Roper Pump
Progressing Cavity Pump
Technical Manual
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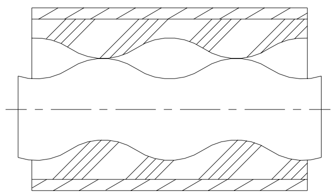
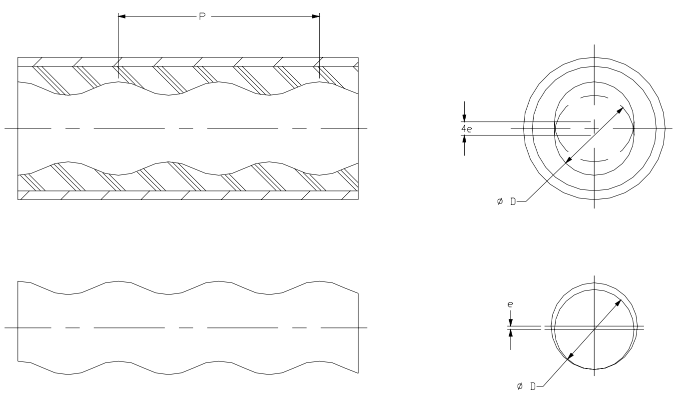
PROGRESSING CAVITY PUMP

The Progressing Cavity Pump is a helical gear pump consisting of an internal gear with a double thread (stator) and an external gear with a single thread (rotor). The meshing of the two gears forms a cavity, which progresses along the axis of the assembly as the rotor is rotated.



The cross section of the stator is two semi-circles of diameter **D** separated by a rectangle with side's **4e** and **D**. The cross section of the rotor is a circle of Diameter **D** that is offset from the centerline by the eccentricity **e**. The lead or pitch of thread of the stator is **P** and twice the lead or pitch of the rotor.



The dimensions of the cavity formed when the rotor and stator are meshed together are equal to the void of the cross section (π **D²/4 + 4eD**) – (π **D²/4**) or **4ed**. This cross sectional void times the stator lead determines the cavity (**4eDP**) which is displaced upon each revolution of the rotor and can be expressed in GPM per 100 rpm. The capacity chart is listed in the data section lists the values for the common elements.

PUMP PERFORMANCE

Being positive displacement, a certain volume of fluid is discharged with each revolution of the rotor. Unlike centrifugal pumps the pump does not develop pressure or a head but will attempt to delivery the same volume of fluid regardless of the pressure (resistance) that must be overcome in the discharge line. With a fluid such as water (1 cp viscosity) and with 0 discharge pressure, the displacement of the pump is only dependent on the revolutions per minute.

As the pressure increases a small amount of the fluid displaced slips back through the elements to the suction side. The amount of slip or leakage is greater as the amount of slip at high pressures more cavities are added in series by lengthening the rotor and stator. This is called staging.

The performance curves are for water and show the decreased delivery of the pump at different pressures and speeds. For thicker more viscous fluids the slip is significantly reduced and can be approximated by dividing the slip on water by the slip index found in the data section.

> **For Example: The slip on water for a 72205 pump at 100 psi is 5.5 gpm. For a 1000 cp fluid the slip would be 5.5/S.I. or 5.5/3.98 = 1.4 gpm.**

POWER REQUIREMENTS

The stator is usually made from an elastomeric material allowing the pump elements to have a compression fit and also offering a good abrasion resistant surface for handling particles in suspension. This compression fit, however, does cause a resistance to turning (torque) which is dependent on the element size and is shown on the performance curve as torque at 0 psi. The initial torque can be expressed as in.-lbs/stage of element and is listed on the element chart in the data section. There is also an initial starting torque, which must be overcome. This value is roughly 4 times the initial torque. The work done by a pump is the rate of delivery (displacement) against a certain pressure. In the case of the progressing cavity pump this is the volume of the cavity times the working pressure and is a constant for each size element. These values are also shown in the data section for each element size.

The progressing cavity pump is very rarely applied on a fluid with a viscosity of 1 and there is an added torque, which is dependent on the viscosity of the fluid and the size of the pump. The curves-0511 gives the relationship between viscosity and torque per stage for each element. Table B on the performance gives these values for certain viscosities.

On slurries the added torque required is dependent upon the particle size and concentration of solids. Table C on the performance curves is a guide to the added torque for slurries.

The basis for determining pumps power requirements in the technical manual and in the performance curves is to first determine the torque and then, knowing the speed, calculate the required horsepower. This is a method that may be unfamiliar to some since most positive displacement pump performance curves already include hp lines. Since the progressing cavity pump is considered a constant torque pump, using torque initially is a simplified and more accurate method of determining the power requirements and will allow better selection of drive components especially hydraulic, variable frequency, and SCR type drives.
Torque is a measure of force and length and has the units in in-lbs, ft-lbs, or kg-m. HP is a measure of work and is torque at a certain rpm. To determine the hp when knowing the torque the following formula is used.

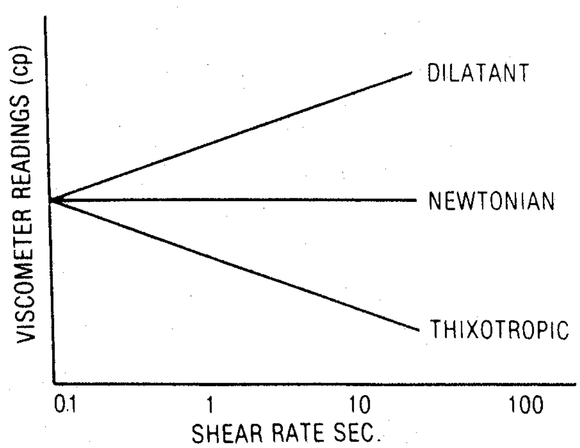
$$\textbf{HP} = \frac{\textbf{Torque (in. lbs.) x Speed (RPM)}}{\textbf{63025}}$$

Fluid Velocity and Shear Rate

With rotation of the rotor, fluid in the cavity moves in a spiral path along the centerline of the pump. The velocity of the fluid will be dependent on the speed of rotation as well as the distance from the centerline and the rate of shear of the fluid and the maximum particle size can be determined from D, e and P. The values of velocity and shear are listed on the element chart in the data section.

Fluid Viscosity

The viscosity of a fluid is that property of the fluid, which resists flow and is a ratio of the shearing stress to the rate of shear. For fluids other than oil the most common unit of measure for absolute viscosity is centipoises. Sometimes the fluid viