH) group (e.g., water, alcohols and polyols).

Chemical/Fluid	Metals					Plastics			Elastomers				Minerals		
	304 SS	316 SS	Alloy-20	Alloy-C	Ti	Teflon	Ryton	PEEK	Viton	EPDM	NBR	Kalrez	C	SiC	Cer.
<b>E</b>															
Ethanol	A	A	A	A	A	A	A	A	C	A	C	A	A	A	A
Ether	A	A	A	B	A	A	A	A	C	C	D	A	A	A	A
Ethyl Acetate	A	A	A	A	A	A	A	A	D	B	D	A	A	A	A
Ethylene Glycol	A	A	A	A	A	A	A	A	A	A	A	A	A	A	A
Ethylene Oxide	A	A	A	A	A	A	N	A	D	C	D	A	A	A	A
<b>F</b>															
Fatty Acid	B	A	A	A	B	A	N	A	A	C	B	A	A	A	A
Ferric Chloride	D	D	D	B	A	A	A	B	A	A	A	A	C	A	A
Ferric Sulfate	B	A	A	A	A	A	A	A	A	A	A	A	A	A	A
Fluorosilicic Acid	C	C	C	B	D	A	A	N	B	B	A	A	A	A	C
Formaldehyde	A	A	A	B	A	A	N	A	D	B	C	A	A	A	A
Freon, general	A	A	A	A	A	A <sub>2</sub>	A	A	----- See Footnote 3 -----				A	A	A
Fuel Oil	A	A	A	A	A	B	A	A	A	D	A	A	A	A	A
<b>G</b>															
Gasoline, unleaded	A	A	A	A	A	A	A	A	A	D	A	A	A	A	A
Gasoline, high-aromatic	A	A	A	A	B	A	A <sub>2</sub>	A <sub>2</sub>	A	D	A	A	A	A	A
Glucose (Corn Syrup)	A	A	A	A	A	A	N	A	A	A	A	A	A	A	A
Glycerin (Glycerol)	A	A	A	A	A	A	A	A	A	A	A	A	A	A	A
<b>H</b>															
Heptane	A	A	A	A	A	A	A	A	A	D	A	A	A	A	A
Hexane	A	A	A	A	A	A	A	A	A	D	A	A	A	A	A
Hydraulic Fluid, petro.	A	A	A	A	N	A	N	A	A	D	A	A	B	A	A
Hydraulic Fluid, synth.	A	A	A	A	N	A	N	A	A	D	B	A	B	A	A
Hydrazine	A	A	A	A	A	A	N	A	D	A	B	A	A	A	N
Hydrobromic Acid, 20%	D	D	D	A	A	A	N	D	A	A	D	A	A	A	N
Hydrochloric Acid, 37%	D	D	D	B	D	A	D	A	A	C	D	A	A	A	C
Hydrofluoric Acid, 20%	D	D	D	B	D	A	A	D	D	D	D	A	A	A	D
Hydrogen Peroxide, 50%	B	B	B	D	A	A	D	A	A	C	D	A	C	A	N
<b>I-J-K-L</b>															
Isopropyl Alcohol	A	A	A	A	A	A	A	A	A	A	B	A	A	A	A
Jet Fuel	A	A	A	A	A	A	A	A	A	D	B	A	A	A	A
Kerosene	A	A	A	A	A	A	A	A	A	D	A	A	A	A	A
Lactic Acid	B	B	B	B	A	A	A	A	A	A	A	A	A	A	A
<b>M</b>															
Maleic Acid	A	B	B	B	A	A	N	A	A	D	D	A	A	A	N
Methanol	A	A	A	A	B	A	A	A	D	A	D	A	A	A	A
Methyl Ethyl Ketone (MEK)	A	A	A	A	A	A	A	A	D	A	D	A	A	A	A
Methylene Chloride	B	B	B	B	B	A <sub>2</sub>	A	A	B	D	D	A	C	A	A
Methylene p-diphenyl-Di-Isocyanate (MDI) <sup>4</sup>	N	A	N	A	N	A	N	N	D	B	D	A	A <sub>4</sub>	A	A
Mineral Oil	A	A	A	A	A	A	A	A	A	D	A	A	A	A	A
<b>N</b>															
Naphtha	A	A	A	A	A	A	A	A	A	D	B	A	A	A	A
Naphthalene	A	A	A	B	A	A	A	A	A	D	D	A	A	A	A
Nitric Acid, 20%	A	A	A	A	A	A	D	B	A	B	D	A	A	A	A
Nitrous Acid	D	B	B	D	B	A	N	A	C	A	C	A	A	A	A
<b>O</b>															
Oleic Acid	B	B	B	A	A	A	A	A	B	D	C	A	A	A	A
Oleum	A	A	A	B	D	A	B	N	A	D	D	A	D	A	A
Oxalic Acid	B	A	A	B	A	A	A	A	A	A	B	A	A	A	A

## Footnotes:

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	304 SS	316 SS	Alloy-20	Alloy-C	Ti	Teflon	Ryton	PEEK	Viton	EPDM	NBR	Kalrez	C	SiC	Cer.
<b>P</b>															
Paraffin	A	A	A	B	A	A	N	A	A	D	A	A	A	A	A
Pentane	B	B	N	A	A	A	N	A	A	D	A	A	A	A	A
Perchloric Acid	D	D	D	B	D	A	A	A	A	A	D	A	D	A	A
Perchloroethylene	B	B	B	A	A	A <sub>2</sub>	A	A	A	D	B	A	A	A	A
Phenol	A	A	A	A	A	A	A <sub>2</sub>	D	A	D	D	A	A	A	A
Phosphoric Acid, 40%	B	B	B	A	C	A	C	A	A	B	D	A	A	A	A
Phthalic Acid	B	A	A	B	A	A	N	A	C	A	C	A	A	A	N
Potassium Hydroxide, 50%	B	B	B	B	C	A	A	A	D	A	B	A	A	A	D
Potassium Nitrate	B	B	B	B	A	A	A	A	A	A	A	A	A	A	B
Potassium Permanganate	B	B	B	A	A	A	A	A	C	A	C	A	A	A	A
Propyl Alcohol	A	A	A	A	A	A	A	A	A	A	A	A	A	A	A
Propylene Glycol	B	B	B	B	A	A	N	N	C	A	C	A	A	A	A
Pyridine	A	A	A	B	B	A	A	A	A	B	D	A	A	A	A
<b>S</b>															
Silver Nitrate	B	A	A	A	A	A	A	A	A	A	B	A	A	A	A
Sodium Bicarbonate	A	A	A	B	A	A	A	A	A	A	A	A	A	A	A
Sodium Bisulfite	B	B	B	B	B	A	A	A	A	A	A	A	A	A	A
Sodium Chlorate	B	B	A	A	A	A	A	A	C	A	C	A	A	A	A
Sodium Chloride	B	B	B	A	A	A	A	A	A	C	A	A	A	A	A
Sodium Hydroxide, 50%	B	B	B	A	B	A	A	A	B	A	B	A	A	A	D
Sodium Hypochlorite, ≤20%	D	D	D	A	A	A	C	A	A	A	B	A	B	A	A
Sodium Peroxide	A	A	A	B	N	A	N	A	A	A	B	A	A	A	A
Sodium Silicate	A	A	A	B	A	A	A	A	A	A	A	A	A	A	A
Sodium Sulfide	B	B	B	B	A	A	A	A	A	A	A	A	A	A	A
Sulfur, molten	B	A	A	A	A	A	N	N	A	C	D	A	A	A	A
Sulfuric Acid, <10%	D	B	A	A	D	A	A	A	A	A	A	A	A	A	A
Sulfuric Acid, 10-75%	D	D	A	A	D	A	B	D	A	B	B	A	A	A	A
Sulfuric Acid, 75-93%	D	D	A	A	D	A	B	D	A	B	C	A	A	A	A
Sulfuric Acid, 93-100%	A	A	A	A	D	A	B	D	A	C	N	A	A	A	A
Sulfurous Acid	B	B	B	B	A	A	A	A	A	B	B	A	A	A	A
<b>T</b>															
Tall Oil (Liquid Rosin)	B	B	B	A	N	A	N	N	A	D	A	A	A	A	A
Tallow	A	A	N	N	N	A	N	A	A	D	A	A	A	A	A
Tetrahydrofuran (THF)	A	A	A	A	B	A	A	A	D	B	D	A	A	A	A
Thionyl Chloride	N	D	N	A	N	A <sub>2</sub>	N	N	A	D	B	A	A	A	A
Toluene	A	A	A	A	A	A	A <sub>2</sub>	A <sub>2</sub>	A	D	D	A	A	A	A
Toluene Di-Isocyanate (TDI) 4	N	A	N	A	N	A	N	N	D	B	D	A	A <sub>4</sub>	A	A
Toluenesulphonic Acid	N	N	N	A	N	A	N	N	C	A	C	A	A	A	A
Trichloroethane	A	A	A	A	A	A <sub>2</sub>	N	A	A	D	D	A	A	A	A
Trichloroethylene	B	B	B	A	A	A <sub>2</sub>	C	A	A	D	C	A	A	A	A
Trichlorotrifluoroethane	A	A	A	A	A	A <sub>2</sub>	A	A	B	D	A	C	A	A	A
<b>U-V-W-X</b>															
Urea	B	B	B	B	A	A	A	A	A	A	B	N	A	A	B
Vinyl Chloride	B	A	A	A	A	A <sub>2</sub>	N	A	A	D	B	A	A	A	A
Water, distilled	A	A	A	A	A	A	A	A	A	A	A	A	A	A	A
Water, deionized	A	A	N	A	A	A	A	A	A	A	A	A	A	A	B
Xylene	A	A	A	A	A	A	A <sub>2</sub>	A <sub>2</sub>	A	D	D	A	A	A	A

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## NEWTONIAN &amp; NON-NEWTONIAN MATERIALS

## NEWTONIAN MATERIALS

Newton deduced that the viscosity of a given liquid should be constant at any particular temperature and pressure and independent of the rate of shear, as illustrated in **Figure 1** below. In such “**Newtonian fluids**”, shear stress is directly proportional to rate of shear. At temperatures above their cloud points most mineral oils are Newtonian fluids.

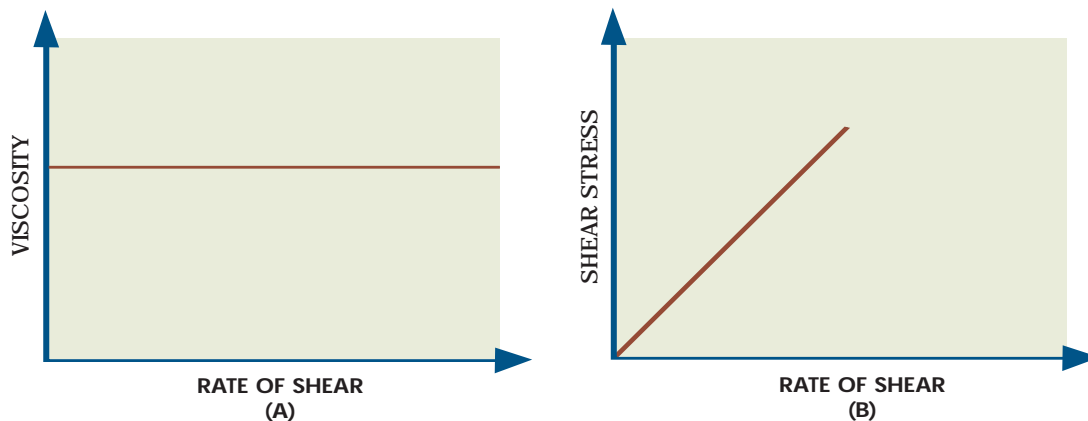


FIGURE 1: **Characteristics of Newtonian liquids. (A) Viscosity is independent of Rate of Shear. (B) Shear Stress is directly proportional to Rate of Shear.**

## NON-NEWTONIAN MATERIALS

The viscosities of some materials, such as greases and polymer-thickened mineral oils, are affected by shearing effects, and these materials are termed *Non-Newtonian*. In other words, the viscosity of a Non-Newtonian fluid will depend on the rate of shear at which it is measured. Since a Non-Newtonian fluid can have an unlimited number of viscosity values (as the shear rate is varied) the term **apparent viscosity** is used to describe its viscous properties. **Apparent viscosity** is a measure of the resistance to flow at a given rate of shear. It has meaning only if the rate of shear is given. It is obtained experimentally by measuring and dividing the shear stress by the rate of shear. A “**rheogram**” or “**flow curve**” relating shear stress to rate of shear is frequently used to completely describe the viscous properties of a non-Newtonian material.

Non-Newtonian materials may be divided into five types: *plastic*, *pseudo-plastic*, *dilatant*, *thixotropic*, and *rheopectic*. **Figure 2** presents characteristic rheograms in which shear stress (e.g., pressure in a steady flow system) is plotted against the rate of shear (which is proportional to the flow velocity). **Figure 3** illustrates how the apparent viscosities of Non-Newtonian materials vary with changing rates of shear.

As illustrated in curve 1 of **Figure 2**, a **plastic** material such as a grease, putty or molding clay, is characterized by a “yield point” or “yield value”. This means that a definite minimum stress or force must be applied to the material before any flow takes place. From a rheological standpoint, tomato ketchup is a common example of a plastic material. If a bottle is shaken only gently, its contents may not flow out because its “yield point” has not been exceeded. However, if the bottle is struck or shaken more vigorously, the yield point is exceeded, the viscosity is reduced and the ketchup gushes forth.

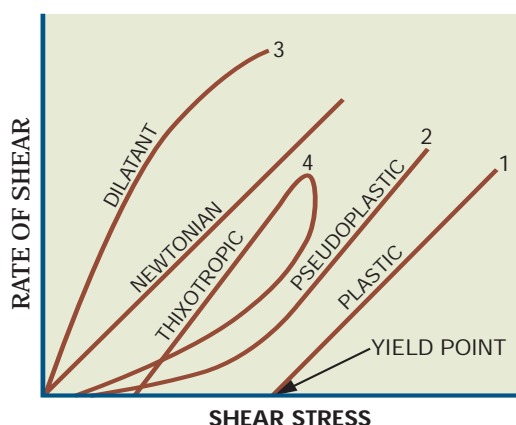


FIGURE 2: **Characteristics of various types of materials.**

While a **pseudo-plastic fluid** has no yield point, its apparent viscosity also decreases with increasing shear rates but stabilizes only at very high rates of shear. Many emulsions such as water-base fluids and resinous materials show this type of behavior.

Oppositely, the apparent viscosity of a **dilatant fluid** increases as the rate of shear increases. Such a fluid often solidifies at high rates of shear. Examples are pigmented-vehicle suspensions such as paints, printing inks and some starches.

The three fluids described above – plastic, pseudo-plastic and dilatant are also known as time-independent non-Newtonian fluids since their rheological or flow properties are independent of time. The rate of shear at any point in the fluid is a simple function of the shear stress at that point.

On the other hand, the flow properties of the other two Non-Newtonian materials – **thixotropic** and **rheopectic** – are dependent on time. The apparent viscosity of these more complex fluids depends not only on the magnitude of the shear rate but also on the length of time during which the shear rate has been applied, as illustrated in **Figure 3**.



## NEWTONIAN &amp; NON-NEWTONIAN MATERIALS (continued)

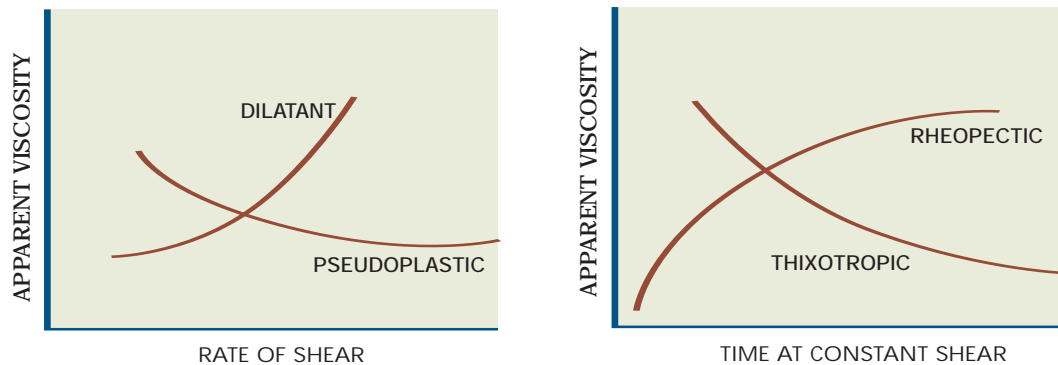


FIGURE 3:  
*Different types  
of Non-  
Newtonian  
behavior.*

If a **thixotropic** fluid is subjected to a constant rate of shear for some time, its structure is gradually broken down and its apparent viscosity decreases to some minimum value. When the shear effect is removed and the fluid is at rest, the structure rebuilds gradually and the apparent viscosity increases with time to the original value. This is called *reversible thixotropy*. If, however, upon removing the shear stress, a value less than the original viscosity is obtained with time, the phenomenon is known as *irreversible thixotropy*. Some oils containing high molecular weight polymers and mineral oils at temperatures below their cloud points show this latter effect.

During rotary drilling of deep oil wells, a very special “drilling mud” with **thixotropic** properties is pumped down the hollow drill stem to force cuttings back to the surface. As long as the mud is agitated by rotation of the drill stem and by pumping, it remains fluid and removes drilling debris. However, whenever drilling is stopped, the drilling mud solidifies to a gel, holds the cuttings in suspension, and thereby prevents them from settling and interfering with subsequent drilling.

Quicksand is also **thixotropic** since it becomes more and more fluid when agitated; therefore, anyone caught in this water-and-sand mixture improves his chance of survival by remaining as motionless as possible.

If a **rheopectic** fluid is subjected to a constant rate of shear for a given period of time, its apparent viscosity increases to some maximum value. Upon cessation of shearing and resting for a time, its apparent viscosity decreases again.

Some greases are intentionally manufactured to have partial rheopectic properties which facilitate pumping from a drum or central grease storage in which the grease is in a relatively fluid condition. Upon shearing in a bearing, however, the grease builds up to a higher apparent viscosity or consistency and stays in place. Such grease does not have full rheopectic characteristics, however, since after shearing and resting, it still retains a higher consistency.

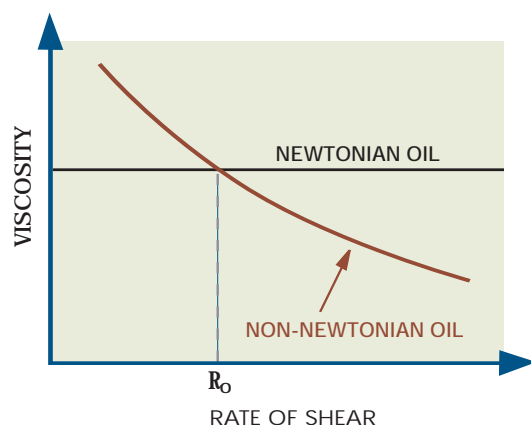


FIGURE 4: *Viscosity vs. Rate of Shear for Newtonian and Non-Newtonian oils.*

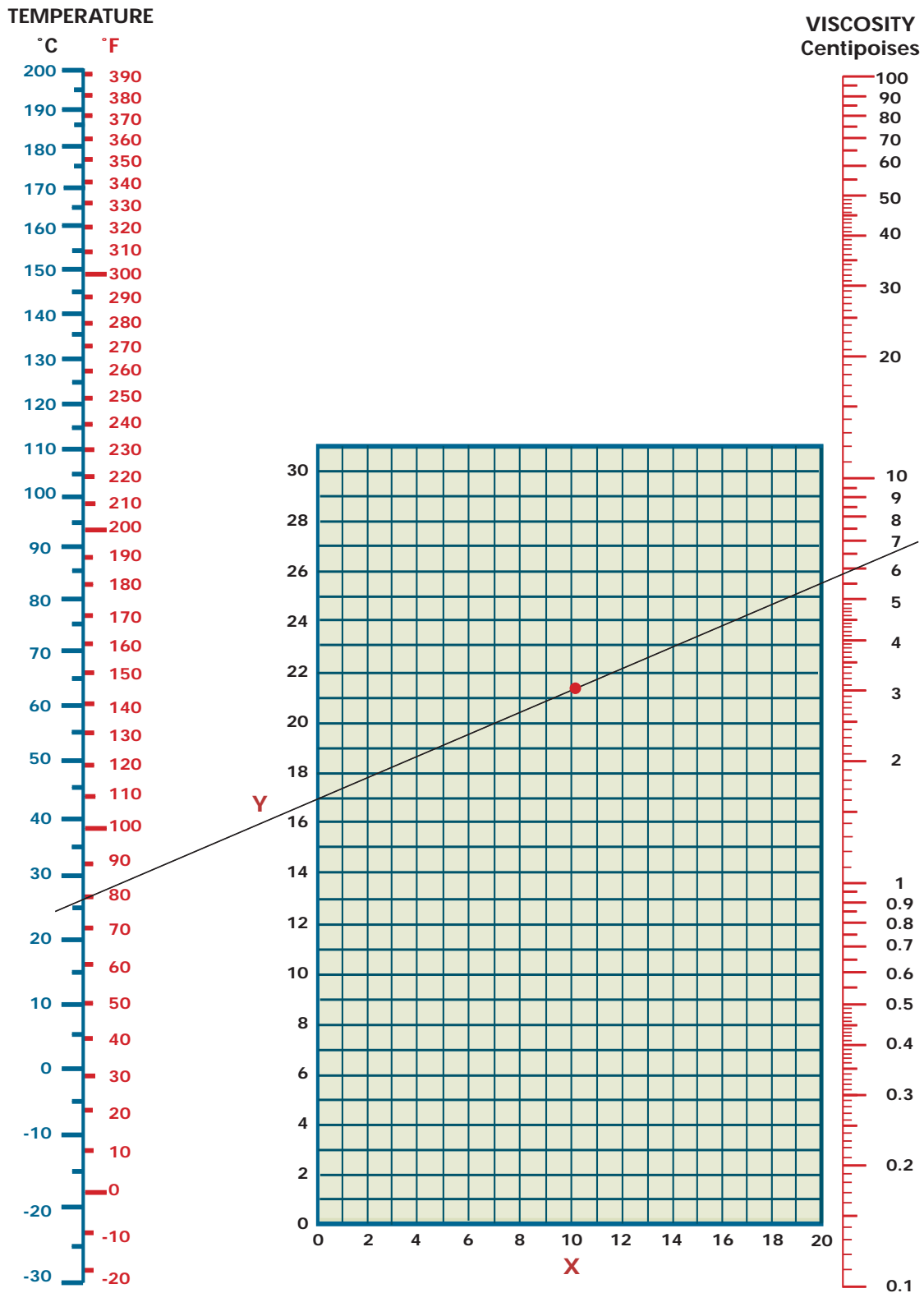
Since the viscosity of a Non-Newtonian lubricant is dependent upon the rate of shear acting on it, the importance of measuring viscosity at various shear rates that will be encountered in the use of such a lubricant can readily be seen. In some machine elements, shear rates up to 3 million reciprocal seconds may be encountered, while in other applications only a few reciprocal seconds or a few tenths are the order of magnitude. In dispensing greases, shear rates as low as 0.1 reciprocal seconds are sometimes encountered, while leakage from housings during periods of shutdown involves an even lower range.

As illustrated in **Figure 4**, the determination of viscosity of a Non-Newtonian liquid only at one shear rate is not usually sufficient. Incorrect conclusions would be drawn and application difficulties would be invited if the viscosities of a Newtonian and Non-Newtonian oil were measured at some specific shear rate  $R_0$ , where the two curves happen to cross each other. While both oils have the same apparent viscosity at this one point, the remainders of their viscosity-shear curves are entirely different.

## VISCOSITY vs. TEMPERATURE for SPECIFIC CHEMICALS

Coordinates for Chart on following page:

Liquid	X	Y	Liquid	X	Y
Acetaldehyde.....	15.2	4.8	Freon-113.....	12.5	11.4
Acetic acid, 100%.....	12.1	14.2	Glycerol, 100% .....	2.0	30.0
Acetic acid, 70%.....	9.5	17.0	Glycerol, 50% .....	6.9	19.6
Acetic anhydride .....	12.7	12.8	Heptane .....	14.1	8.4
Acetone, 100%.....	14.5	7.2	Hexane .....	14.7	7.0
Acetone, 35%.....	7.9	15.0	Hydrochloric acid, 31.5% .....	13.0	16.6
Acetonitrile.....	14.4	7.4	Iodobenzene .....	12.8	15.9
Acrylic acid.....	12.3	13.9	Isobutyl alcohol .....	7.1	18.0
Allyl alcohol.....	10.2	14.3	Isobutyric acid .....	12.2	14.4
Allyl bromide .....	14.4	9.6	Isopropyl alcohol .....	8.2	16.0
Allyl iodide.....	14.0	11.7	Isopropyl bromide .....	14.1	9.2
Ammonia, 100%.....	12.6	2.0	Isopropyl chloride .....	13.9	7.1
Ammonia, 26%.....	10.1	13.9	Isopropyl iodide .....	13.7	11.2
Amyl acetate.....	11.8	12.5	Kerosene .....	10.2	16.9
Amyl alcohol.....	7.5	18.4	Linseed oil, raw .....	7.5	27.2
Aniline.....	8.1	18.7	Mercury .....	18.4	16.4
Anisole.....	12.3	13.5	Methanol, 100% .....	12.4	10.5
Arsenic trichloride.....	13.9	14.5	Methanol, 90% .....	12.3	11.8
Benzene.....	12.5	10.9	Methanol, 40% .....	7.8	15.5
Brine, CaCl <sub>2</sub> , 25%.....	6.6	15.9	Methyl acetate .....	14.2	8.2
Brine, NaCl, 25%.....	10.2	16.6	Methyl acrylate .....	13.0	9.5
Bromine.....	14.2	13.2	Methyl i-butyrate .....	12.3	9.7
Bromotoluene.....	20.0	15.9	Methyl n-butyrate .....	13.2	10.3
Butyl acetate.....	12.3	11.0	Methyl chloride .....	15.0	3.8
Butyl acrylate.....	11.5	12.6	Methyl ethyl ketone .....	13.9	8.6
Butyl alcohol.....	8.6	17.2	Methyl formate .....	14.2	7.5
Butyric acid.....	12.1	15.3	Methyl iodide .....	14.3	9.3
Carbon dioxide.....	11.6	0.3	Methyl propionate .....	13.5	9.0
Carbon disulfide.....	16.1	7.5	Methyl propyl ketone .....	14.3	9.5
Carbon tetrachloride.....	12.7	13.1	Methyl sulfide .....	15.3	6.4
Chlorobenzene.....	12.3	12.4	Napthalene .....	7.9	18.1
Chloroform.....	14.4	10.2	Nitric acid, 95% .....	12.8	13.8
Chlorosulfonic acid.....	11.2	18.1	Nitric acid, 60% .....	10.8	17.0
Chlorotoluene, ortho.....	13.0	13.3	Nitrobenzene .....	10.6	16.2
Chlorotoluene, meta.....	13.3	12.5	Nitrogen dioxide .....	12.9	8.6
Chlorotoluene, para.....	13.3	12.5	Nitrotoluene.....	11.0	17.0
Cresol, meta.....	2.5	20.8	Octane .....	13.7	10.0
Cyclohexanol.....	2.9	24.3	Octyl alcohol .....	6.6	21.1
Cyclohexane.....	9.8	12.9	Pentachloroethane .....	10.9	17.3
Dibromomethane.....	12.7	15.8	Pentane .....	14.9	5.2
Dichloroethane.....	13.2	12.2	Phenol .....	6.9	20.8
Dichloromethane.....	14.6	8.9	Phosphorus tribromide .....	13.8	16.7
Diethyl ketone.....	13.5	9.2	Phosphorus trichloride .....	16.2	10.9
Diethyl oxalate.....	11.0	16.4	Propionic acid .....	12.8	13.8
Diethylene glycol.....	5.0	24.7	Propyl acetate .....	13.1	10.3
Diphenyl.....	12.0	18.3	Propyl alcohol .....	9.1	16.5
Dipropyl ether.....	13.2	8.6	Propyl bromide .....	14.5	9.6
Dipropyl oxalate.....	10.3	17.7	Propyl chloride .....	14.4	7.5
Ethyl acetate.....	13.7	9.1	Propyl formate .....	13.1	9.7
Ethyl acrylate.....	12.7	10.4	Propyl iodide .....	14.1	11.6
Ethyl alcohol, 100%.....	10.5	13.8	Sodium .....	16.4	13.9
Ethyl alcohol, 95%.....	9.8	14.3	Sodium hydroxide, 50% .....	3.2	25.8
Ethyl alcohol, 40%.....	6.5	16.6	Stannic chloride .....	13.5	12.8
Ethyl benzene .....	13.2	11.5	Succinonitrile .....	10.1	20.8
Ethyl bromide .....	14.5	8.1	Sulfur dioxide .....	15.2	7.1
2-Ethyl butyl acrylate .....	11.2	14.0	Sulfuric acid, 110% .....	7.2	27.4
Ethyl chloride .....	14.8	6.0	Sulfuric acid, 100% .....	8.0	25.1
Ethyl ether .....	14.5	5.3	Sulfuric acid, 98% .....	7.0	24.8
Ethyl formate .....	14.2	8.4	Sulfuric acid, 60% .....	10.2	21.3
2-Ethyl hexyl acrylate .....	9.0	15.0	Sulfuryl chloride .....	15.2	12.4
Ethyl iodide .....	14.7	10.3	Tetrachloroethane .....	11.9	15.7
Ethyl propionate .....	13.2	9.9	Thiophene .....	13.2	11.0
Ethyl propyl ether .....	14.0	7.0	Titanium tetrachloride .....	14.4	12.3
Ethyl sulfide .....	13.8	8.9	Toluene .....	13.7	10.4
Ethylene bromide .....	11.9	15.7	Trichloroethylene .....	14.8	10.5
Ethylene chloride .....	12.7	12.2	Triethylene glycol .....	4.7	24.8
Ethylene glycol.....	6.0	23.6	Turpentine .....	11.5	14.9
Ethylidene chloride .....	14.1	8.7	Vinyl acetate .....	14.0	8.8
Fluorobenzene .....	13.7	10.4	Vinyl toluene .....	13.4	12.0
Formic acid .....	10.7	15.8	Water .....	10.2	13.0
Freon-11 .....	14.4	9.0	Xylene, ortho .....	13.5	12.1
Freon-12 .....	16.8	15.6	Xylene, meta .....	13.9	10.6
Freon-21 .....	15.7	7.5	Xylene, para .....	13.9	10.9
Freon-22 .....	17.2	4.7			

VISCOSITY vs. TEMPERATURE for SPECIFIC CHEMICALS *(continued)*

Example: How to find viscosity of a specific chemical at a specified temperature.

Find the viscosity of 60% Sulfuric Acid at 80°F

- 1) Locate Sulfuric acid, 60% in chemical list on previous page.
- 2) Plot X-Y coordinates (X=10.2, Y=21.3) in the above grid.
- 3) Draw a line from 80°F through the plotted point (see above).
- 4) Viscosity for 60% Sulfuric Acid at 80°F is 5.8 centipoises.

## FRICION LOSS MODULUS - Introduction

### DETERMINATION OF FRICTION LOSS MODULUS

The graphs on the following pages are used for calculating pressure losses in piping systems. If the Flow Rate and Viscosity are known, the graph gives the **Friction Loss Modulus** used for calculating the **Friction Loss in Lengths of Pipe**.

The area above the dotted line on the graph indicates the region of Laminar Flow; below the dotted line is the region of Turbulent Flow.

The line for Water (1.13 CS) shows that for flows less than 1.5 GPM using 1/2" pipe that the Friction Loss Modulus is negligible.

The line for Fluid Viscosities of (.46 CS) lies between that of water and the lower limit line indicating turbulent flow.

The graph shows some normally expected trends:

- (1) Friction Loss Modulus increases as the flow rate increases.
- (2) Friction Loss Modulus increases as the viscosity of the fluid increases.

SAMPLE  
FRICTION LOSS  
MODULUS GRAPH  
FOR 1/2" PIPE

#### Example 1:

1/2" = Pipe Size  
3 GPM = Flow Rate  
100 SSU = Viscosity  
Specific Gravity = 1  
Length of Pipe = 150 ft.

From the graph the Flow Loss Modulus = 11

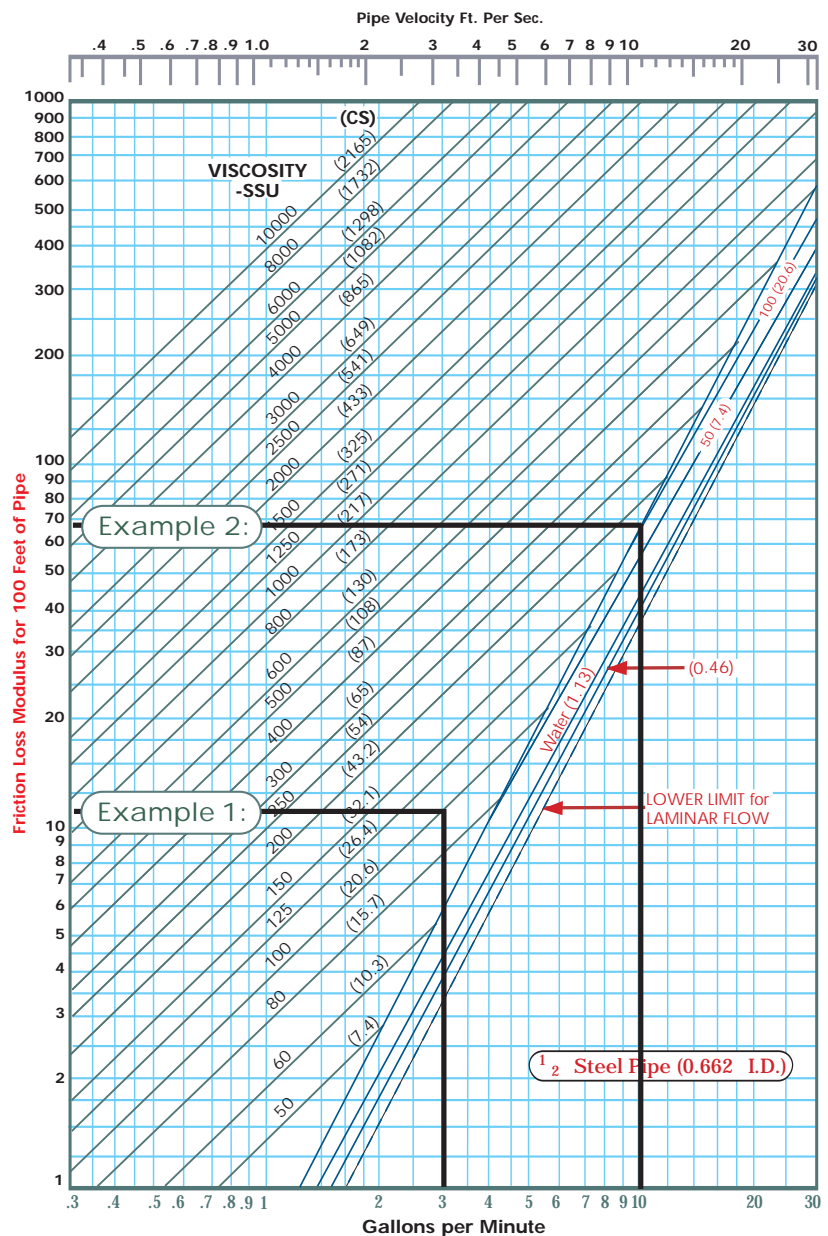
$$\text{Pipe Friction Loss} = \frac{11 \text{ PSI}}{100 \text{ ft}} \times 150 \text{ ft} = 17 \text{ PSI}$$

#### Example 2:

1/2" = Pipe Size  
10 GPM = Flow Rate  
100 SSU = Viscosity  
Specific Gravity = 1.5  
Length of Pipe = 150 ft.

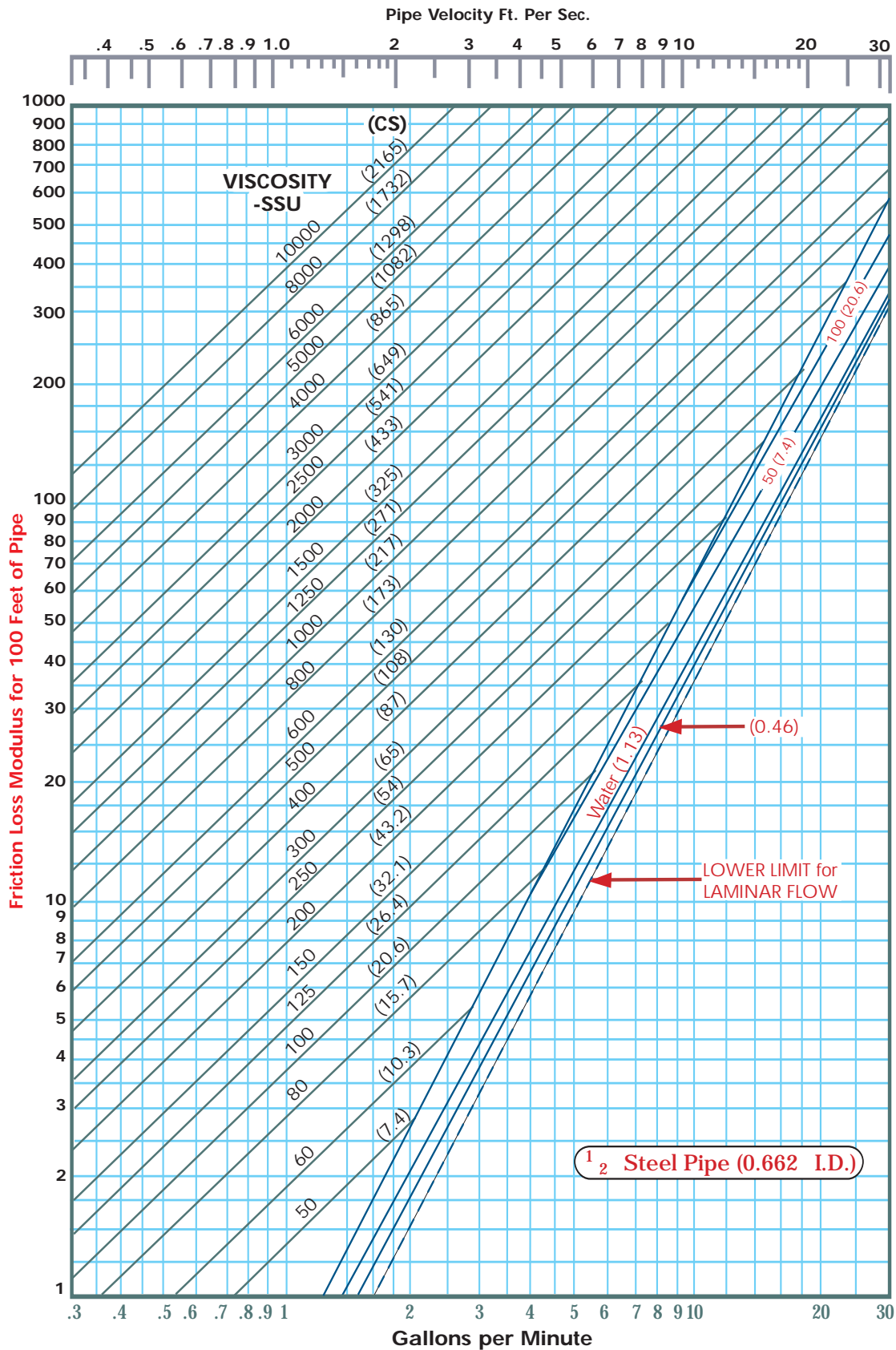
From the graph the Flow Loss Modulus = 68

$$\text{Pipe Friction Loss} = \frac{68 \text{ PSI}}{100 \text{ ft}} \times 150 \text{ ft} \times 1.5 = 153 \text{ PSI}$$



Pipe Losses (PSI) = Modulus x Specific Gravity  
Pipe Losses (Feet of Head) = Modulus x 2.31

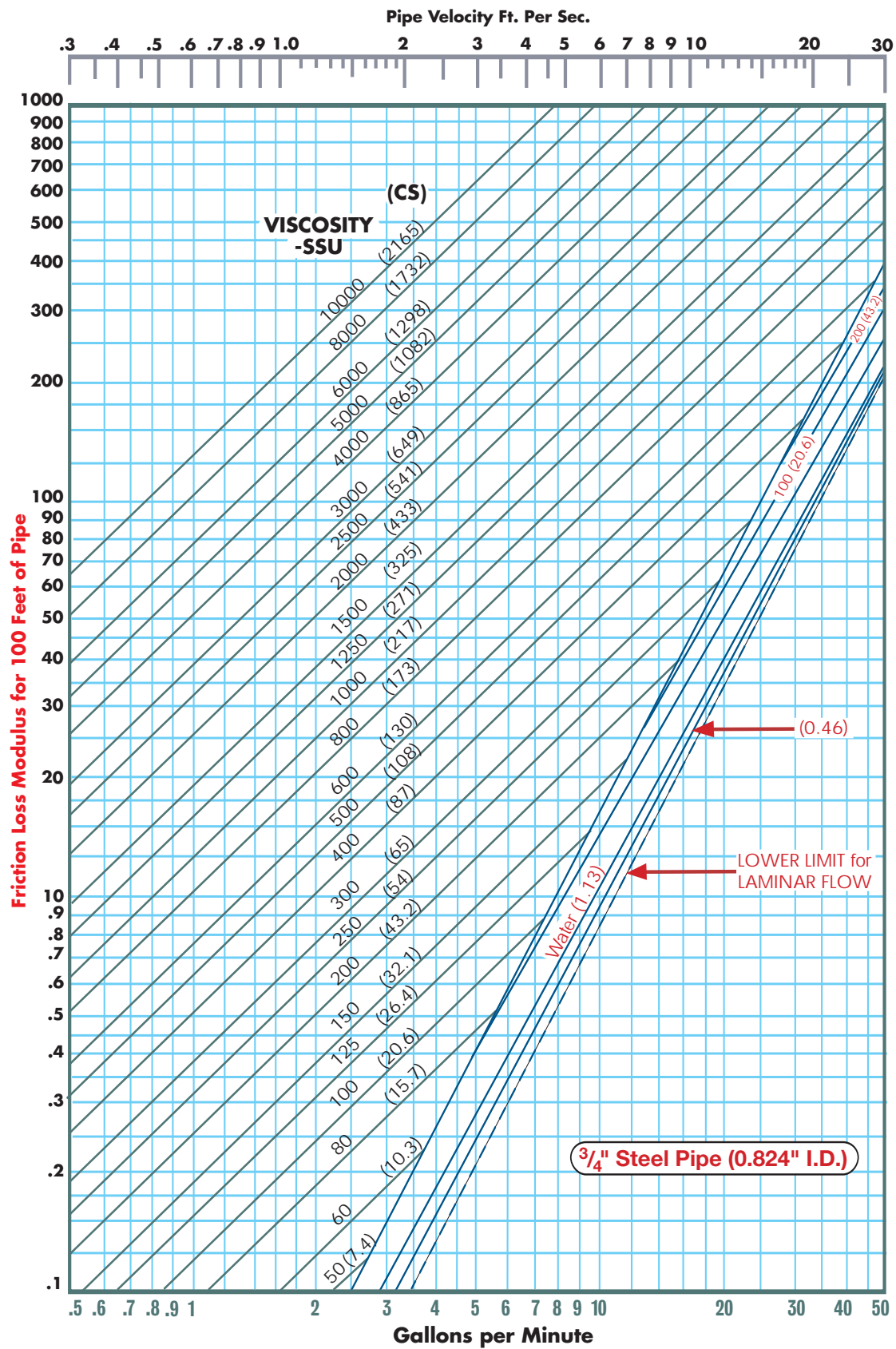
Graph 1: FRICTION LOSS MODULUS - 1/2" Steel Pipe



Pipe Losses (PSI) = Modulus x Specific Gravity  
 Pipe Losses (Feet of Head) = Modulus x 2.31

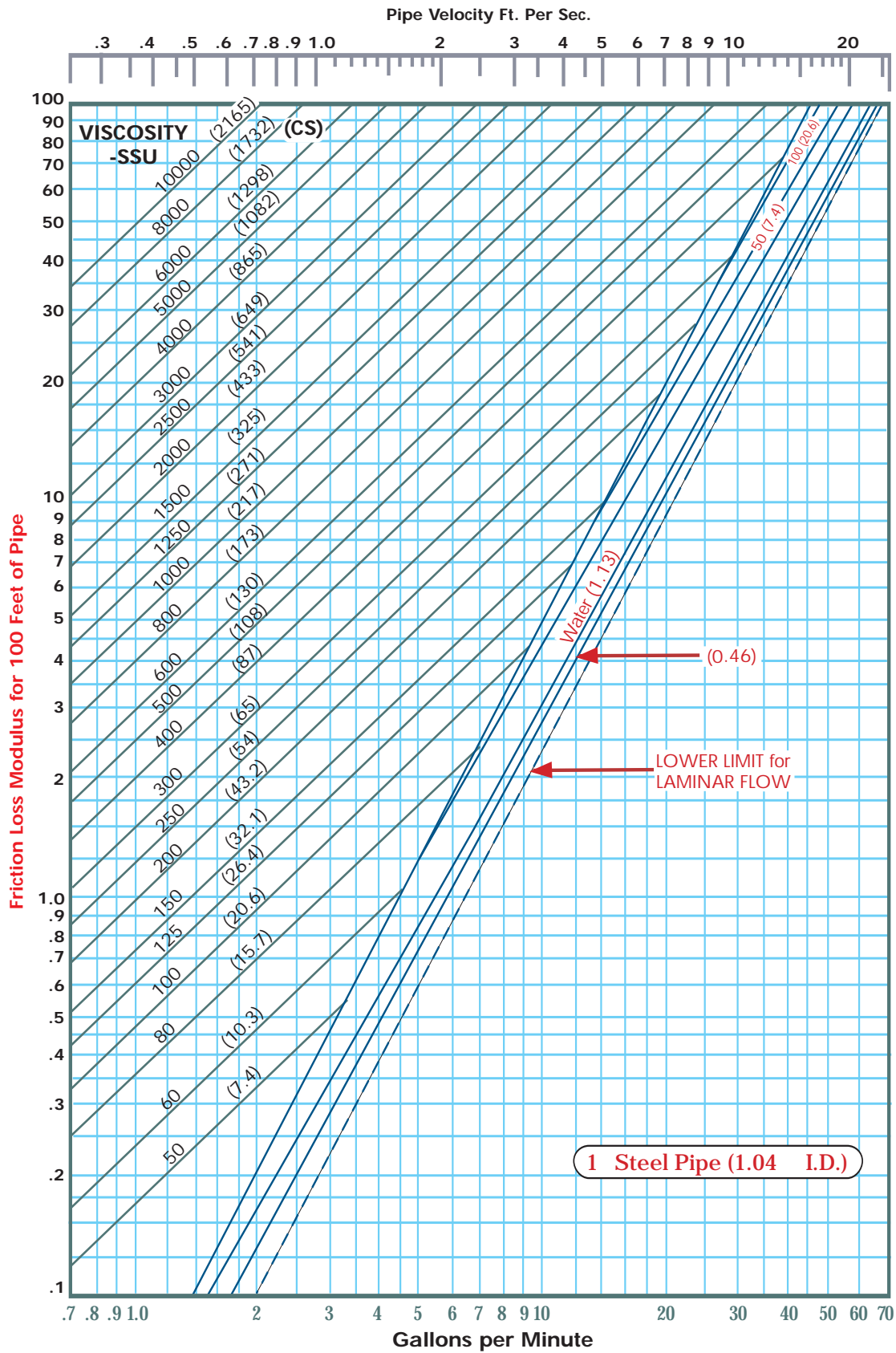


Graph 2: FRICTION LOSS MODULUS - 3/4" Steel Pipe



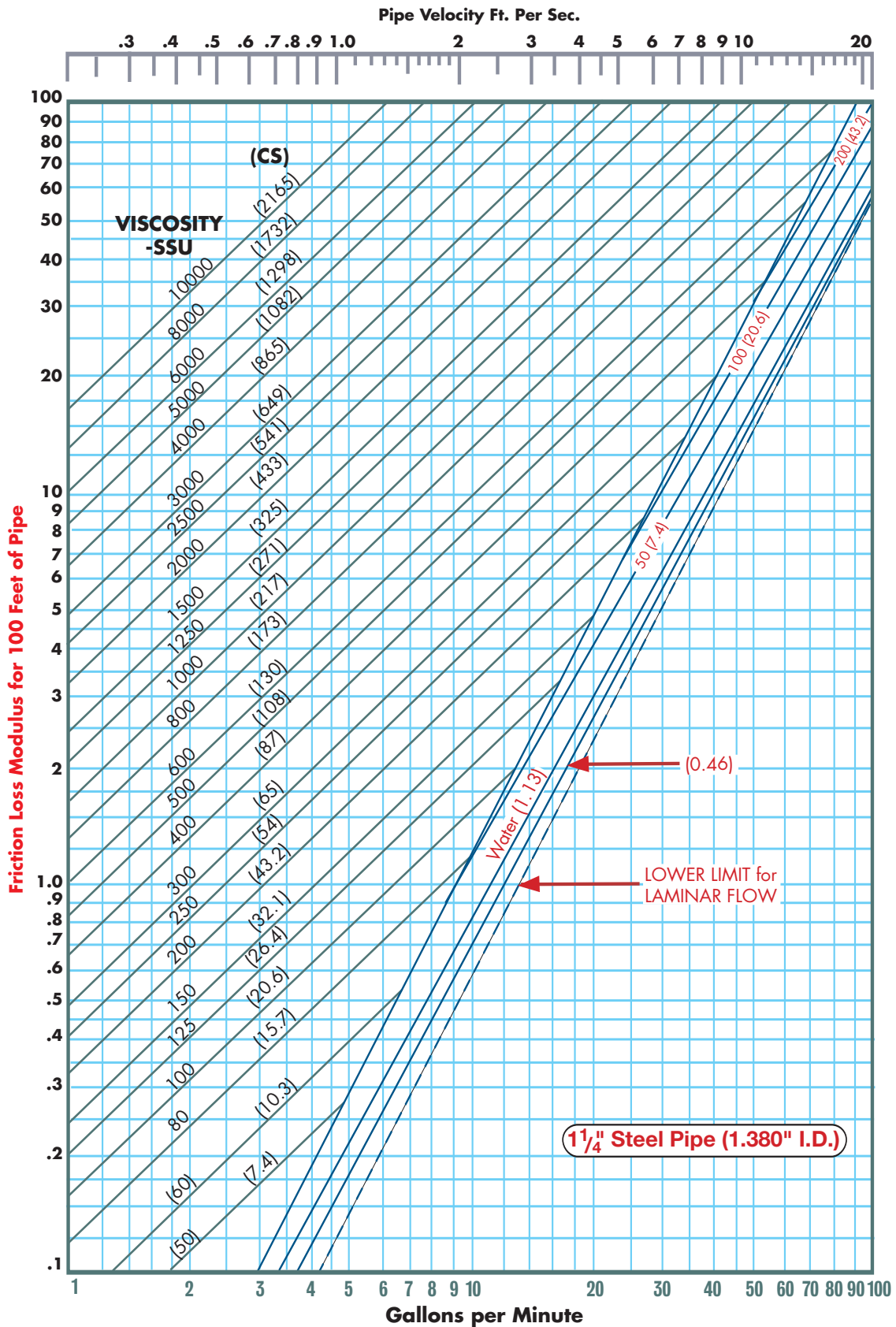
Pipe Losses (PSI) = Modulus x Specific Gravity  
 Pipe Losses (Feet of Head) = Modulus x 2.31

Graph 3: FRICTION LOSS MODULUS - 1" Steel Pipe



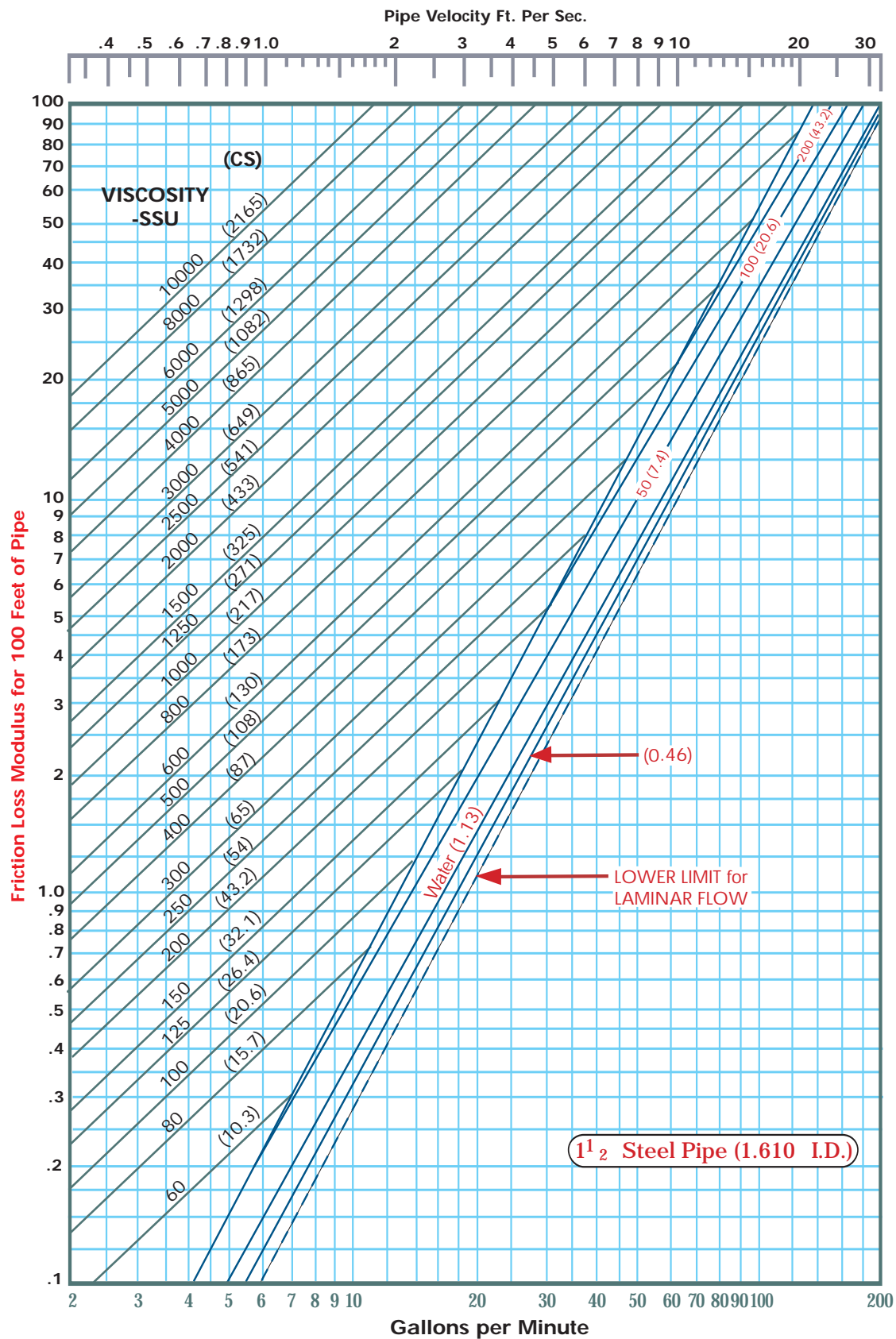
Pipe Losses (PSI) = Modulus x Specific Gravity  
 Pipe Losses (Feet of Head) = Modulus x 2.31

Graph 4: FRICTION LOSS MODULUS - 1 1/4" Steel Pipe



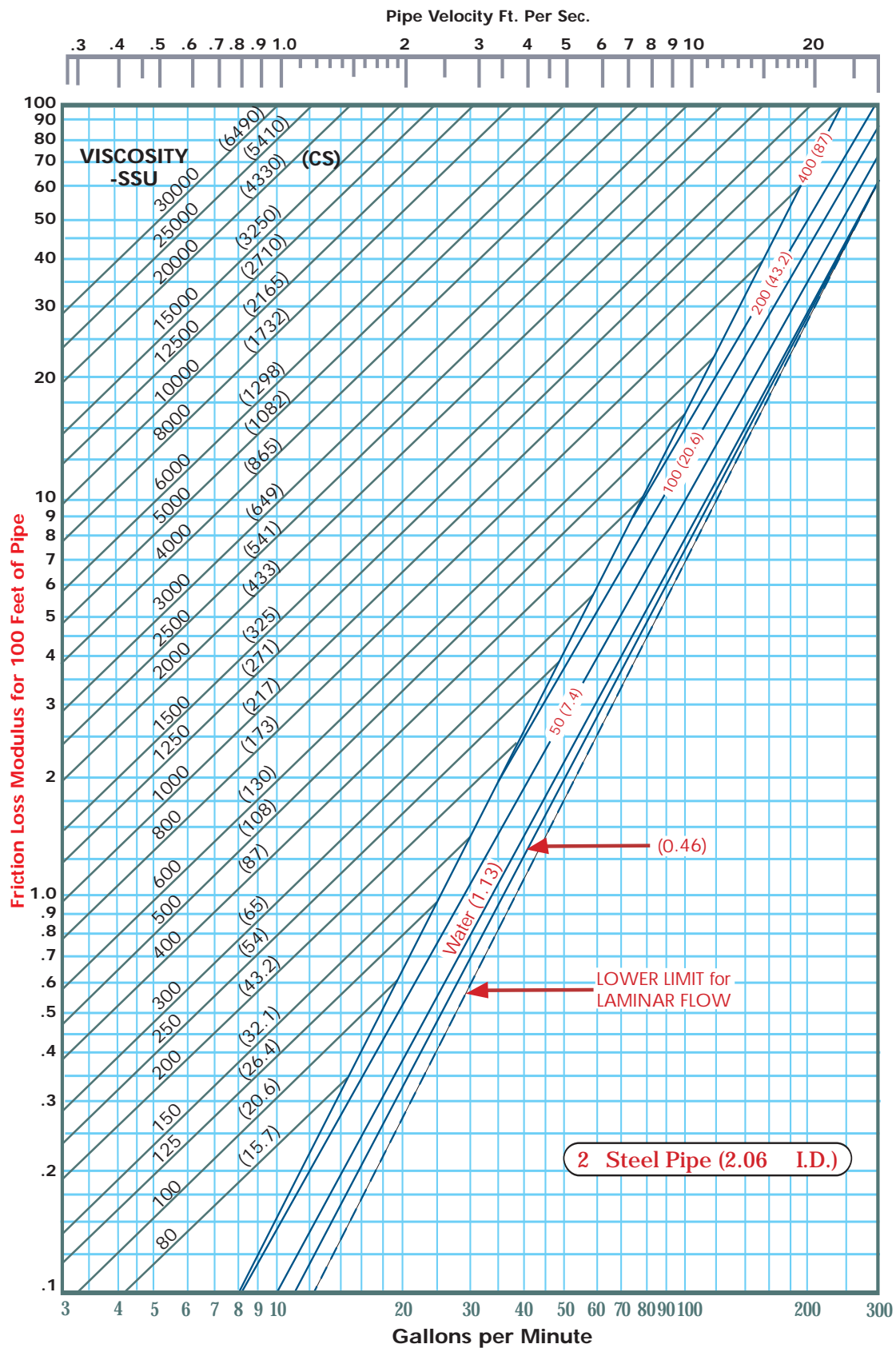
**Pipe Losses (PSI) = Modulus x Specific Gravity**  
**Pipe Losses (Feet of Head) = Modulus x 2.31**

Graph 5: FRICTION LOSS MODULUS - 1½" Steel Pipe



**Pipe Losses (PSI) = Modulus x Specific Gravity**  
**Pipe Losses (Feet of Head) = Modulus x 2.31**

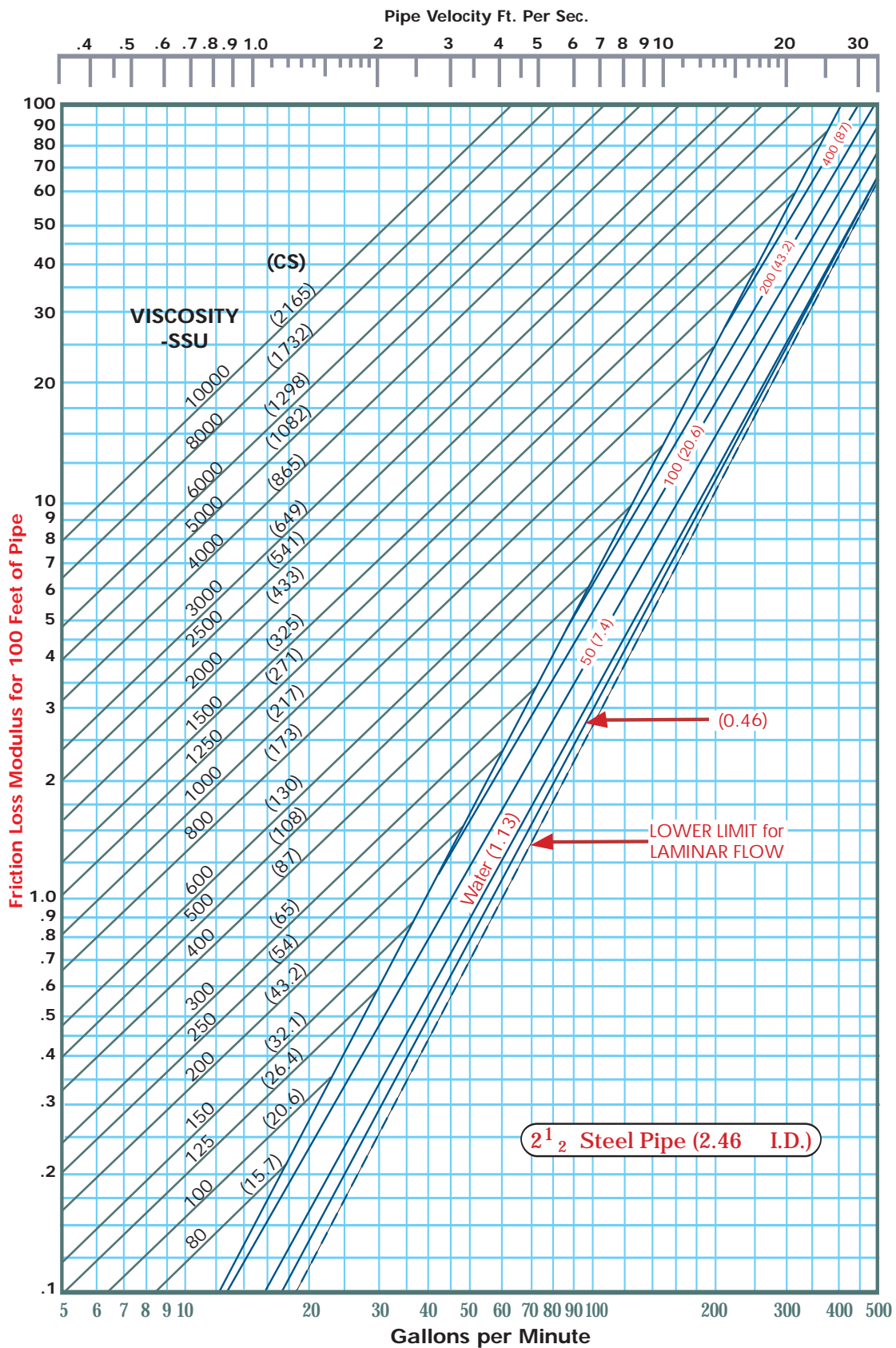
Graph 6: FRICTION LOSS MODULUS - 2" Steel Pipe



Pipe Losses (PSI) = Modulus x Specific Gravity  
 Pipe Losses (Feet of Head) = Modulus x 2.31



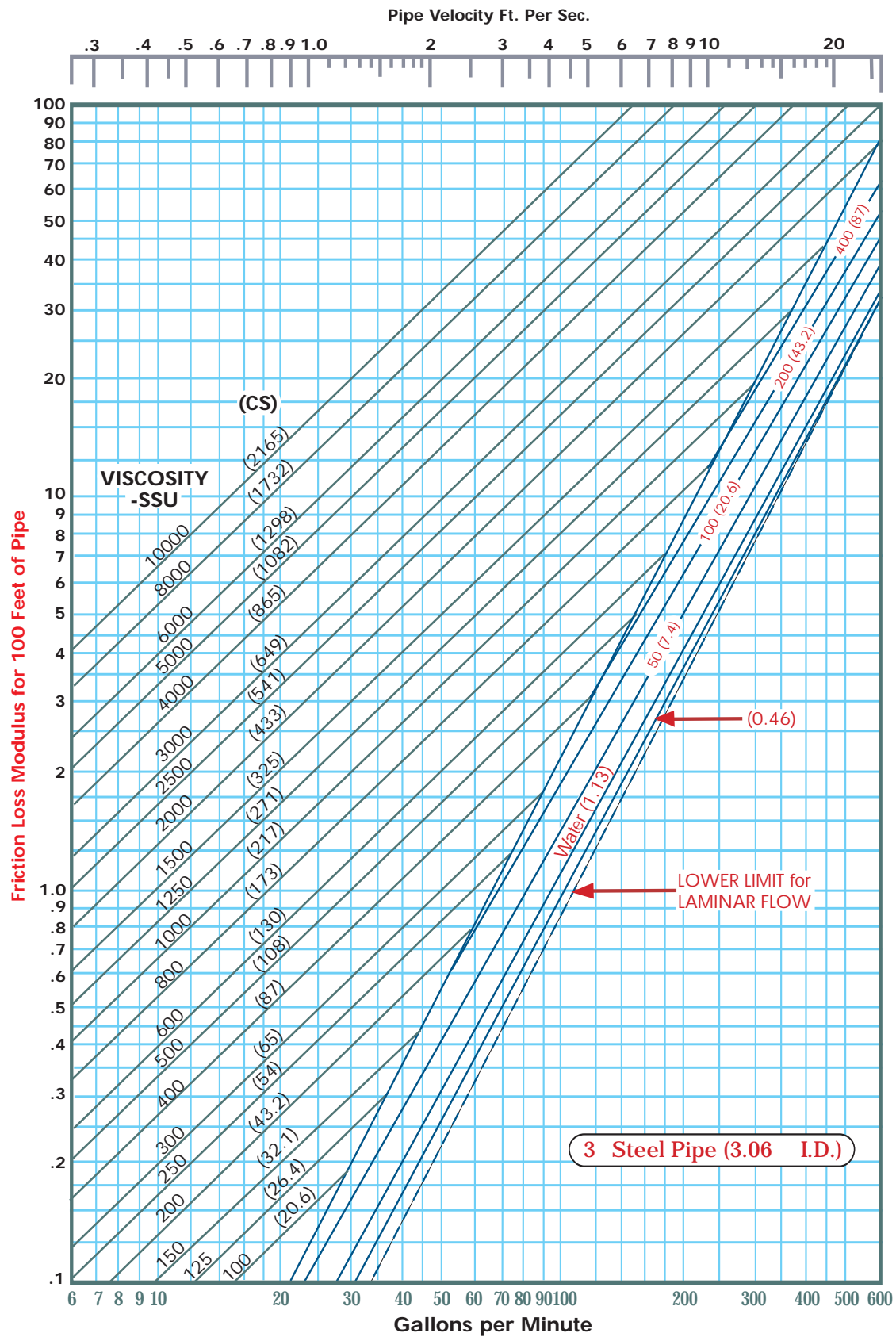
Graph 7: FRICTION LOSS MODULUS - 2 1/2" Steel Pipe



Pipe Losses (PSI) = Modulus x Specific Gravity  
 Pipe Losses (Feet of Head) = Modulus x 2.31

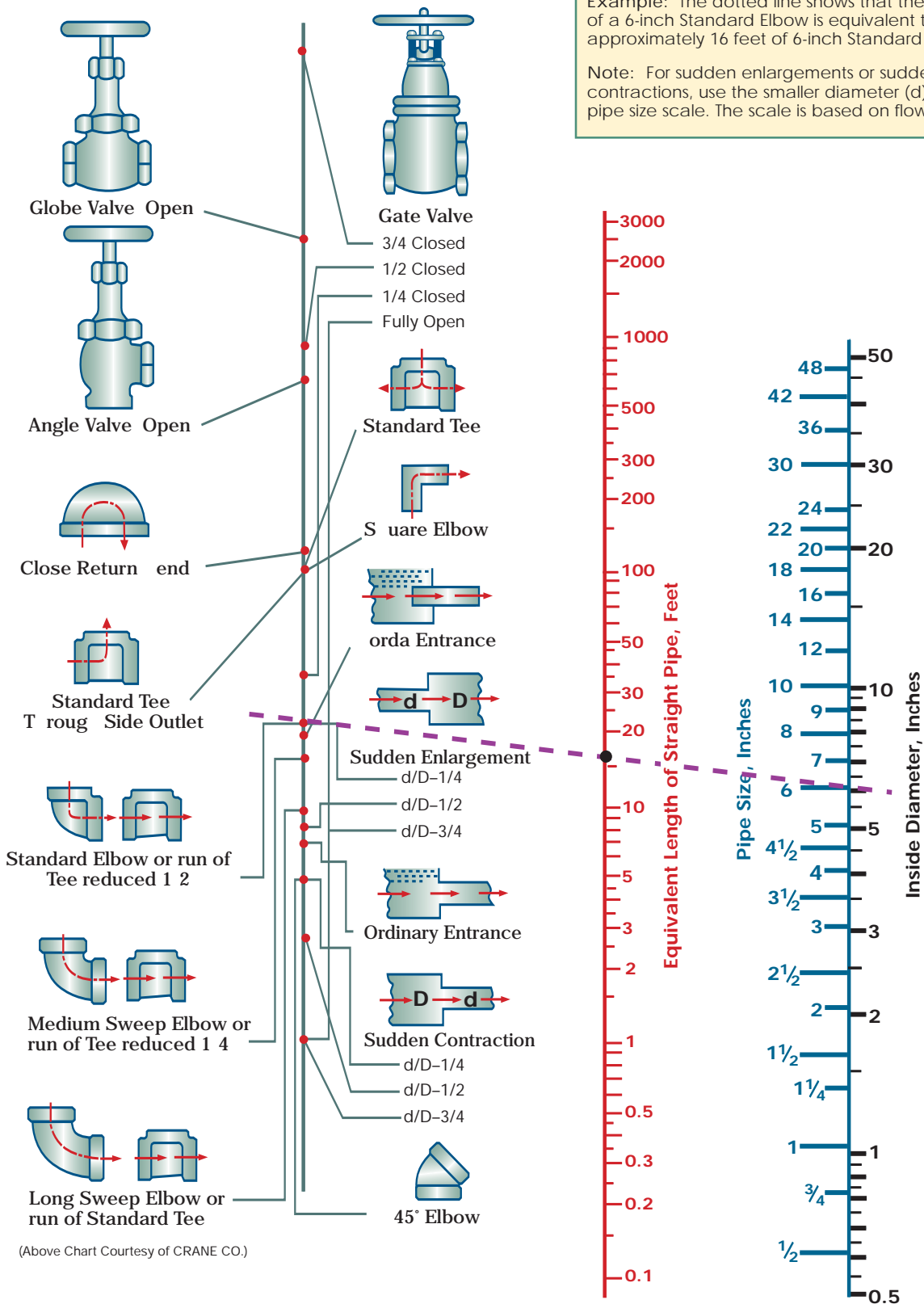


Graph 8: FRICTION LOSS MODULUS - 3" Steel Pipe



Pipe Losses (PSI) = Modulus x Specific Gravity  
 Pipe Losses (Feet of Head) = Modulus x 2.31

## RESISTANCE OF VALVES & FITTINGS TO FLOW OF FLUIDS



(Above Chart Courtesy of CRANE CO.)

## FASTENER TORQUE SPECIFICATIONS

FASTENER MAXIMUM TORQUE VALUES (inch-pounds)			
SIZE	18-8 SS	316 SS	Low Carbon Steel
5 - 40	7.7	8.1	6.9
5 - 44	9.4	9.8	8.5
6 - 32	9.6	10.1	8.7
6 - 40	12.1	12.7	10.9
8 - 32	19.8	20.7	17.8
8 - 36	22.0	23.0	19.8
10 - 24	22.8	23.8	20.8
10 - 32	31.7	33.1	29.7
1/4 - 20	75.2	78.8	65.0
1/4 - 28	94.0	99.0	90.0
5/16 - 18	132.0	138.0	129.0
5/16 - 24	142.0	147.0	139.0
3/8 - 16	236.0	247.0	212.0
3/8 - 24	259.0	271.0	232.0
7/16 - 14	376.0	393.0	338.0
7/16 - 20	400.0	418.0	361.0
1/2 - 13	517.0	542.0	465.0
1/2 - 20	541.0	565.0	487.0
9/16 - 12	682.0	713.0	613.0
9/16 - 18	752.0	787.0	668.0
5/8 - 11	1110.0	1160.0	1000.0
5/8 - 18	1244.0	1301.0	1140.0
3/4 - 10	1530.0	1582.0	1259.0
3/4 - 16	1490.0	1558.0	1230.0
7/8 - 9	2328.0	2430.0	1919.0
7/8 - 14	2318.0	2420.0	1911.0
1 - 8	3440.0	3595.0	2832.0
1 - 14	3110.0	3250.0	2562.0

## DIMENSIONAL DATA FOR WELDED &amp; SEAMLESS PIPE

PIPE DIMENSIONAL DATA									
SIZE	Outside Diameter (in)	Hand-Tight Engagement (in)	Wrench Make-Up (in)	Total Thread Engagement (in)	Threads per inch	Schedule	Wall Thickness (in)	Inside Diameter (in)	Weight per ft. (lbs)
1/8	.405	0.1615	0.1111	0.2726	27	40	.068	.269	.244
						80	.095	.215	.314
1/4	.540	0.2278	0.1667	0.3945	18	40	.088	.364	.424
						80	.119	.302	.535
3/8	.675	0.2400	0.1667	0.4067	18	40	.091	.493	.567
						80	.126	.423	.738
1/2	.840	0.3200	0.2143	0.5343	14	40	.109	.622	.850
						80	.147	.546	1.087
						160	.188	.464	1.311
3/4	1.050	0.3390	0.2143	0.5533	14	40	.113	.824	1.130
						80	.154	.742	1.473
						160	.219	.612	1.944
1	1.315	0.4000	0.2609	0.6609	11 <sup>1</sup> / <sub>2</sub>	40	.133	1.049	1.678
						80	.179	.957	2.171
						160	.250	.815	2.840
1 <sup>1</sup> / <sub>4</sub>	1.660	0.4200	0.2609	0.6809	11 <sup>1</sup> / <sub>2</sub>	40	.140	1.380	2.272
						80	.191	1.278	2.996
						160	.250	1.160	3.764
1 <sup>1</sup> / <sub>2</sub>	1.900	0.4200	0.2609	0.6809	11 <sup>1</sup> / <sub>2</sub>	40	.145	1.610	2.717
						80	.200	1.500	3.631
						160	.281	1.338	4.858
2	2.375	0.4360	0.2609	0.6969	11 <sup>1</sup> / <sub>2</sub>	40	.154	2.067	3.650
						80	.218	1.939	5.020
						160	.344	1.687	7.460
2 <sup>1</sup> / <sub>2</sub>	2.875	0.6820	0.2500	0.9320	8	40	.203	2.469	5.790
						80	.276	2.323	7.660
						160	.375	2.125	10.010
3	3.500	0.7660	0.2500	1.0160	8	40	.216	3.068	7.580
						80	.300	2.900	10.250
						160	.438	2.624	14.310
4	4.500	0.8440	0.2500	1.0940	8	40	.237	4.026	10.790
						80	.337	3.826	14.980
						120	.438	3.624	18.980
						160	.531	3.438	22.520

## ELECTRICAL STANDARDS &amp; MOTOR ENCLOSURE TYPES

**IEEE**

The Institute of Electrical and Electronics Engineers is a technical organization that publishes various recommended practices, which have been adopted as industry standards. Recommended practices do not require approval action or special testing.

**NEC**

National Electric Code is an ASA standard sponsored by the national Fire Protection Association for the purpose of safe-guarding persons and buildings from electrical hazards arising from the use of electricity for light, heat, power, and other purposes.

**NEMA**

National Electrical Manufacturers Association. Standards are voluntary that represent general practices in the industry. They define a product, process, or procedure with reference to nomenclature, composition, construction, dimensions, tolerances, operating characteristics, performance, quality, rating and testing.

**Temperature Codes ("T" Codes)**

**NEC** requires explosion-proof motors to indicate the maximum operating surface temperature. A **"T" Code** is used as the temperature identification value and it is taken from the National Electrical Code. The **"T" Code** is based on the minimum ignition temperature of the substance to be found in the hazardous location. The Code, when marked on the motor nameplate, represents the highest temperature that could be obtained on the motor surface under any normal or abnormal operating conditions, including locked rotor, single phasing or over-loading, and is chosen to be below the ignition temperature of the substance in the hazardous area.

**Enclosure Types**

The selection of the proper enclosure is vital to the successful and safe operation of industrial motors. The user must recognize the specific environmental conditions and specify the correct enclosure.

**Open Drip-proof Motor (ODP)**

An **ODP** motor is an open machine having all openings limited in size to prevent accidental contact with moving or hazardous parts. Internal parts are protected from drip, splash, and falling objects. ODP motors are suitable for general industrial use in indoor locations with relatively clean atmospheres.

**Totally Enclosed (TE) Motor**

A totally enclosed motor prevents the free exchange of air between the inside and outside of the motor case. They are not considered air-tight. Totally enclosed motors are suitable for use in dusty, dirty locations and humid environments.

**Totally Enclosed Non-Ventilated (TENV)** – These motors are not equipped with a ventilation fan for totally cooling the outside skin. Generally limited to low horsepower ratings or short-time rated motors.

**Totally Enclosed Fan-Cooled (TEFC)** – Exterior surface of the motor is cooled by an external fan on the motor shaft. Motor cooling is therefore dependent on motor speed.

**Totally Enclosed Blower-Cooled (TEBC)** – Cooling is achieved with an external blower that is mounted on the back of the motor. The blower provides continuous cooling that is independent of the motor shaft speed.

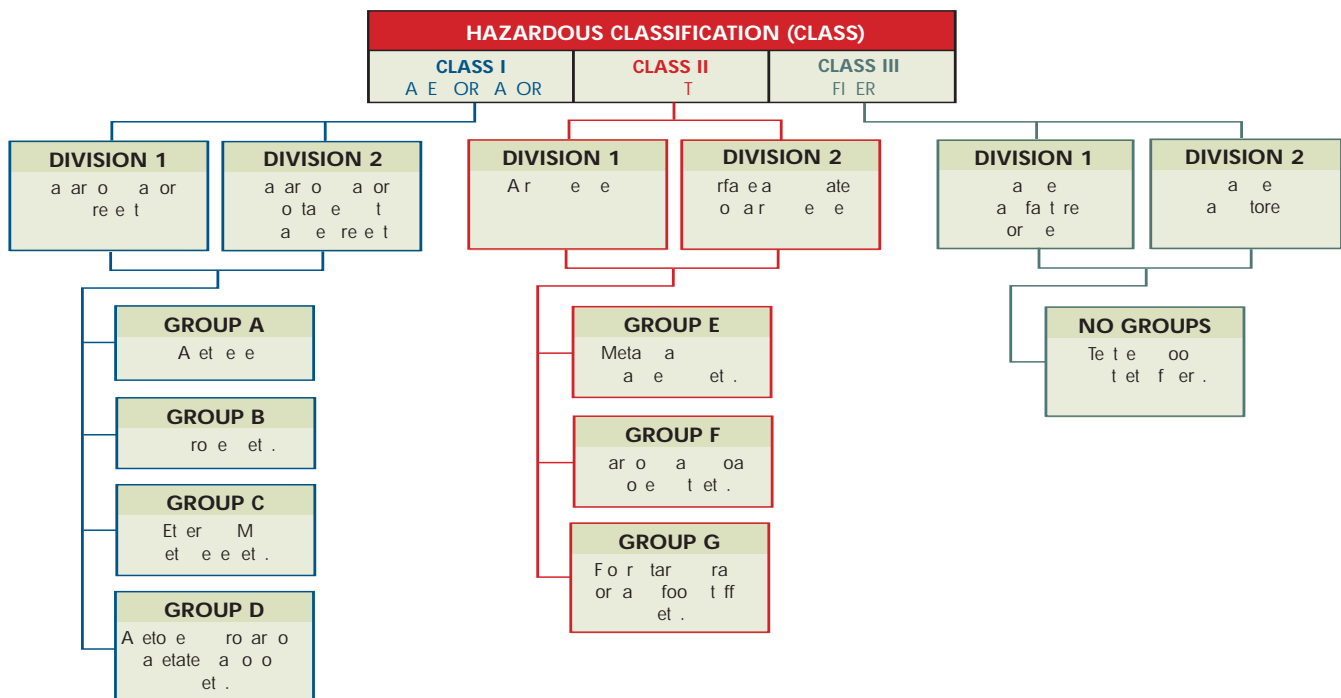
**Explosion Proof Motor for Hazardous Locations**

An **Explosion-Proof (XP)** motor is totally enclosed and designed to operate in an environment in which flammable gas or vapor is present. This is accomplished by limiting the external surface temperature of the motor below the ignition temperature of the hazardous gas or vapor surrounding the motor. It is very important that the end user order a motor with a proper temperature code. The enclosure is also designed to contain an explosion on the inside which may be caused by arcing, sparking, or abnormal heat. A flame caused by such a motor will not propagate the motor fits and ignite the gas or vapor surrounding the motor. An explosion-proof motor is suitable for operation in a **Class I** hazardous environment.

**Dust Ignition Proof Motor for Hazardous Locations**

A **Dust Ignition-Proof (DIP)** motor is a totally enclosed machine designed to operate safely in flammable dust environments. The motor enclosure prevents the admittance of dust inside the motor. Also, the motor is designed to operate cool enough as not to ignite any dust which might accumulate on the external surface of the motor. Should the motor become blanketed with dust and reduce its normal cooling, winding thermostats will take the motor offline before the motor's external surface temperature reaches the ignition temperature of the hazardous dust. A Dust Ignition Proof motor is suitable for operation in a **Class II** hazardous environment.

## NEMA CHARTS - HAZARDOUS &amp; NON-HAZARDOUS CLASSIFICATIONS



## ENCLOSURES FOR NON-HAZARDOUS LOCATIONS

## NEMA

Type 1	General Purpose – Indoor
Type 2	General Purpose – Indoor
Type 3	Dust-tight, Rain-tight and Sleet (Ice)-Resistant – Outdoor
Type 3R	Rainproof and Sleet (Ice)-Resistant – Outdoor
Type 3S	Dust-tight, Rain-tight and Sleet (Ice)-Proof – Outdoor
Type 4	Watertight and Dust-tight – Indoor and Outdoor
Type 4X	Watertight, Dust-tight and Corrosion-Resistant – Indoor and Outdoor
Type 4/12	Features of Type 4 and Type 12 combined
Type 5	Superseded by Type 12 for Control Apparatus
Type 6	Submersible, Watertight, Dust-tight Rainproof and Sleet (Ice)-Resistant – Indoor and Outdoor
Type 11	Corrosion-Resistant and Drip-proof – Oil immersed – Indoor
Type 12	Industrial Use – Dust-tight and Drip-tight – Indoor
Type 13	Oil-tight and Dust-tight – Indoor

## ENCLOSURES FOR HAZARDOUS LOCATIONS

Type 7	<b>Class I, Group A, B, C or D</b> – Indoor Hazardous Locations – Air-break equipment
Type 8	<b>Class I, Group A, B, C or D</b> – Indoor Hazardous Locations – Oil immersed equipment
Type 9	<b>Class II, Group E, F, or G</b> – Indoor Hazardous Locations – Air-Brake equipment
Type 10	Bureau of Mines



## TORQUE, SPEED, POWER &amp; EFFICIENCY

## TORQUE

The Full Load Torque of a motor is expressed as follows:

$$\text{Full-Load Torque (ft-lbs)} = \frac{\text{Power (Hp)} \times 5252}{\text{Full-Load Speed (RPM)}}$$

Since different loads present different torque requirements at starting (break-away), minimum (pull-up), break-down (pull-out), and full load, the National Electrical Manufacturers Association (NEMA) has defined 4 design classes of squirrel cage induction motors, which are described in the following table:

NEMA DESIGN	STARTING TORQUE	STARTING CURRENT	BREAK-DOWN TORQUE	FULL LOAD SLIP	TYPICAL APPLICATIONS
A	Normal	High	High	Low	Machine Tools
B	Normal	Normal	Normal	Normal	General Industrial
C	High	Normal	Normal	Normal	Loaded Compressors
D	Very High	Low	—	High	Punch Presses or Hoists

## SPEED &amp; SLIP

Synchronous Speed vs. Rotor Speed:

The actual operating speed of an AC induction motor is determined by the synchronous speed and slip. Slip is the difference between the speed of the rotating magnetic field (which is always synchronous) and the rotor speed. Slip generally increases with an increase in motor torque; therefore, actual operating speed generally decreases with an increase in motor torque. (See figures on next page.)

$$\text{Synchronous Speed (RPM)} = \frac{\text{Power Supply Frequency (Hz)} \times 120}{\text{Number of Poles (2, 4, 6, etc.)}}$$

## POWER

The mechanical power output of a motor is given by:

$$\text{Power (Hp)} = \frac{\text{Torque (in-lbs)} \times \text{Speed (RPM)}}{63,025}$$

$$\text{Power (kW)} = \frac{\text{Torque (N-m)} \times \text{Speed (Hz)}}{159.2}$$

## EFFICIENCY

Motor efficiency generally increases as the horsepower rating of the motor increases. High-efficiency motors can save energy and money. The chart below shows typical annual cost savings from using a high-efficiency motor over a standard efficiency motor.

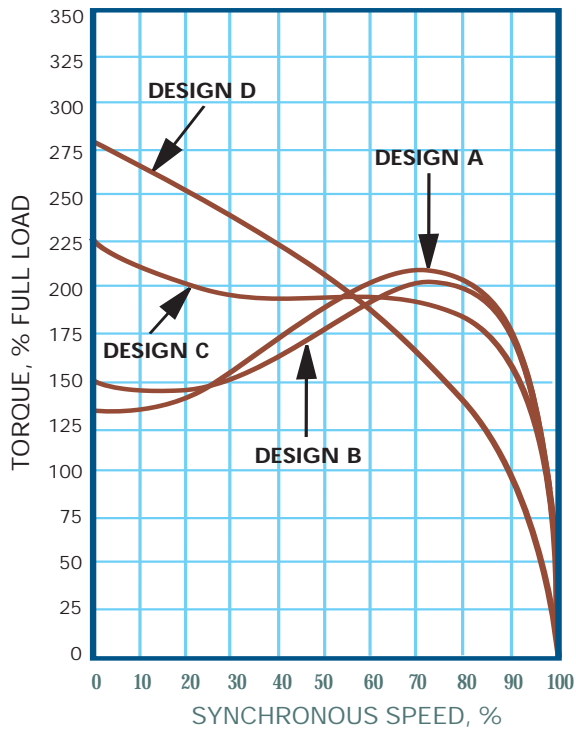
$$\text{Motor Efficiency (\%)} = \frac{\text{Mech. Output Power}}{\text{Input Power}} \times 100$$

Chart based on \$ .10 and 7,200 hours annual run time:  
kW-hr

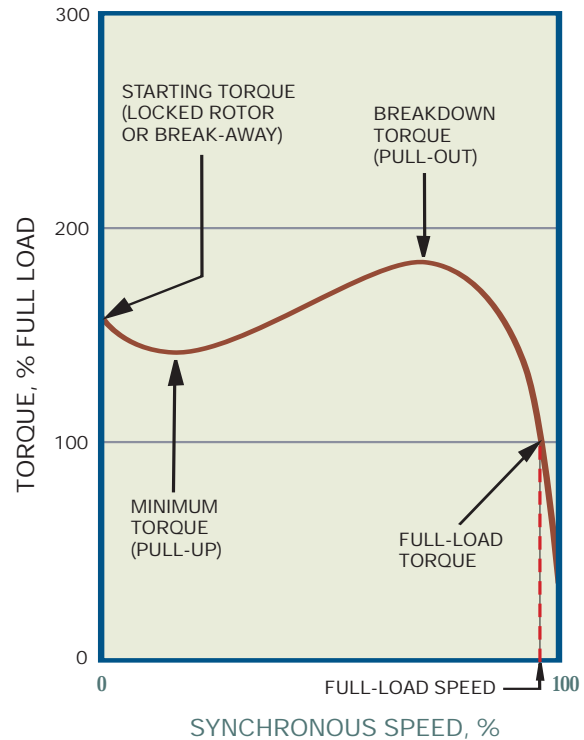
HP	HIGH EFFICIENCY MOTOR (%)	STANDARD EFFICIENCY MOTOR (%)	ELECTRICAL POWER DIFF. (W)	ENERGY SAVED (kW-hr)	ANNUAL SAVINGS (\$ US)
1	84.0	76.5	87.1	627	63
1.5	85.5	78.5	117	840	84
2	86.5	80.8	122	876	88
3	88.5	79.9	272	1960	196
5	88.5	83.1	274	1972	197
7.5	90.2	83.8	474	3411	341
10	90.2	85.0	506	3643	364
15	91.7	86.5	734	5282	528
20	92.4	87.4	924	6651	665
50	93.6	90.4	1411	10157	1016
100	95.4	91.6	3244	23357	2336

## TORQUE vs. SPEED CURVES & AIR MOTORS

**Typical Torque-Speed Curves for NEMA Design Motors**



**Typical Torque-Speed Curve for NEMA Design B Motor**



### AIR MOTORS

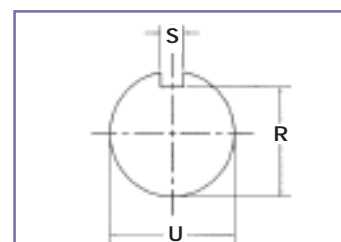
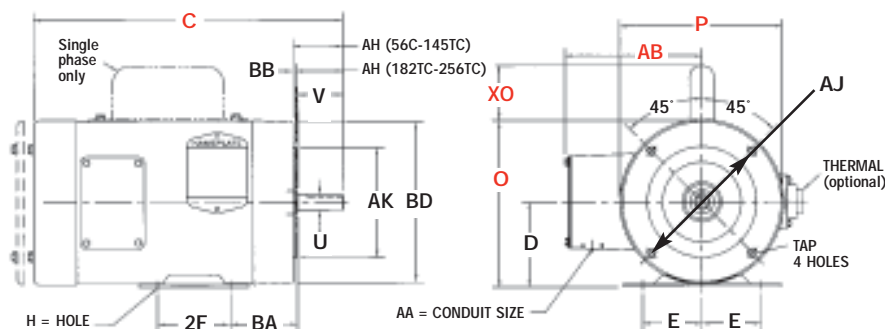
Air motors are an excellent choice for certain applications with pumps. Air motors are much lighter, inherently safe, and easy to install, which is especially important on applications which are commonly moved throughout the plant. Air motors are more expensive to operate relative to the energy they consume.

#### Advantages of Air Motors

- Air motors are compact and lightweight. Air motors weigh only a quarter as much and occupy only one sixth the space of an equivalent electric motor.
- The torque of an air motor increases with load.
- Air motors can be run indefinitely without over-heating or sustaining other damage. They can be started and stopped repeatedly to an almost unlimited extent.
- Torque and power output can be adjusted by varying the inlet air pressure.

## NEMA MOTORS - Quick Reference Chart

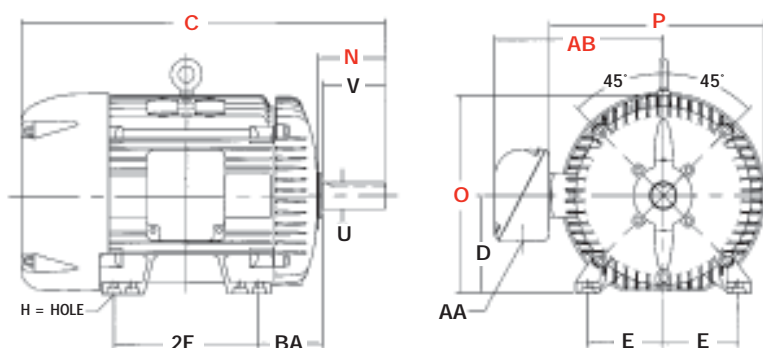
### C-FACE MOTOR (Close-Coupled)



Shaft Dimensions (Units: inches)

NEMA Shaft Dia.	Keyseat Dimensions	
U	R	S
3/8	21/64	Flat
1/2	29/64	Flat
5/8	33/64	3/16
7/8	49/64	3/16
1 1/8	63/64	1/4
1 3/8	113/64	5/16
1 5/8	113/32	3/8

### STANDARD MOTOR (Base Mounted)



Note: Drawings represent standard TEFC (totally enclosed fan cooled) motors. Dimensions C, N, O, P, AB & XO depend on the motor manufacturer.

BA Dimensions (Units: inches)

NEMA C-Face	BA
143TC & 145TC	2 3/4
182TC & 184TC	3 1/2
213TC & 215TC	4 1/4
254TC & 256TC	4 3/4

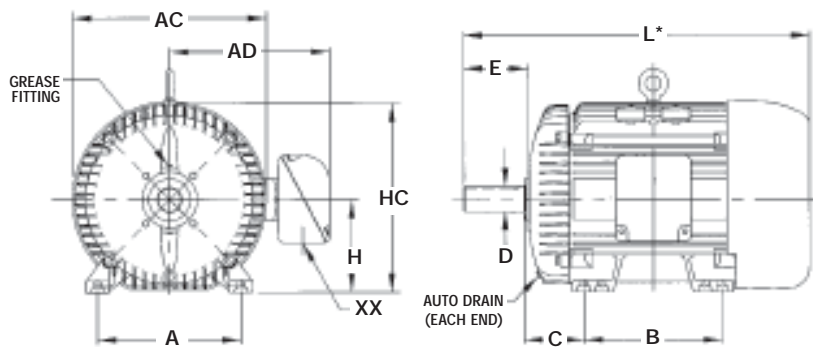
### Standard & C-Face Motor Dimensions (Units: inches)

NEMA FRAME		D	E	2F	H	U	V	AA	AH	AJ	AK	BA	BB	BD	TAP
Standard	C-Face														
42	42C	25/8	13/4	11 1/16	9/32 Slot	3/8	1 1/8	3/8	15/16	3 3/4	3	2 1/16	1/8	45/8	1/4 -20
48	48C	3	2 1/8	2 3/4	11/32 Slot	1/2	1 1/2	1/2	11 1/16	3 3/4	3	2 1/2	1/8	55/8	1/4 -20
56	56C	3 1/2	2 7/16	3	11/32 Slot	5/8	1 7/8	1/2	2 1/16	5 7/8	4 1/2	2 3/4	1/8	6 1/2	3/8 -16
56H	56HC	3 1/2	2 7/16	5	11/32 Slot	5/8	1 7/8	1/2	2 1/16	5 7/8	4 1/2	2 3/4	1/8	6 1/2	3/8 -16
143T	143TC	3 1/2	2 3/4	4	11/32	7/8	2 1/4	3/4	2 1/8	5 7/8	4 1/2	2 1/4	1/8	6 1/2	3/8 -16
145T	145TC	3 1/2	2 3/4	5	11/32	7/8	2 1/4	3/4	2 1/8	5 7/8	4 1/2	2 1/4	1/8	6 1/2	3/8 -16
182	182C	4 1/2	3 3/4	4 1/2	13/32	7/8	2 1/4	3/4	2 1/8	5 7/8	4 1/2	2 3/4	1/8	6 1/2	3/8 -16
184	184C	4 1/2	3 3/4	5 1/2	13/32	7/8	2 1/4	3/4	2 1/8	5 7/8	4 1/2	2 3/4	1/8	6 1/2	3/8 -16
182T	182TC	4 1/2	3 3/4	4 1/2	13/32	1 1/8	2 3/4	3/4	2 5/8	7 1/4	8 1/2	2 3/4	1/4	9	1/2 -13
184T	184TC	4 1/2	3 3/4	5 1/2	13/32	1 1/8	2 3/4	3/4	2 5/8	7 1/4	8 1/2	2 3/4	1/4	9	1/2 -13
213	213C	5 1/4	4 1/4	5 1/2	13/32	1 1/8	3	3/4	2 3/4	7 1/4	8 1/2	3 1/2	1/4	9	1/2 -13
215	215C	5 1/4	4 1/4	7	13/32	1 1/8	3	3/4	2 3/4	7 1/4	8 1/2	3 1/2	1/4	9	1/2 -13
213T	213TC	5 1/4	4 1/4	5 1/2	13/32	1 3/8	3 3/8	3/4	3 1/8	7 1/4	8 1/2	3 1/2	1/4	9	1/2 -13
215T	215TC	5 1/4	4 1/4	7	13/32	1 3/8	3 3/8	3/4	3 1/8	7 1/4	8 1/2	3 1/2	1/4	9	1/2 -13
254U	254UC	6 1/4	5	8 1/4	17/32	1 3/8	3 3/4	1	3 1/2	7 1/4	8 1/2	4 1/4	1/4	10	1/2 -13
256U	256UC	6 1/4	5	10	17/32	1 3/8	3 3/4	1	3 1/2	7 1/4	8 1/2	4 1/4	1/4	10	1/2 -13
254T	254TC	6 1/4	5	8 1/4	17/32	1 5/8	4	1	3 3/4	7 1/4	8 1/2	4 1/4	1/4	10	1/2 -13
256T	256TC	6 1/4	5	10	17/32	1 5/8	4	1	3 3/4	7 1/4	8 1/2	4 1/4	1/4	10	1/2 -13

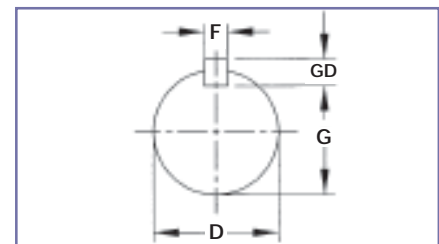
Note: BA dimensions in blue apply only for Standard motor frames. See separate chart above for C-Face motor BA dimensions.

## IEC MOTORS - Quick Reference Chart

### IEC MOTOR (B3 Foot Mount)



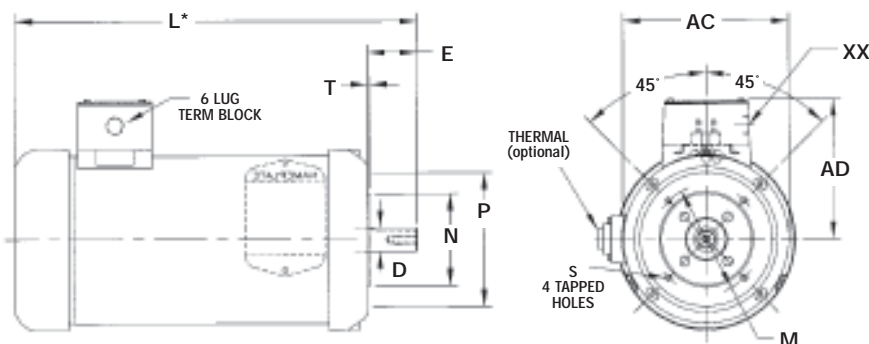
\* Contact motor manufacturer for "L" dimensions  
Drawings represent standard TEFC Motors  
(totally enclosed fan cooled).



Shaft Dimensions (Units: mm)

IEC Frame	D	G	F	GD
63	11	8.5	4	4
71	14	11	5	5
80	19	15.5	6	6
90	24	20	8	7

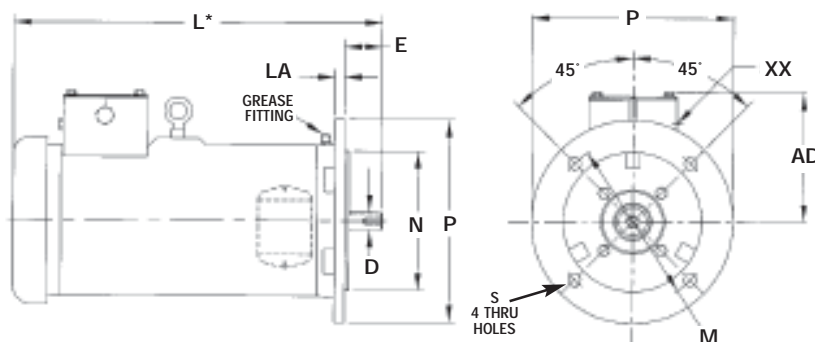
### IEC MOTOR (B14 Face)



### IP Protection Numbers

First # Protection Against Solid Objects	Second # Protection Against Liquids
0 No protection	0 No protection
1 Protection against solid objects up to 50 mm	1 Protection Against vertical droplets of water
2 Protection against solid objects up to 12 mm	2 Protection Against falling water up to 15° from the vertical
3 Protection against solid objects over 2.5 mm	3 Protection Against falling water up to 60° from the vertical
4 Protection against solid objects over 1 mm	4 Protection Against splashing water from all directions
5 Protection against dust	5 Protection against low-pressure jets of water from all directions
6 Totally protected against all dust	6 Protection against strong jets of water
	7 Protection against immersion
	8 Protection against submersion

### IEC MOTOR (B5 Flange)



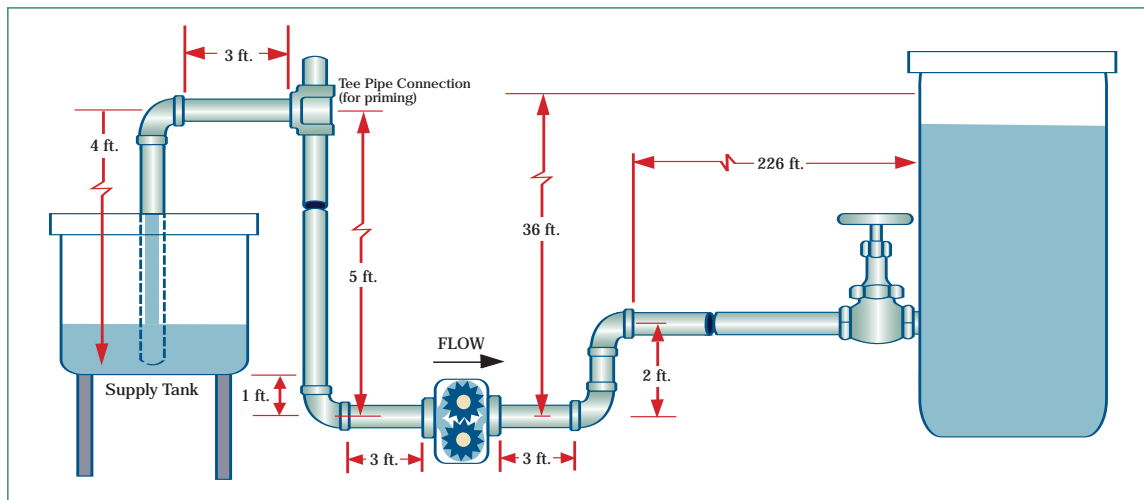
<b>IP22</b> =	Open Drip-Proof AC or DC motors
<b>IP54</b> =	All standard TEFC AC and DC motors
<b>IP55</b> =	All TEFC Chemical Processing, Dirty Duty and Washdown Duty Motors

\* "d" in chart below designates DC Motors

### IEC MOTOR (Units: mm/inches)

IEC FRAME	Type	B3 Foot Mounting				Shaft		B5 Flange						B14 Face					General			
		A	B	C	H	D	E	LA	M	N	P	S	T	M	N	P	S	T	AC	AD	HC	XX
63	300	100	80	40	63	11	23	8	115	95	140	9	3	75	60	90	M5	2.5	119	102	121	13
		3.937	3.150	1.570	2.480	.433	.906	.313	4.528	3.740	5.512	.354	.118	2.953	2.362	3.540		0.098	4.690	4.16d*	4.760	.500
71	300 & 400	112	90	45	71	14	30	8	130	110	160	10	3.5	85	70	105	M6	2.5	119	102	131	18
		4.409	3.543	1.770	2.800	.551	1.181	.313	5.118	4.331	6.299	.393	.138	3.347	2.756	4.130		0.098	4.690	4.16d*	5.140	.690
80	400 & 500	125	100	50	80	19	40	13	165	130	200	11	3.5	100	80	120	M6	3	145d	102	149d*	21d*
		4.921	3.937	1.969	3.150	.748	1.575	.500	6.496	5.118	7.874	.430	.138	3.937	3.150	4.724		.118	5.69d*	4.16d*	5.38d*	.884d*
90	S	140	100	56	90	24	50	13	165	130	200	12	3.5	115	95	140	M8	3	145d	102	162d*	21d*
		5.511	3.937	2.205	3.543	.945	1.969	.500	6.496	5.118	7.874	.472	.138	4.530	3.740	5.512		.118	6.61d*	5.120	6.810	.880
90	L	140	125	56	90	24	50	13	165	130	200	12	3.5	115	95	140	M8	3	144d*	107d*	165d*	21d*
		5.511	4.921	2.205	3.543	.945	1.969	.500	6.496	5.118	7.874	.472	.138	4.530	3.740	5.512		.118	5.69d*	4.25d*	6.53d*	.844d*

## GEAR PUMP &amp; MOTOR SELECTION EXAMPLE



Conditions of Service:

Flow Rate: 10 GPM  
 Specific Gravity (SG) = 1.2  
 Pipe Size = 1 inch  
 Viscosity = 300 SSU

$P_B$  = Barometric Pressure = 14.7 PSIA  
 $L_H$  = Static Suction Pressure  
 $h_f$  = Friction Loss  
 $V_p$  = Vapor Pressure

**PROBLEM 1:** Find the **SUCTION PRESSURE** and **DISCHARGE PRESSURE** of the system above.

**STEP 1: SUCTION PRESSURE:** This is the pressure at the inlet side of the pump.  
 For the system above, Suction Pressure is given as follows:

$$\text{Suction Pressure} = \text{Barometric Pressure } (P_B) + \text{Static Suction Pressure } (L_H) - \text{Friction Loss } (h_f)$$

Note: If pump is below liquid level,  $L_H$  has a positive value;  
 If pump is above liquid level,  $L_H$  has a negative value.

Static Suction Pressure: This is the amount of pressure required to lift the fluid or the amount of pressure the fluid elevation imparts on the pump inlet. Since the pump is below the level of the fluid by 1 ft., the net static suction pressure is positive.

From our example: the pump is 1 ft. below the liquid level.

$$\text{Static Suction Pressure } (L_H) = (1 \text{ ft.}) \times \text{Specific Gravity } (1.2) \times (0.433 \text{ PSI/ft.}) = 0.52 \text{ PSI}$$

Friction Loss depends on:

- Total equivalent length of pipe
- Diameter of pipe & pipe material
- Viscosity of fluid
- Flow rate

Reference the Nomograph in the Fluid Section to find equivalent pipe length for fittings and valves.  
 Reference Friction Loss Modulus graph for 1" steel pipe to determine the Loss Modulus for a Fluid with a Viscosity of 300 SSU at 10 GPM Flow. Friction Loss Modulus is found from the graph to be 15 for 1" pipe at 10 GPM Flow for a Fluid with a Viscosity of 300 SSU. This value is for 100 ft. length of pipe: therefore, divide by 100.

$$\text{Friction Loss} = \text{Equivalent Pipe Length} \times \text{Specific Gravity} \times \text{Friction Loss Modulus}$$

From our example:

$$\begin{aligned} \text{Total equivalent pipe length} &= \text{Actual pipe length} & (4 \text{ ft.} + 3 \text{ ft.} + 5 \text{ ft.} + 3 \text{ ft.}) &= 15 \text{ ft.} \\ &+ \text{Equivalent length of (2) elbows} & (2 \times 4 \text{ ft.}) &= 8 \text{ ft.} \\ &+ \text{Equivalent length of (1) pipe Tee} & (1 \times 4 \text{ ft.}) &= 4 \text{ ft.} \end{aligned}$$

$$\text{Total equivalent pipe length} = 27 \text{ ft.}$$

$$\text{Friction Loss} = \frac{27 \text{ ft} \times (1.2 \text{ SG}) \times 15}{100} = 4.86 \text{ PSI}$$

$$\text{Suction Pressure} = P_B + L_H - h_f$$

$$\text{SUCTION PRESSURE} = 14.7 + (0.52) - 4.86 = 10.36 \text{ PSIA} \rightarrow 10.36 \text{ PSIA} = -4.34 \text{ PSIG}$$

If Vapor Pressure ( $V_p$ ) = 2.0 PSIA, then

$$\text{Net Positive Suction Pressure Available} = 10.36 \text{ PSIA} - 2.0 \text{ PSIA} = 8.36 \text{ PSIA}$$

## GEAR PUMP &amp; MOTOR SELECTION EXAMPLE (continued)

**STEP 2: DISCHARGE PRESSURE:** The discharge pressure is the sum of the static pressure required to pump the liquid to a desired height and friction loss in the pipe created by the resistance of the moving fluid.

$$\text{Discharge Pressure} = \text{Static Pressure} + \text{Friction Loss}$$

$$\text{Static Pressure} = (36 \text{ ft.}) \times \text{Specific Gravity (1.2)} \times (.433 \text{ PSI/ft.}) = 18.71 \text{ PSI}$$

$$\text{Friction Loss} = \text{Equivalent Pipe Length} \times \text{Specific Gravity} \times \text{Friction Loss Modulus}$$

Friction Loss Modulus is found from the graph to be 15 for 1" pipe at 10 GPM Flow for a Fluid with a Viscosity of 300 SSU. This value is for 100 ft. length of pipe: therefore, divide by 100.

$$\begin{aligned} \text{Total equivalent pipe length} &= \text{Actual pipe length} && (231 \text{ ft.}) \\ &+ \text{Equivalent length for valve} && (17 \text{ ft.}) \\ &+ \text{Equivalent length for elbows} && (2 \times 4 \text{ ft.} = 8 \text{ ft.}) \end{aligned}$$

$$\text{Total equivalent pipe length} = 256 \text{ ft.}$$

$$\text{Friction Loss} = \frac{256 \text{ ft} \times (1.2 \text{ SG}) \times 15}{100} = 46.08 \text{ PSI}$$

$$\text{Discharge Pressure} = \text{Static Pressure} + \text{Friction Loss}$$

$$\text{DISCHARGE PRESSURE} = 18.71 + 46.08 = 64.79 \text{ PSIG}$$

**PROBLEM 2:** What Liquiflo gear pump should be selected for the service conditions of Problem 1?

**SOLUTION:** Since the flow rate is over 3.5 GPM, this excludes the 2 and 4-Series pumps. The differential pressure is less than 100 PSI, so an H-Series pump is not required, although it can be used to extend the working life of the pump. The smallest 3-Series pump that can produce a flow of 10 GPM @ ~70 PSI  $\Delta P$  with a moderate viscosity fluid (300 SSU = 65 cSt = 78 cP) is a 37F gear pump operating at approximately 1625 RPM (based on 100 cP performance curve).

**PROBLEM 3:**

a) What is the Brake Horsepower (BHP) required for the service conditions of Problem 1? Assume a pump efficiency of 35%.

b) What Standard Horsepower Motor should be selected for this application?

**STEP 1:** Collect the required data:

$$\begin{aligned} Q &= \text{Flow rate} && = 10 \text{ GPM} \\ P_{\text{suction}} &&& = -4.34 \text{ PSIG} \\ P_{\text{discharge}} &&& = 64.79 \text{ PSIG} \\ E &= \text{Efficiency} && = 0.35 \text{ (35\%)} \end{aligned}$$

**STEP 2:** Calculate the Differential Pressure.

$$\Delta P = P_{\text{discharge}} - P_{\text{suction}} = 64.79 \text{ PSIG} - (-4.34 \text{ PSIG}) = 69.13 \text{ PSI}$$

**STEP 3:** Calculate the Fluid Horsepower.

$$\text{FHP} = \frac{\Delta P [\text{PSI}] \times Q [\text{GPM}]}{1714} = \frac{69.13 [\text{PSI}] \times 10 [\text{GPM}]}{1714} = 0.403$$

**STEP 4:** Calculate the Brake Horsepower.

$$\text{BHP} = \frac{\text{FHP}}{E} = \frac{0.403}{0.35} = 1.15$$

**What Motor should be chosen for this application?**

In addition to determining the motor enclosure type, the mounting method and the input power requirements for the motor, the mechanical output power rating must be determined. This power rating must meet or exceed the BHP of the application, but should be the smallest standard motor available. The closest standard Hp motors to 1.15 BHP are 1.0 Hp and 1.5 Hp motors: therefore:

Select a 1.5 Hp motor



## MATERIALS OF CONSTRUCTION FOR LIQUIFLO GEAR PUMPS

## H-SERIES &amp; 3-SERIES GEAR PUMPS ( ea e Ma r e)

HOUSINGS	GEARS	WEAR PLATES	BEARINGS	SHAFTS
316 SS Alloy-C <sup>1</sup> Alloy-20 <sup>2</sup>	316 SS Alloy-C <sup>1</sup> PEEK <sup>3</sup> Ryton <sup>4</sup> Teflon <sup>5</sup> Carbon <sup>2</sup> Delrin <sup>2</sup>	Carbon 60 Silicon Carbide <sup>6</sup> PEEK <sup>3</sup> Teflon <sup>5</sup>	Carbon 60 Silicon Carbide <sup>6</sup> PEEK <sup>3</sup> Teflon <sup>5</sup>	316 SS Alloy-C <sup>1</sup> TC-Coated CO-Coated

## 2-SERIES GEAR PUMPS (Ma r e)

HOUSINGS	GEARS	WEAR PLATES & BEARINGS COMBINATION	SHAFTS
316 SS	316 SS Alloy-C <sup>1</sup> PEEK <sup>3</sup>	Carbon 60 PEEK <sup>3</sup>	316 SS TC-Coated CO-Coated

## 4-SERIES GEAR PUMPS (Ma r e)

HOUSINGS	GEARS	WEAR PLATES	BEARINGS	SHAFTS
316 SS Alloy-C <sup>1</sup> Titanium	316 SS Alloy-C <sup>1</sup> Titanium PEEK <sup>3</sup> Ryton <sup>4</sup> Teflon <sup>5</sup>	Carbon 60 Silicon Carbide <sup>6</sup> PEEK <sup>3</sup> Teflon <sup>5</sup>	Carbon 60 Silicon Carbide <sup>6</sup> PEEK <sup>3</sup> Teflon <sup>5</sup>	316 SS Alloy-C <sup>1</sup> TC-Coated CO-Coated Titanium, TO-Coated

MAX<sup>®</sup>-SERIES GEAR PUMPS ( ea e Ma r e)

HOUSINGS	GEARS	WEAR PLATES	BEARINGS	SHAFTS
316 SS Titanium <sup>8</sup>	17-4 PH SS <sup>7</sup> 316 SS Titanium <sup>8</sup> PEEK <sup>3</sup> Teflon <sup>5</sup>	Carbon 60 Silicon Carbide <sup>6</sup> PEEK <sup>3</sup> Teflon <sup>5</sup>	Carbon 60 Silicon Carbide <sup>6</sup> PEEK <sup>3</sup>	17-4 PH SS <sup>7</sup> 316 SS TC-Coated CO-Coated Titanium, TO-Coated <sup>8</sup>

## PFA-LINED GEAR PUMPS (Ma r e)

HOUSINGS	GEARS	WEAR PLATES	BEARINGS	SHAFTS
PFA-Lined SS <sup>9</sup>	PEEK <sup>3</sup> Ryton <sup>4</sup> Teflon <sup>5</sup>	Silicon Carbide <sup>6</sup>	Silicon Carbide <sup>6</sup>	Silicon Carbide <sup>6</sup>

## Footnotes:

- 1) C-22 grade for Housings; C-276 grade for internal components.
- 2) Available only in 3-Series. (Delrin is available for repair or replacement only.)
- 3) Bearing Grade PEEK (poly-ether-ether-ketone with 30% total filler of carbon fiber, graphite and PTFE).
- 4) 25% Glass-filled & 15% PTFE-filled PPS (poly-phenylene sulfide).
- 5) 25% Glass-filled PTFE (poly-tetra-fluoro-ethylene).
- 6) Self-sintered Silicon Carbide (SiC).
- 7) Available only as integral Gear-Shaft combination for Sealed pumps.
- 8) Available only for Mag-drive pumps.
- 9) PFA (per-fluoro-alkoxy) plastic & 300-Series Stainless Steel.

## MATERIALS OF CONSTRUCTION FOR LIQUIFLO GEAR PUMPS *(continued)*

**NOTE:** The chemical compatibility of a material is highly dependent on the temperature and concentration of the fluid and the grade of the material. Specific corrosion and mechanical data must be known to determine if a material is suitable for the application. The following information gives general guidelines for material selection. Consult Liquiflo Application Engineering for specific applications.

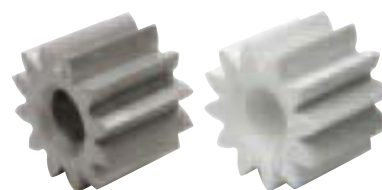
### HOUSINGS:

- 316 SS:** 316 Stainless Steel is an iron-based alloy containing significant amounts of nickel, chromium and molybdenum. 316 SS offers both high strength and high corrosion resistance, and is the primary choice for pumping most chemicals.
- Alloy-20:** Carpenter-20® is a stainless steel with significantly higher amounts of nickel and chromium than 316 SS, and also contains a specific amount of copper (3-4%). Alloy-20 offers improved chemical resistance over 316 SS and is an economical alternative to Alloy-C for pumping Sulfuric Acid.
- Alloy-C:** Hastelloy-C® is a nickel-based alloy containing significant amounts of chromium and molybdenum. It is used for pumping certain corrosive chemicals that are not compatible with stainless steels. Alloy-C is typically used to pump Brine, aqueous Chlorine, Calcium Hydroxide, Copper Sulfate, Chlorosulfonic Acid, Chromic Acid, Hydrobromic Acid, Hydrochloric Acid, Hydrofluoric Acid, Concentrated Nitric Acid, Concentrated Phosphoric Acid and Sulfuric Acid.
- Titanium:** Elemental Titanium is used for pumping certain chemicals that are not compatible with stainless steels or nickel-based alloys. Titanium is typically used for pumping Sodium Hypochlorite and aqueous Ferric Chloride.
- PFA-Lined SS:** **PFA-lined pumps can be used on virtually any chemical fluid up to 212°F (100°C) at differential pressures up to 100 PSI.** PFA (per-fluoro-alkoxy) polymer resin offers excellent chemical resistance to extremely corrosive liquids that are known to attack metallic pumps. These services are typically strong inorganic acids, bases and salts, such as Hydrochloric Acid, Hydrofluoric Acid, Sulfuric Acid, Sodium Hydroxide and Sodium Hypochlorite. The housing base material is 300-Series Stainless Steel, which provides structural integrity and resistance to corrosive environments. PFA-lined pumps are ideal for high-purity services (where contact with metallic parts must be avoided) and they are an economical alternative to pumps made with expensive metal alloys.



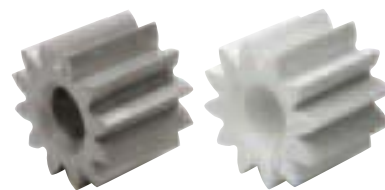
### GEARS:

- 316 SS:** 316 SS is typically used for the drive gear material when 316 SS housings are selected. When pumping thin, non-lubricating fluids (e.g., water, solvents, dilute solutions, etc.), a nonmetal idler gear is required to produce a normal wear rate of the gears. For moderately viscous fluids (10-99 cP), double metal gears can be used for special combinations of speed and differential pressure. For viscous fluids of 100 cP or higher, double metal gears can always be used. If both gears are made of 316 SS, galling of the gears can occur. To prevent galling, an Alloy-C drive gear is typically run against a 316 SS idler gear when double metal gears are used.
- Alloy-C:** Alloy-C is typically used for the drive gear material when Alloy-C or Alloy-20 housings are selected. The same rules given above for gear combinations apply to Alloy-C gears, except that galling is not a problem for this material, so double Alloy-C gears can always be used if the viscosity is 100 cP or higher.
- Titanium:** Titanium is typically used for the drive gear material when Titanium pumps are required. The idler gear is typically a plastic material, such as PEEK, Ryton or Teflon.
- 17-4 PH SS:** 17-4 Precipitate Hardened Stainless Steel drive and idler gears are used with 316 SS housings and on non-aggressive viscous fluids, such as polymers and oils. **The gears are manufactured as an integral gear-shaft combination.** 17-4 PH SS material is significantly harder than 316 SS and will resist galling of the gears. The gears can be applied at differential pressures up to 350 PSI for viscosities down to 20 cP. For slightly lower viscosities, 10-15 cP, the gears can be applied up to 150 PSI.



MATERIALS OF CONSTRUCTION FOR LIQUIFLO GEAR PUMPS *(continued)*GEARS: *(continued)*

- PEEK:** PEEK is the primary choice when nonmetal gears are required, due to its overall properties of high strength, high temperature capability, low thermal expansion, low wear characteristics and broad chemical resistance. PEEK is not compatible with some strong acids, halogens and halogenated compounds. The grade used by Liquiflo contains carbon and PTFE, which also needs to be checked for chemical compatibility when considering using PEEK. PEEK gears can be applied at differential pressures up to 175 PSI for 1 cP fluids at room temperature. Pressure-derating below 175 PSI is required for higher viscosities and temperatures. The maximum use temperature for a PEEK gear is **400°F**. At temperatures over 120-200 °F (depends on the specific pump model), the gears will need to be trimmed in length and diameter to account for the thermal expansion of the material.
- Ryton®:** Ryton is typically used as the idler gear material in low-viscosity applications where PEEK is not acceptable. Ryton has good overall chemical resistance, good strength and low thermal expansion, but is much more brittle than PEEK. Ryton is attacked by some strong acids and bases, and halogens. It is also questionable in some halogenated compounds. The grade used by Liquiflo contains glass and PTFE, which also needs to be checked for chemical compatibility when considering using Ryton. Ryton gears can be applied at differential pressures up to 175 PSI for 1 cP fluids at room temperature. Pressure-derating below 175 PSI is required for higher viscosities and temperatures. The maximum use temperature for a Ryton gear is **300°F**. At temperatures over 120-200 °F (depends on the specific pump model), the gears will need to be trimmed in length and diameter to account for the thermal expansion of the material.
- Teflon®:** Teflon is extremely chemically resistant but it has limited use as a gear due to its low strength and high thermal coefficient of expansion (TCE). The primary use for Teflon is in low viscosity, very aggressive or ultra pure services, at ambient temperature and moderate pressures, up to 50 PSI. Pressure-derating below 50 PSI is required for elevated viscosities and temperatures. Teflon is not recommended for radioactive service and may swell in halogenated solvents. The grade used by Liquiflo contains glass (for reinforcement), which also needs to be checked for chemical compatibility when considering using Teflon. The maximum use temperature for a Teflon gear is **230°F**. At temperatures over 80-100 °F (depends on the specific pump model), the gears will need to be trimmed in length and diameter to account for the thermal expansion of the material.
- Carbon:** Carbon is typically used as the non-metal idler gear material for compatible low-viscosity fluids at temperatures over 400°F (where a plastic gear is no longer an option) and moderate pressures, typically below 50 PSI. This carbon is a different grade than that used for the wear plates and bearings, and does not have the same chemical resistance. Carbon gears can be used up to the pump's maximum temperature (**500°F**). When using a carbon idler gear at temperatures over 200-300 °F (depends on the specific pump model), the gear bore will need to be trimmed (i.e., slightly increased in diameter) to account for the thermal expansion of the idler shaft.



## WEAR PLATES:

- Carbon 60:** Carbon 60 is the primary choice for the wear plates due to its excellent combination of chemical and physical properties: good overall chemical resistance, self-lubricating, low thermal coefficient of expansion (TCE) and wide temperature range. Carbon 60 is a carbon grade that offers greater resistance to acids than non-graphite grades, and is therefore also known as "acid-grade carbon." Carbon 60 wear plates can be applied up to the pump's maximum temperature (**500°F**).
- Silicon Carbide:** Silicon Carbide (SiC) wear plates have excellent chemical resistance, wear resistance and compressive strength, and negligible thermal expansion. The self-sintered SiC grade used by Liquiflo is resistant to both Sodium Hydroxide and Hydrofluoric Acid, unlike Ceramic Aluminum Oxide. SiC wear plates can only be used with a metallic gear if the fluid viscosity is above 100 cP. They can be used with double plastic gears regardless of the viscosity. They are typically used with double metal gears on viscous fluids if a small amount of particulate is expected. They are used with double plastic gears for high purity services such as Deionized Water or just for their excellent wear characteristic against the plastic gears. SiC wear plates can be applied up to the pump's maximum temperature (**500°F**).
- PEEK:** PEEK wear plates are typically used in conjunction with PEEK bearings and when other materials are not acceptable. These wear plates are comprised of Bearing Grade PEEK, so the filler materials must also be considered when determining the suitability of PEEK for an application. The maximum use temperature for PEEK wear plates is **450°F**.
- Teflon®:** Teflon is extremely chemically resistant but it has limited use as a wear plate due to its low strength and high thermal coefficient of expansion (TCE). The primary use for Teflon is in low viscosity, very aggressive or ultra pure services, at ambient temperature and moderate pressures, up to 50 PSI. Teflon is not recommended for radioactive service and may swell in halogenated solvents. The maximum use temperature for Teflon wear plates is **260°F**.



MATERIALS OF CONSTRUCTION FOR LIQUIFLO GEAR PUMPS *(continued)*

## BEARINGS:

- Carbon 60:** Carbon 60 is the primary choice for the bearings due to its excellent combination of chemical and physical properties: good overall chemical resistance, self-lubricating, low thermal coefficient of expansion (TCE) and wide temperature range. In addition carbon is hard, and has good compressive strength, heat transfer and wear properties, making it an excellent bearing choice. This is the only available bearing material that can tolerate short periods of dry running. Carbon 60 is a carbon grade that offers greater resistance to acids than non-graphite grades, and is therefore also known as "acid-grade carbon." Carbon 60 bearings can be applied up to the pump's maximum temperature (**500°F**). When using Carbon 60 bearings at elevated temperatures, the bearing bores may need to be trimmed (i.e., slightly increased in diameter) to account for the thermal expansion of the shafts.
- Silicon Carbide:** Self-sintered Silicon Carbide (SiC) bearings have excellent chemical resistance, wear resistance and compressive strength, and negligible thermal expansion. SiC bearings must be run against a hardened shaft. When used against a TC-coated shaft (or a SiC shaft), the best life is achieved. SiC is extremely hard and has no tolerance for dry running. SiC bearings can be applied up to the pump's maximum temperature (**500°F**). When using SiC bearings at elevated temperatures, the bearing bores may need to be trimmed (i.e., slightly increased in diameter) to account for the thermal expansion of the shafts.
- PEEK:** PEEK bearings have ~20% lower compressive strength than Carbon 60 bearings and are typically used when other bearing materials are not acceptable. The grade used by Liquiflo is called Bearing Grade PEEK, which contains carbon fiber, graphite and PTFE. These filler materials increase lubrication and enable the bearing to handle higher PV loads, but they must also be checked for chemical compatibility when considering using PEEK. The maximum use temperature for PEEK bearings is **350°F**. When using PEEK bearings at elevated temperatures, the bearings may need to be trimmed to account for the thermal expansion of the material.
- Teflon®:** Teflon is extremely chemically resistant but it has limited use as a bearing material due to its low compressive strength, high thermal expansion and tendency to cold flow under load. The primary use for Teflon is in low viscosity, very aggressive or ultra pure services, at ambient temperature and moderate pressures, up to 50 PSI. Teflon is not recommended for radioactive service and may swell in halogenated solvents. The grade used by Liquiflo contains glass (for reinforcement), which also needs to be checked for chemical compatibility when considering using Teflon. In addition, the glass filler can be abrasive to uncoated metal shafts, so hard-coated shafts are recommended for thin fluids. The maximum use temperature for Teflon bearings is **180°F**. When using Teflon bearings at elevated temperatures, the bearings may need to be trimmed to account for the thermal expansion of the material.



## SHAFTS:

- 316 SS & Alloy-C, Uncoated:** 316 SS drive and idler shafts are used when 316 SS housings are selected. Alloy-C drive and idler shafts are used when Alloy-C or Alloy-20 housings are selected. Uncoated 316 SS and Alloy-C shafts are relatively soft and subject to contact wear at the bearing-shaft interfaces, especially at times when hydrodynamic film lubrication is lost or when PV loads are high. Since these conditions frequently occur during normal pump operation, **Liquiflo recommends using hard-coated shafts (TC or CO) for virtually all applications.** Hardened shafts will significantly reduce the wear rate and extend pump life.
- TC-Coated:** 316 SS and Alloy-C shafts are available with Tungsten Carbide (TC) coating in the areas of the shaft that contact the interior of the bearings (journal-bearing areas). TC-coated shafts are the primary choice for hardened shafts because of the hardness, bond strength and wide temperature range of the coating material. Unlike the ceramic coatings, the metal matrix allows the coating to expand and contract with the base metal of the shaft, so it doesn't tend to delaminate and crack like the ceramic. The metal matrix (Ni with a trace of Co) of the TC coating can sometimes be a problem as it can be attacked by the fluid or act as a catalyst and cause the decomposition of the fluid, as can occur with Hydrogen Peroxide service. TC-coated shafts can be applied up to the pump's maximum temperature (**500°F**).
- CO-Coated:** 316 SS and Alloy-C shafts are available with Chrome Oxide (CO) coating in the areas of the shaft that contact the interior of the bearings (journal-bearing areas). The ceramic CO is very chemically resistant but is not as hard as Tungsten Carbide, and therefore not as wear resistant. For Hydrogen Peroxide service, CO-coated shafts should be used in lieu of TC-coated shafts to prevent the catalytic decomposition of the fluid, which can produce H<sub>2</sub> and O<sub>2</sub> gases. CO-coated shafts have limited temperature capability because the TCE of the ceramic coating is much lower than that of the base metal of the shaft. This can cause the ceramic coating to delaminate and crack at high or low temperatures. The maximum use temperature for CO-coated shafts is **250°F**.
- Titanium, TO-Coated:** Titanium (Ti) shafts are used exclusively in Titanium pumps. Ti shafts are produced with Titanium Oxide (TO) coating in the areas of the shaft that contact the interior of the bearings (journal-bearing areas). TO-coated shafts are typically used with Carbon 60 or Silicon Carbide bearings to provide an effective wear-resistant bearing-shaft combination. The maximum use temperature for TO-coated shafts is **250°F**, due to the negligible thermal expansion of the ceramic coating.



## MATERIAL SELECTION GUIDELINES FOR BEARINGS & SHAFTS

### Bearing & Shaft Material Selection for Liquiflo Gear Pumps

Successful pump operation is based on matching the capabilities of the pump with the conditions of the application. In constructing a gear pump, one of the most important things to consider is the selection of the bearing and shaft materials. This section explains the relation of the bearing and shaft materials to pump performance and longevity, and describes a standard method for selecting the optimum materials for the application – both from the standpoint of tribology (i.e., friction, lubrication and wear).

### Differential Pressure & Shaft-Bearing Contact

The *differential pressure* ( $\Delta P$ ) across the gear pump is the difference in the discharge pressure ( $P_d$ ) and the suction pressure ( $P_s$ ):

$$\Delta P = P_d - P_s$$

The discharge and suction pressures can be accurately measured using calibrated gauges located as close as possible to the pump outlet and inlet, respectively. With normal pump operation, the discharge pressure is greater than the suction pressure. This results in a net positive differential pressure which is the load resistance that the pump must work against. The discharge pressure tends to push the shafts towards the suction side of the pump, which can cause the shafts to contact the bearings. *Shaft-bearing contact* and the capability of the product-lubricated bearings are the main reasons why differential pressure must be limited in gear pumps.

### Bearings & Lubrication

Gear pumps operating continuously rely on the pumped product to fully support the load at the bearing-shaft interfaces through viscous forces alone. This type of lubrication is called *hydrodynamic lubrication* or *fluid film lubrication*. When the bearings are operating with hydrodynamic lubrication, the fluid film is sufficiently formed to support the rotating shaft and prevent contact with the bearings. The higher the viscosity, the better the fluid film is at supporting the shaft. In theory, any bearing and shaft materials can be used because the fluid film is preventing the contact of parts.

### Characteristic Number

The bearing *characteristic number* is a dimensionless quantity defined by  $\mu n / P$ , where  $\mu$  is the fluid's absolute viscosity,  $n$  is the shaft speed and  $P$  is the **pressure on the projected area of the bearing** (not the same as  $\Delta P$ ). The characteristic number is an indication of the quality of fluid film lubrication at the pump's operating speed. The characteristic number ( $Z$ ) is related to the fluid viscosity, pump speed and differential pressure by the following equation:

$$Z = \frac{k_1 \cdot \mu \cdot n}{\Delta P}$$

Here  $k_1$  is a constant dependent on the geometry of the specific pump model,  $\mu$  is the fluid viscosity in cP,  $n$  is the shaft speed in RPM and  $\Delta P$  is the differential pressure in PSI. From the above equation, it can be seen that film lubrication improves with increasing viscosity and speed, but degrades with increasing differential pressure.

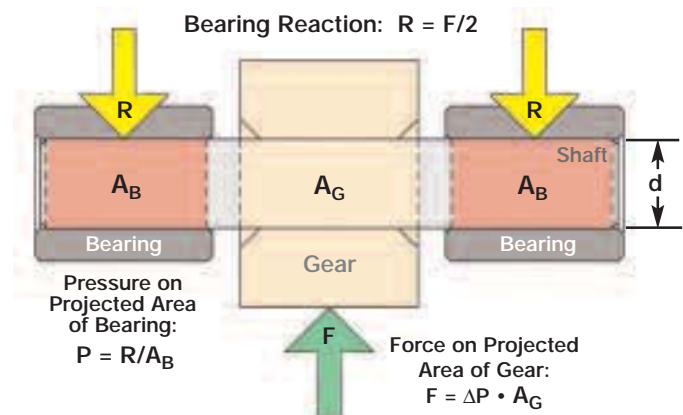
If the viscosity is low and the fluid film is thin or absent, it cannot support the shaft. The differential pressure will then displace the shaft from the higher-pressure discharge side to the lower-pressure suction side, and the parts will contact. In this case, the bearing and shaft materials are extremely important for resisting wear.

### PV Value

The dynamic bearing loading is given by the *PV value*, where  $P$  is the same as described above and  $V$  is the **peripheral velocity** (or surface speed) of the rotating shaft over the inner surface of the stationary bearing. **PV** (the product of  $P$  and  $V$ ) is related to the differential pressure and pump speed by the following equation:

$$PV = k_2 \cdot \Delta P \cdot n$$

Here  $k_2$  is a constant dependent on the geometry of the specific pump model,  $\Delta P$  is the differential pressure in PSI and  $n$  is the shaft speed in RPM. The above formula is derived below (refer to diagram at right):



### Dynamic Bearing Loading: PV Value or PV Load

$A_B$  = Projected area of bearing;  $A_G$  = Projected area of gear  
 $d$  = diameter of shaft (in);  $C$  = Circumference of shaft =  $\pi \times d$   
 $n$  = shaft speed (RPM = rev/min = min<sup>-1</sup>)

$$P = R/A_B = (F/2)/A_B = ((\Delta P \cdot A_G)/2)/A_B = (A_G/(2A_B)) \times \Delta P \text{ (PSI)}$$

$$V = C \times n = \pi \times d \text{ (in)} \times n \text{ (min}^{-1}) \times (1 \text{ ft}/12 \text{ in}) = (\pi d/12) \times n \text{ (ft/min)}$$

$$PV = ((\pi d A_G)/(24 A_B)) \times \Delta P \times n = k_2 \times \Delta P \times n \text{ (PSI-ft/min)}$$



## MATERIAL SELECTION GUIDELINES FOR BEARINGS &amp; SHAFTS (continued)

## PV Limits for Common Bearing Materials

The ability of different materials to resist frictional wear is characterized by the *PV limit*, which is the maximum PV load a material can safely withstand based on certain conditions. In addition to the PV limit, the *load capacity* (PSI) and *maximum surface speed* (ft/min) for a material must not be exceeded. The limiting PV for a material is based on the following test conditions: the contact material, surface speed, temperature and testing method. In actual service, the limiting PV is also influenced by other conditions, such as the presence or absence of a lubricating fluid. However, under similar conditions, PV limits provide useful guidelines for determining the relative wear resistance of different materials. The following table gives typical PV limits for some common bearing materials:

Bearing Material	Also Known As	PV Limit (PSI-ft/min)
PTFE	Virgin Teflon	1,800
Glass-Filled PTFE <sup>1</sup>	G. F. Teflon	12,000
Iron	Ferrum or Fe	30,000
Bronze	Copper-Tin Alloy	60,000
Bearing Grade PEEK <sup>1</sup>	B. G. PEEK	115,000
Graphite Grade Carbon <sup>1</sup>	Graphite or Carbon 60 <sup>2</sup>	120,000
Silicon Carbide <sup>1</sup>	SiC or Carborundum	400,000

1) Standard bearing materials offered by Liquiflo.

2) Graphite and Carbon 60 are allotropes (i.e., different naturally occurring forms) of the element Carbon.

The table shows, for example, that Silicon Carbide bearings are more effective than Carbon bearings at resisting contact wear; Carbon is more effective than Bronze or Iron; and Bearing Grade PEEK is more effective than Glass-filled PTFE.

## Z &amp; PV and the Relation to Wear Rates

For Liquiflo gear pumps,  $Z < 10$  indicates *thin film lubrication* (or boundary lubrication),  $10 \leq Z < 30$  indicates *mixed film lubrication* (or partial hydrodynamic lubrication), and  $Z \geq 30$  indicates *thick film lubrication* (or full hydrodynamic lubrication). With thin film lubrication, there is continuous contact between the bearing and shaft surface asperities at all speeds, which results in a very high or excessive wear rate. *In this situation of insufficient film lubrication, pump life will be dramatically reduced and therefore the application is not suitable for continuous duty.* With mixed film lubrication, intermittent contact occurs between the bearing and shaft surfaces at the operating speed, causing wear. In this case, the materials must be carefully chosen to provide adequate wear resistance. With thick film lubrication, there exists a substantial fluid film that supports the shaft and prevents contact with the bearings when the pump is operating at normal speed. However, during starting and stopping, pump speeds are lower and the fluid film is either nonexistent or too weak to fully support the shaft. During these conditions, bearing-shaft contact and wear can occur. This wear can be significantly reduced with the proper selection of wear resistant materials.

The theoretical wear rate of a material is directly proportional to the PV load, up to the PV limit. The proportionality constant is called the *wear factor* or *wear coefficient* of the material, which, like the PV limit, also depends on the test or service conditions. Below the PV limit, the wear rate is normal and increases gradually and linearly with the PV load. *Above the PV limit, the wear rate increases rapidly and nonlinearly with the PV load. This situation must be avoided to prevent an excessive wear rate and ensuing failure of the bearings.* As explained above, materials with higher PV limits are more effective at resisting contact wear than materials with lower PV limits. For each application, the materials must be chosen such that the PV limit, load capacity and maximum surface speed of the bearing material will not be exceeded. If all criteria are met, the materials can withstand the frictional forces resulting from periods of contact and the wear rate will be relatively low.

## Bearing &amp; Shaft Material Selection

Liquiflo application engineers utilize a computer program to evaluate the characteristic number (Z) and PV value for each application to determine what bearing and shaft materials can reliably handle the conditions of the application. A letter from **A** to **E** (Bearing Index) is assigned based on the values of Z and PV for the application, where **A** represents the lowest wear resistance required by the materials and **E** the highest (see table below). For mixed film lubrication, rigid bearings and hard-coated shafts are suggested (Bearing Index = **C** to **E**). For full hydrodynamic lubrication, the suggested materials can vary from lowest to highest in wear resistance (Bearing Index = **A** to **E**). In each case, the degree of wear resistance required is dependent on the PV value of the application. This system ensures that the bearings and shafts will have adequate wear resistance during periods of contact – which frequently occurs with mixed film lubrication or during acceleration and deceleration periods (when the fluid film is too weak to support the shaft) – thereby allowing the wear rates to be limited to acceptable levels.

Bearing Index	Suggested Bearing & Shaft Material Combination
<b>E</b>	Silicon Carbide Bearing vs. Tungsten Carbide-Coated Shaft
<b>D</b>	Any Rigid Bearing* vs. Tungsten Carbide-Coated Shaft
<b>C</b>	Any Rigid Bearing* vs. Ceramic-Coated Shaft
<b>B</b>	Any Rigid Bearing* vs. Uncoated Shaft
<b>A</b>	Any Bearing and Shaft Combination

\* Rigid bearings include B. G. PEEK, Carbon and SiC, but exclude G. F. Teflon and PTFE.

↑  
Increasing  
Wear  
Resistance

**Note:** The above system is based only on the tribological properties of the materials. The chemical and thermal properties of the materials must also be considered. Consult Liquiflo for specific applications.



## CENTRIFUGAL PUMP SELECTION FOR VISCOUS SERVICE

**Determination of Pump Performance When Handling Viscous Liquids**

The performance of centrifugal pumps is affected when handling viscous liquids. A marked increase in brake horsepower, a reduction in head, and some reduction in capacity occur with moderate and high viscosities.

**Graph 1** and **Graph 2** on the following pages provide a means of determining the performance of a conventional centrifugal pump handling a viscous liquid when its performance on water is known. **Graph 1** and **Graph 2** can also be used as an aid in selecting a pump for a given application. The values shown in **Graph 2** are averages of tests from conventional single stage pumps of 2" to 8" size, handling petroleum oils. The values shown in **Graph 1** were prepared from other tests on several smaller pumps (1" and below). Therefore, the correction curves are not exact for any particular pump.

**Limitations on use of Viscous Liquids Performance Correction Chart**

Use only for pumps of conventional hydraulic design, in the normal operating range, with open or closed impellers. Do not use for mixed flow or axial flow pumps, or for pumps of special hydraulic design for either viscous or non-uniform liquids. Use only where adequate NPSH is available in order to avoid cavitation.

Use only Newtonian (uniform liquids). Gels, slurries, paper stock and other non-uniform liquids may produce widely varying results depending on the particular characteristics of the liquids.

**Symbols and Definitions Used in Determination of Pump Performance When Handling Viscous Liquids**

$Q_{vis}$	= Viscous capacity in GPM – The capacity when pumping a viscous fluid.
$H_{vis}$	= Viscous head in feet – The head when pumping a viscous fluid.
$n_{vis}$	= Viscous efficiency in percent – The efficiency when pumping a viscous fluid.
$bhp_{vis}$	= Viscous brake horsepower – The horsepower required by the pump for the viscous conditions.
$Q_w$	= Water capacity in GPM – The capacity when pumping water.
$H_w$	= Water head in feet – The head when pumping water.
$n_w$	= Water efficiency in percent – The efficiency when pumping water.
SG	= Specific gravity.
$C_Q$	= Capacity correction factor.
$C_H$	= Head correction factor.
$C_n$	= Efficiency correction factor.
$Q_{NW}$	= Water capacity in which maximum efficiency is obtained.

The following equations are used for determining the viscous performance when the water performance of the pump is known:

$$\begin{aligned} Q_{vis} &= C_Q \times Q_w \\ H_{vis} &= C_H \times H_w \\ n_{vis} &= C_n \times n_w \\ bhp_{vis} &= \frac{Q_{vis} \times H_{vis} \times SG}{3960 \times n_{vis}} \end{aligned}$$

$C_Q$ ,  $C_H$  and  $C_n$  are determined from **Graph 1** and **Graph 2** which are based on water performance. **Graph 1** is to be used for small pumps having capacity at best efficiency point (BEP) of less than 100 GPM (water performance).

The following equations are used for approximating the water performance when the desired viscous capacity and head are given:

$$\begin{aligned} Q_w &= \frac{Q_{vis}}{C_Q} \\ H_w &= \frac{H_{vis}}{C_H} \end{aligned}$$

The values of  $C_Q$  and  $C_H$  are found from **Graph 1** and **Graph 2** using  $Q_{vis}$  and  $H_{vis}$

## CENTRIFUGAL PUMP SELECTION FOR VISCOUS SERVICE (continued)

## Selecting a Pump for a Given Head-Capacity-Viscosity Condition

Given the desired capacity and head of the viscous liquid to be pumped, and the viscosity and specific gravity at the pumping temperature, **Graph 1** or **Graph 2** can be used to find approximate equivalent capacity and head when pumping water.

Enter appropriate chart with the desired viscous capacity, ( $Q_{vis}$ ) and proceed upward to the desired viscous head ( $H_{vis}$ ) in feet of liquid. For multi-stage pumps use head per stage. Proceed horizontally (either left or right) to the fluid viscosity, and then go upward to the correction curves. Divide the viscous capacity ( $Q_{vis}$ ) by the capacity correction factor ( $C_Q$ ) to get the approximate equivalent to water capacity ( $Q_w$ ). Divide the viscous head ( $H_{vis}$ ) by the head correction factor ( $C_H$ ) from the curve marked "1.0 x  $Q_{NW}$ " to get the approximate equivalent water head ( $H_w$ ) at BEP. Using the new equivalent water head capacity point, select a pump in the usual manner. The viscous efficiency ( $n_{vis}$ ) and the viscous brake horsepower ( $bhp_{vis}$ ) may then be calculated.

## Example 1:

Select a pump to deliver **750 gpm** at **100 feet total head** of a liquid having a viscosity **1000 SSU** and a specific gravity of 0.90.

Flow =  $Q_{vis}$  = **750 gpm**      Viscosity = **1000 SSU**  
Head =  $H_{vis}$  = **100 feet**      Specific Gravity = **0.90**

Enter Graph 2 at 750 gpm, go up to 100 feet head, over to 1000 SSU and then up to the correction factors:

$$\begin{aligned} C_Q &= 0.95 \\ C_H &= 0.92 \text{ (for } 1.0 \times Q_{NW}) \\ C_n &= 0.64 \\ Q_w &= \frac{Q_{vis}}{C_Q} = \frac{750}{0.95} = 790 \text{ gpm} \\ H_w &= \frac{H_{vis}}{C_H} = \frac{100}{0.92} = 109 \text{ feet of head} \end{aligned}$$

Select a pump with a water capacity of **790 gpm** at **109 feet of head**. The selection should be at or close to the maximum efficiency point for water performance. If the pump selected has an efficiency on water of 81 percent at **790 gpm**, then the efficiency for the viscous liquid will be as follows:

$$n_{vis} = C_n \times n_w = 0.64 \times 81\% = 51.8\%$$

The brake horsepower for pumping the viscous liquid will be:

$$bhp_{vis} = \frac{Q_{vis} \times H_{vis} \times SG}{3960 \times n_{vis}} = \frac{750 \times 100 \times 0.90}{3960 \times 0.518} = 32.9 \text{ hp}$$

## Example 2:

Select a pump to deliver **53 gpm** at **50 feet total head** of a liquid having a viscosity **200 SSU** and a specific gravity of 0.95.

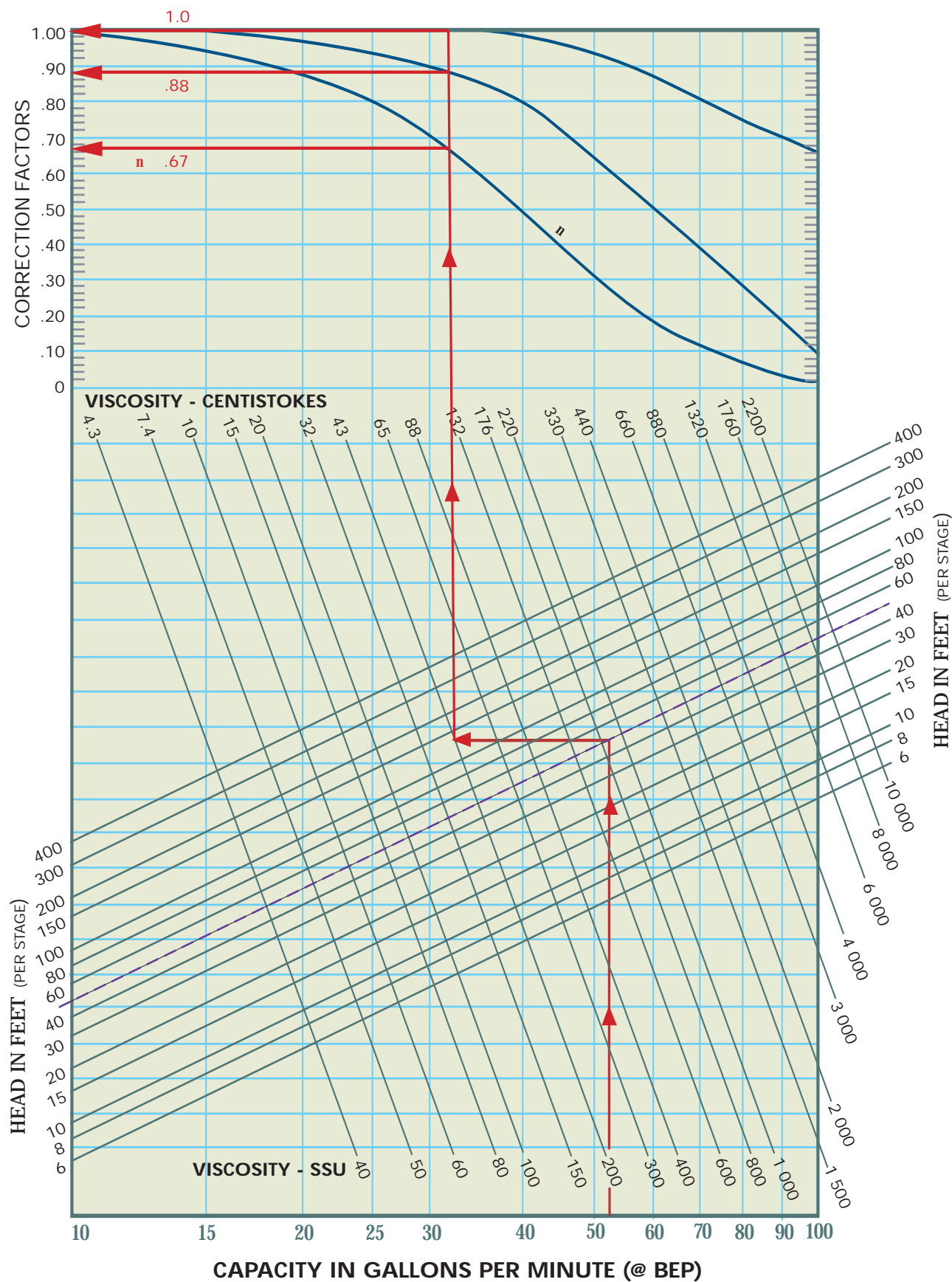
Flow =  $Q_{vis}$  = **53 gpm**      Viscosity = **200 SSU**  
Head =  $H_{vis}$  = **50 feet**      Specific Gravity = **0.95**

Enter Graph 1 at 53 gpm, go up to 50 feet head, over to 200 SSU and then up to the correction factors:

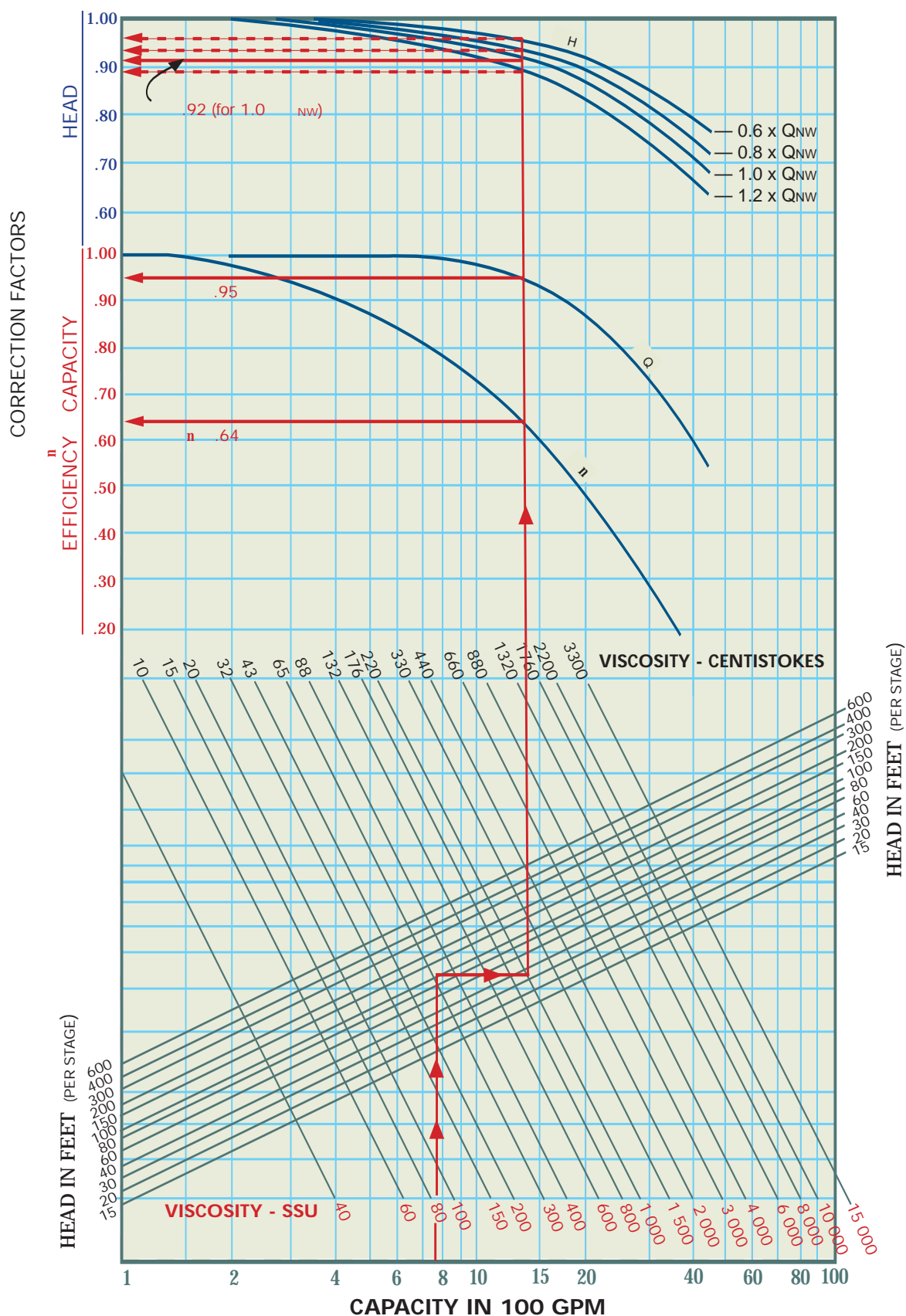
$$\begin{aligned} C_Q &= 0.88 \\ C_H &= 1.0 \\ C_n &= 0.67 \\ Q_w &= \frac{Q_{vis}}{C_Q} = \frac{53}{0.88} = 60 \text{ gpm} \\ H_w &= \frac{H_{vis}}{C_H} = \frac{50}{1.0} = 50 \text{ feet of head} \end{aligned}$$

Select a pump with a water capacity of **60 gpm** at **50 feet of head**.

**Graph1: PERFORMANCE CORRECTION FOR VISCOUS LIQUIDS**



Graph 2: PERFORMANCE CORRECTION FOR VISCOUS LIQUIDS



## CALCULATION OF SYSTEM NPSHA

### NPSHA

In order for pumps to function properly, there must be enough suction pressure available to force the fluid into the pump's inlet. In addition, the fluid must be kept in a liquid state or cavitation will occur. The following four parameters must be known in order to calculate the **net positive suction head available (NPSHA)** for a pumping system:

1.  $P_B$  = Absolute Barometric Pressure in Feet of Liquid (+), or  $p$  = Absolute Pressure on Surface of Liquid in Closed Supply Tank in Feet of Liquid (+)
2.  $L_H$  = Static Suction Head (+), or  $L_S$  = Suction Lift (-)
3.  $h_f$  = Friction Loss in Suction Line (-)
4.  $V_p$  = Vapor Pressure of Liquid (-)

The *NPSHA* is the algebraic sum of these four terms. Figures A thru D below, illustrate how to calculate the NPSHA for four common pumping situations.

### Requirement: $NPSHA > NPSHR$

The **net positive suction head required (NPSHR)** by a pump is determined by testing and is usually specified at a particular operating speed and based on pumping water. A requirement for all pumps is that the NPSHA must be greater than the NPSHR. Failure to meet this condition will result in *cavitation*. The symptoms of cavitation are decreased flow, increased vibrations and noise from the pump. Cavitation results when the fluid vaporizes because there is insufficient pressure available to keep the fluid in a liquid state. This condition must be avoided when running a pump because it will cause rapid wear and can damage the pump in a very short time.

**Example:** An application requires that 180°F water be pumped at 3 GPM with a gear pump at sea level. The supply tank is open to the atmosphere and the surface of the liquid in the tank will be no more than 6 ft below the pump's inlet. Assume that the friction losses in the suction line are equivalent to 2 ft of static head. The vapor pressure of water is about 380 mmHg at the pumping temperature.

#### a: What is the NPSHA for the pumping system?

**Answer:** Since the tank is aerated and is located at sea level, the Absolute Barometric Pressure ( $P_B$ ) is about 14.7 PSI or 34 ft. When the liquid level is near the bottom of the tank, the pump must be able to lift the water 6 ft, so the worst-case Suction Lift ( $L_S$ ) is 6 ft. The Friction Loss ( $h_f$ ) in the suction line is given as 2 ft. The Vapor Pressure ( $V_p$ ) of the hot water is equivalent to 17 ft. Therefore,

$$NPSHA = P_B - L_S - h_f - V_p = 34 \text{ ft} - 6 \text{ ft} - 2 \text{ ft} - 17 \text{ ft} = 34 \text{ ft} - 25 \text{ ft} = 9 \text{ ft}$$

#### b: What pumping situation is described in Part a?

**Answer:** Situation A (see figure below).

#### c: What must the gear pump's NPSHR be to successfully handle the application?

**Answer:** The NPSHA for a pumping system should be at least a couple of feet more than the NPSHR by the pump. This will provide a safety margin to account for any variation in the system parameters. Therefore, the NPSHR by the pump should be about 7 ft or less.

### CALCULATION of SYSTEM NPSHA for COMMON PUMPING SITUATIONS

<p><b>A. Suction Supply Open to Atmosphere with Suction Lift</b></p> <p><math>NPSHA = P_B - L_S - h_f - V_p</math></p>	<p><b>B. Closed Suction Supply with Suction Lift</b></p> <p><math>NPSHA = p - L_S - h_f - V_p</math></p>
<p><b>C. Suction Supply Open to Atmosphere with Suction Head</b></p> <p><math>NPSHA = P_B + L_H - h_f - V_p</math></p>	<p><b>D. Closed Suction Supply with Suction Head</b></p> <p><math>NPSHA = p + L_H - h_f - V_p</math></p>

## Application Data Sheet (A.D.S.)

**I. Fluid Data: Physical & Chemical Properties** (please specify units)

**FLUID** \_\_\_\_\_ % Concentration \_\_\_\_\_ Boiling Point \_\_\_\_\_ ☐ °F ☐ °C  
**PUMPING TEMPERATURE (PT)** \_\_\_\_\_ Min Temp. \_\_\_\_\_ Max Temp. \_\_\_\_\_ ☐ °F ☐ °C  
**VISCOSITY @ PT** \_\_\_\_\_ Min Visc. \_\_\_\_\_ Max Visc. \_\_\_\_\_ ☐ cP ☐ cSt ☐ SSU  
**SPECIFIC GRAVITY @ PT** \_\_\_\_\_ Vapor Pressure @ PT \_\_\_\_\_ ☐ PSI ☐ mmHg

Suspended Solids: ☐ Yes ☐ No % Solids Conc. \_\_\_\_\_ Particle Size \_\_\_\_\_ μm SG \_\_\_\_\_

☐ Toxic ☐ Noxious ☐ Flammable ☐ Explosive ☐ Corrosive  
☐ Abrasive ☐ Crystallizes ☐ Polymerizes ☐ Shear Sensitive ☐ Heat Sensitive  
☐ Other Properties \_\_\_\_\_

**II. Performance Data: Flow & Pressure** (please specify units)

**FLOW RATE:** Normal \_\_\_\_\_ Min \_\_\_\_\_ Max \_\_\_\_\_ ☐ GPM ☐ GPH ☐ LPM ☐ LPH ☐ mL/min ☐ m³/hr  
 Suction: ☐ Flooded ☐ Lift \_\_\_\_\_ ☐ ft ☐ m ☐ NPSHA \_\_\_\_\_ ☐ ft ☐ m  
**SUCTION PRESSURE** \_\_\_\_\_ ☐ PSIA ☐ bar (abs) **DISCHARGE PRESSURE** \_\_\_\_\_ ☐ PSIA ☐ bar (abs)  
**DIFFERENTIAL PRESSURE/HEAD** \_\_\_\_\_ ☐ PSI ☐ bar ☐ MPa ☐ kg/cm² ☐ ft ☐ m

**III. Motor Data:**

Type: ☐ AC MOTOR ☐ DC MOTOR ☐ AIR MOTOR  
 Power: \_\_\_\_\_ ☐ Hp ☐ kW Speed: \_\_\_\_\_ RPM  
 AC Power Supply Frequency \_\_\_\_\_ Hz ☐ 1-Phase ☐ 3-Phase Voltage: \_\_\_\_\_  
 Motor Enclosure: ☐ ODP ☐ TE ☐ XP-"T" Code \_\_\_\_\_ ☐ Other \_\_\_\_\_  
 NEMA Frame: \_\_\_\_\_ IEC Frame: \_\_\_\_\_ ☐ FootMount (B3) ☐ Flange Mount (B5) ☐ Face Mount (B14)

**IV. Location Data:**

Type: ☐ Indoor ☐ Outdoor  
 Ambient Temperature Range: Min \_\_\_\_\_ Max \_\_\_\_\_ ☐ °F ☐ °C  
☐ Sea Level ☐ Elevation \_\_\_\_\_ ☐ ft ☐ m  
 Hazardous Classification: ☐ Class I (Gases or Vapors) ..... Division: ☐ 1 ☐ 2 ..... Group: ☐ A ☐ B ☐ C ☐ D  
                                   ☐ Class II (Dust) ..... Division: ☐ 1 ☐ 2 ..... Group: ☐ E ☐ F ☐ G  
                                   ☐ Class III (Fibers) ..... Division: ☐ 1 ☐ 2

**V. Comments:**



## ENGINEERING FORMULAS

These basic formulas are used for the design, analysis and operation of gear and centrifugal pumps.

### Parameters:

FHP	=	Fluid Horsepower
BHP	=	Brake Horsepower
$\Delta P$	=	Differential Pressure
Q	=	Flow Rate
$Q_{min}$	=	Minimum Flow Rate
T	=	Torque
n	=	Speed
$n_s$	=	Specific Speed
H	=	Total Head
E	=	Efficiency
P	=	Power
$P_m$	=	Mechanical Power
D	=	Impeller Diameter
$\Delta T$	=	Temperature Rise
$C_p$	=	Specific Heat of Liquid
SG	=	Specific Gravity of Liquid
$A_i$	=	Pipe Inner Cross-sectional Area
$d_i$	=	Pipe Inner Diameter
D	=	Displacement
g	=	Gravitational Constant
$\delta$	=	Density of Liquid
$\delta_w$	=	Density of Water
v	=	Velocity of Liquid
p	=	Pressure, static
z	=	Elevation
s	=	Suction
d	=	Discharge
V	=	Voltage
I	=	Current
PF	=	Power Factor

Note: PF = Power Factor = Ratio of Real Power to Apparent Power =  $\cos \theta$  = Cosine of phase-shift angle ( $\theta$ ) between voltage and current sine waves.

### Units:

$^{\circ}F$	=	Degrees Fahrenheit
$^{\circ}C$	=	Degrees Celsius
PSI	=	Pounds per Square Inch
GPM	=	Gallons per Minute
RPM	=	Revolutions per Minute
in-lbs	=	Inch-Pounds
ft	=	Feet
Hp	=	Horsepower
kW	=	Kilowatts
Btu	=	British Thermal Unit
cSt	=	Centistokes
cP	=	Centipoises
$V_{dc}$	=	DC Volts
$V_{rms}$	=	AC Volts
$A_{dc}$	=	DC Amperes
$A_{rms}$	=	AC Amperes

### ① TEMPERATURE CONVERSIONS:

$$1a) \quad ^{\circ}F = (1.8 \times ^{\circ}C) + 32$$

$$1b) \quad ^{\circ}C = 0.556 \times (^{\circ}F - 32)$$

### ② FLUID HORSEPOWER (FHP):

$$2a) \quad FHP = \frac{\Delta P \text{ [PSI]} \times Q \text{ [GPM]}}{1714}$$

$$2b) \quad FHP = \frac{H \text{ [ft]} \times Q \text{ [GPM]} \times SG}{3960}$$

### ③ BRAKE HORSEPOWER (BHP):

$$3a) \quad BHP = \frac{FHP}{E} = \frac{\Delta P \text{ [PSI]} \times Q \text{ [GPM]}}{1714 \times E}$$

$$3b) \quad BHP = \frac{FHP}{E} = \frac{H \text{ [ft]} \times Q \text{ [GPM]} \times SG}{3960 \times E}$$

### ④ PUMP EFFICIENCY (E):

$$4a) \quad E = \frac{\text{Hydraulic Output Power}}{\text{Total Input Power}} = \frac{FHP}{BHP}$$

### ⑤ MECHANICAL POWER ( $P_m$ ):

$$5a) \quad P_m \text{ [Hp]} = \frac{T \text{ [in-lbs]} \times n \text{ [RPM]}}{63,025}$$

$$5b) \quad P_m \text{ [kW]} = \frac{T \text{ [in-lbs]} \times n \text{ [RPM]}}{84,484}$$

### ⑥ SPECIFIC SPEED ( $n_s$ ) for Centrifugal Pumps:

$$6a) \quad n_s = \frac{n \text{ [RPM]} \times \sqrt{Q \text{ [GPM]}}}{(H \text{ [ft]})^{3/4}}$$

### ⑦ AFFINITY LAWS for Centrifugal Pumps:

$$7a) \quad \frac{Q_1}{Q_2} = \frac{n_1 D_1}{n_2 D_2}$$

$$7b) \quad \frac{H_1}{H_2} = \frac{n_1^2 D_1^2}{n_2^2 D_2^2} \quad (\text{NPSHR formula has the same format as formula 7b.})$$

$$7c) \quad \frac{P_1}{P_2} = \frac{n_1^3 D_1^3}{n_2^3 D_2^3}$$

### ⑧ Centrifugal Pump Operating at Low Capacity:

$$8a) \quad \text{Temperature Rise: } \Delta T \text{ [}^{\circ}F\text{]} = \frac{H \text{ [ft]}}{778 C_p \text{ [Btu/lb-}^{\circ}F\text{]}} \times \frac{1}{E-1}$$

$$8b) \quad \text{Minimum Flow: } Q_{min} \text{ [GPM]} = \frac{3960 \times BHP}{SG (778 C_p \text{ [Btu/lb-}^{\circ}F\text{]} \Delta T \text{ [}^{\circ}F\text{]} + H \text{ [ft]})}$$

## ENGINEERING FORMULAS (continued)

These basic formulas are used for the design, analysis and operation of gear and centrifugal pumps.

### Parameters:

FHP	=	Fluid Horsepower
BHP	=	Brake Horsepower
$\Delta P$	=	Differential Pressure
Q	=	Flow Rate
$Q_{min}$	=	Minimum Flow Rate
T	=	Torque
n	=	Speed
$n_s$	=	Specific Speed
H	=	Total Head
E	=	Efficiency
P	=	Power
$P_m$	=	Mechanical Power
D	=	Impeller Diameter
$\Delta T$	=	Temperature Rise
$C_p$	=	Specific Heat of Liquid
SG	=	Specific Gravity of Liquid
$A_i$	=	Pipe Inner Cross-sectional Area
$d_i$	=	Pipe Inner Diameter
D	=	Displacement
g	=	Gravitational Constant
$\delta$	=	Density of Liquid
$\delta_w$	=	Density of Water
v	=	Velocity of Liquid
p	=	Pressure, static
z	=	Elevation
s	=	Suction
d	=	Discharge
V	=	Voltage
I	=	Current
PF	=	Power Factor

Note: PF = Power Factor = Ratio of Real Power to Apparent Power =  $\cos \theta$  = Cosine of phase-shift angle ( $\theta$ ) between voltage and current sine waves.

### Units:

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GPM	=	Gallons per Minute
RPM	=	Revolutions per Minute
in-lbs	=	Inch-Pounds
ft	=	Feet
Hp	=	Horsepower
kW	=	Kilowatts
Btu	=	British Thermal Unit
cSt	=	Centistokes
cP	=	Centipoises
$V_{dc}$	=	DC Volts
$V_{rms}$	=	AC Volts
$A_{dc}$	=	DC Amperes
$A_{rms}$	=	AC Amperes

### 9 Flow Rate (Q) in terms of Fluid Velocity (v) and Pipe Dimensions

$$9a) Q = vA_i$$

$$9b) Q = \pi/4 \times v d_i^2 = 0.785 v d_i^2$$

### 10 Displacement Rate (D) of Rotary Pumps

$$10a) D [\text{gal/rev}] = \frac{Q [\text{GPM}]}{n [\text{RPM}]}$$

### 11 Total Head (H)

$$11a) \text{ Total Dynamic Head: } H_{dynamic} = \frac{(v_d^2 - v_s^2)}{2g}$$

$$11b) \text{ Total Static Head: } H_{static} = \frac{(p_d - p_s)}{\delta} + (z_d - z_s)$$

$$11c) H = H_{dynamic} + H_{static} = \frac{(v_d^2 - v_s^2)}{2g} + \frac{(p_d - p_s)}{\delta} + (z_d - z_s)$$

### 12 Relation Between Static Head and Pressure

$$12a) H_{[ft]} = \frac{p_{[PSI]} \times 2.31}{SG}$$

$$12b) p_{[PSI]} = \frac{H_{[ft]} \times SG}{2.31}$$

### 13 Relation between Kinematic and Absolute Viscosity

$$13a) \text{ Kinematic Viscosity } [cSt] = \frac{\text{Absolute Viscosity } [cP]}{SG}$$

### 14 Relation between Density and Specific Gravity

$$14a) SG = \frac{\delta}{\delta_w}$$

$$14b) \delta = \delta_w \times SG$$

### 15 Efficiency (E) of DC Motors

$$15a) E = \frac{\text{Mechanical Output Power}}{\text{Electrical Input Power}} = \frac{P_m [\text{Hp}] \times 746}{V [V_{dc}] \times I [A_{dc}]}$$

### 16 Efficiency (E) of Single-Phase AC Motors

$$16a) E = \frac{\text{Mechanical Output Power}}{\text{Electrical Input Power}} = \frac{P_m [\text{Hp}] \times 746}{V [V_{rms}] \times I [A_{rms}] \times PF}$$

### 17 Efficiency (E) of Three-Phase AC Motors

$$17a) E = \frac{\text{Mechanical Output Power}}{\text{Electrical Input Power}} = \frac{P_m [\text{Hp}] \times 746}{V [V_{rms}] \times I [A_{rms}] \times PF \times \sqrt{3}}$$

## VISCOSITY UNITS CONVERSION TABLE

[illegible]

## VISCOSITY UNITS CONVERSION TABLE (continued)

MA MI AEL ( )	AR NER ( ) LIT O RA I 25 (77 F)	AR NER OLT ( ) 25 (77 F)	MO ILEMETER 100 20	ARLIN 20	ARLIN 15	ARLIN 10	ARLIN 7	AR E	RAT LAM ERT F	FOR 4	FOR 3	WE TIN O E	EN LER E IFI ( E REE )	EN LER E IFI ( E ON )	RE WOO ( E ON )	RE WOO A MIRALT ( E ON )	A OLT ( E ON ) ( . F )	A OLT NI ER AL ( E ON ) ( . . )	INEMAT ( ENTI TO E )
750	1,000																		
37.5	50																		
7.5	10																		
15	20																		
100	30																		
300	40																		
15	30																		
25	10																		
100	15																		
200	20																		
300	25																		
400	30																		
500	35																		
600	40																		
700	45																		
800	50																		
900	55																		
1,000	60																		
1,500	75																		
2,000	100																		
2,500	125																		
3,000	150																		
4,000	200																		
5,000	250																		
6,000	300																		
7,000	350																		
8,000	400																		
10,000	500																		
20,000	750																		
30,000	1,000																		
40,000	1,200																		
50,000	1,500																		
60,000	1,700																		
80,000	2,000																		
100,000	2,500																		
200,000	5,000																		
300,000	7,500																		
400,000	10,000																		
500,000	12,500																		
600,000	15,000																		
700,000	17,500																		
800,000	20,000																		
900,000	22,500																		
1,000,000	25,000																		

**VISCOSITY CONVERSION TABLES**

are are ta ( ) are o are t  
ea of er. Te a e o are ta of te  
ot er ae t te o ar o t  
o te ot er ae te ef rat of te  
eto .

(A) et to e a e a e to et o e  
t ef rat .  
( ) et to e a e o erte to to e  
100.  
( ) et to e a e o erte to o e  
100.

## PHYSICS UNITS CONVERSION TABLE

**LENGTH**

1 meter (m) = 1.0936 yards = 3.2808 feet = 39.370 inches  
 1 foot (ft) = 12 inches = 0.3048 meters = 30.48 centimeters  
 1 inch (in) = 1000 mils = 2.54 centimeters (exactly) = 25.4 millimeters  
 1 kilometer (km) = 1000 meters = 3280.8 feet = 0.62137 mile  
 1 mile (mi) = 5280 feet = 1609.3 meters = 1.6093 kilometers  
 1 micron (um) =  $10^{-6}$  meter =  $10^{-3}$  millimeter = 0.03937 mils  
 1 mil =  $10^{-3}$  inches = 25.4 microns = 0.0254 millimeter

**MASS**

1 kilogram (kg) = 1000 grams = 2.2046 pounds  
 1 pound (lb) = 16 ounces = 453.59 grams = 0.45359 kilograms  
 1 ton = 2000 pounds = 907.185 kilograms  
 1 metric ton = 1000 kilograms = 2204.6 pounds

**VOLUME**

1 liter (L) =  $10^{-3}$  m<sup>3</sup> = 1 dm<sup>3</sup> = 1000 mL = 1.0567 quarts = 0.26417 gallons  
 1 gallon (gal) = 128 fluid ounces = 4 quarts = 3.7854 liters  
 1 quart (qt) = 32 fluid ounces = 0.94635 liter

**PRESSURE**

1 pascal (Pa) = 1 N/m<sup>2</sup> =  $10^{-5}$  bar  
 1 atmosphere (atm) = 101.325 kPa = 760 torr (mmHg) = 14.7 PSI  
 1 bar =  $10^5$  Pa = 100 kPa = 14.5 PSI  
 1 kg/m<sup>2</sup> = 0.980 bar = 14.2 PSI

**ENERGY**

1 joule (J) = 1 N-m =  $10^7$  ergs = 0.23901 calorie =  $9.4781 \times 10^{-4}$  Btu  
 1 calorie (cal) = 4.184 joules =  $3.965 \times 10^{-3}$  Btu  
 1 British thermal unit (Btu) = 1055.1 joules = 252.2 calories  
 1 kW-hr =  $3.600 \times 10^6$  joules

**POWER**

1 horsepower (Hp) = 746 watts = 0.746 kilowatts  
 1 kilowatt (kW) = 1000 watts = 1.34 Hp

**FORCE**

1 pound (lb) = 4.45 newtons =  $4.45 \times 10^5$  dyn  
 1 newton (N) = 1 kg-m-s<sup>-2</sup> = 0.2247 pounds =  $10^5$  dyn

**TORQUE**

1 ft-lb = 12 in-lbs = 1.356 N-m = 135.6 N-cm  
 1 N-m = 100 N-cm = 0.7373 ft-lb = 8.847 in-lbs

**SPEED, Linear**

1 mi/hr = 1.6093 km/hr = 1.4667 ft/s = 0.4470 m/s  
 1 km/hr = 0.6214 mi/hr = 0.9113 ft/s = 0.2778 m/s  
 1 ft/s = 0.3048 m/s = 0.6818 mi/hr = 1.0973 km/hr  
 1 m/s = 3.2808 ft/s = 2.2369 mi/hr = 3.6 km/hr

**SPEED, Rotational**

1 hertz (Hz) = 1 rev/s = 60 RPM = 60 rev/min  
 1 RPM = 1 rev/min = 1/60 Hz = 1/60 rev/s

**TEMPERATURE**

Absolute Zero = 0 K = -273.15 °C = -459.67 °F  
 K = °C + 273.15  
 K = °F + 459.67  
 °C = (5/9)(°F - 32)  
 °F = (9/5)(°C) + 32

**VISCOSITY**

1 poise (P) = 1 dyn-s-cm<sup>-2</sup> = 0.1 N-s-m<sup>-2</sup> = 0.1 Pa-s  
 1 stoke (St) = 1 cm<sup>2</sup>/s = 100 mm<sup>2</sup>/s  
 1 centipoise (cP) =  $10^{-2}$  P = 1 mPa-s  
 1 centistoke (cSt) =  $10^{-2}$  St = 1 mm<sup>2</sup>/s

**DENSITY**

1 g/mL = 1 kg/L = 1 kg/dm<sup>3</sup> = 1000 kg/m<sup>3</sup> = 62.43 lb/ft<sup>3</sup>  
 1 lb/gal = 0.1198 kg/L  
 1 kg/L = 8.345 lb/gal

## FLOW &amp; PRESSURE UNITS CONVERSION CHARTS

## FLOW UNITS CONVERSION CHART

To convert from a unit in this column...	To any of these units, perform operation at Row-Column Intersection (• = MULTIPLY BY, ÷ = DIVIDE BY)							
	GPM	GPH	LPM	LPH	ft <sup>3</sup> /hr	m <sup>3</sup> /hr	lb/hr (SG = x)	kg/hr (SG = x)
GPM	L	60 •	3.785 •	227.1 •	8.021 •	.2271 •	500.7 x	227.1 x
GPH	60 ÷	I	.06309 •	3.785 •	.1337 •	.003785 •	8.345 x	3.785 x
LPM	3.785 ÷	.06309 ÷	Q	60 •	2.119 •	.06 •	132.3 x	60 x
LPH	227.1 ÷	3.785 ÷	60 ÷	U	.03531 •	.001 •	2.205 x	x
ft <sup>3</sup> /hr	8.021 ÷	.1337 ÷	2.119 ÷	.03531 ÷	I	.02832 •	62.43 x	28.32 x
m <sup>3</sup> /hr	.2271 ÷	.003785 ÷	.06 ÷	.001 ÷	.02832 ÷	F	2205 x	1000 x
lb/hr (SG = x)	500.7 x	8.345 x	132.3 x	2.205 x	62.43 x	2205 x	L	.4536 •
kg/hr (SG = x)	227.1 x	3.785 x	60 x	x	28.32 x	1000 x	.4536 ÷	O

**Example 1:** Convert 100 ft<sup>3</sup>/hr to flow in GPM.

**Solution:** 100 ft<sup>3</sup>/hr ÷ 8.021 = 12.5 GPM

**Example 2:** Convert 30 GPH to flow in kg/hr (SG = 1.5).

**Solution:** 30 GPH • 3.785 (1.5) = 30 GPH x 5.678 = 170 kg/hr

## PRESSURE UNITS CONVERSION CHART

To convert from a unit in this column...	To any of these units, perform operation at Row-Column Intersection (• = MULTIPLY BY, ÷ = DIVIDE BY)							
	atm	bar	kPa	PSI	kg/cm <sup>2</sup>	in Hg	m (SG = x)	ft (SG = x)
atm	G	1.01325 •	101.325 •	14.70 •	1.034 •	29.92 •	10.35/x •	33.96/x •
bar	1.01325 ÷	E	100 •	14.51 •	1.020 •	29.53 •	10.21/x •	33.51/x •
kPa	101.325 ÷	100 ÷	A	.1451 •	.0102 •	.2953 •	.1021/x •	.3351/x •
PSI	14.70 ÷	14.51 ÷	.1451 ÷	R	.0703 •	2.035 •	.704/x •	2.31/x •
kg/cm <sup>2</sup>	1.034 ÷	1.020 ÷	.0102 ÷	.0703 ÷	P	28.95 •	10.01/x •	32.85/x •
in Hg	29.92 ÷	29.53 ÷	.2953 ÷	2.035 ÷	28.95 ÷	U	.3460/x •	1.135/x •
m (SG = x)	10.35/x	10.21/x	.1021/x	.704/x	10.01/x	.3460/x	M	3.281 •
ft (SG = x)	33.96/x	33.51/x	.3351/x	2.31/x	32.85/x	1.135/x	3.281 ÷	P

**Example 3:** Convert 700 kPa to pressure in PSI.

**Solution:** 700 kPa • .1451 = 102 PSI

**Example 4:** Convert 50 m to pressure in kg/cm<sup>2</sup> (SG = 1.8)

**Solution:** 50 m ÷ 10.01/ (1.8) = 50 m ÷ 5.561 = 9.0 kg/cm<sup>2</sup>



## GREEK ALPHABET &amp; POWER-OF-TEN PREFIXES

## GREEK ALPHABET

NAME	LOWERCASE	UPPERCASE	NAME	LOWERCASE	UPPERCASE
Alpha	$\alpha$	A	Nu	$\nu$	N
Beta	$\beta$	B	Xi	$\xi$	$\Xi$
Gamma	$\gamma$	$\Gamma$	Omicron	$\omicron$	O
Delta	$\delta$	$\Delta$	Pi	$\pi$	$\Pi$
Epsilon	$\epsilon$	E	Rho	$\rho$	P
Zeta	$\zeta$	Z	Sigma	$\sigma$	$\Sigma$
Eta	$\eta$	H	Tau	$\tau$	T
Theta	$\theta$	$\Theta$	Upsilon	$\upsilon$	Y
Iota	$\iota$	I	Phi	$\phi$	$\Phi$
Kappa	$\kappa$	K	Chi	$\chi$	X
Lambda	$\lambda$	$\Lambda$	Psi	$\psi$	$\Psi$
Mu	$\mu$	M	Omega	$\omega$	$\Omega$

## POWER-OF-TEN PREFIXES

NAME	SYMBOL	VALUE	UNIT EXAMPLE		
atto	a	$10^{-18}$	as	attosecond	$10^{-18}$ second
femto	f	$10^{-15}$	fg	femtogram	$10^{-15}$ gram
pico	p	$10^{-12}$	pF	picofarad	$10^{-12}$ farad
nano	n	$10^{-9}$	nH	nanohenry	$10^{-9}$ henry
micro	$\mu$	$10^{-6}$	$\mu$ V	microvolt	$10^{-6}$ volt
milli	m	$10^{-3}$	mA	milliampere	$10^{-3}$ ampere
centi	c	$10^{-2}$	cSt	centistoke	$10^{-2}$ stoke
deci	d	$10^{-1}$	dB	decibel	$10^{-1}$ bel
deca	D	$10^1$	DL	decaliter	10 liters
hecto	H	$10^2$	Hm	hectometer	$10^2$ meters
kilo	k	$10^3$	kG	kilogauss	$10^3$ gauss
mega	M	$10^6$	MPa	megapascal	$10^6$ pascals
giga	G	$10^9$	GHz	gigahertz	$10^9$ hertz
tera	T	$10^{12}$	T $\Omega$	terohm	$10^{12}$ ohms
peta	P	$10^{15}$	PJ	petajoule	$10^{15}$ joules
exa	E	$10^{18}$	EW	exawatt	$10^{18}$ watts



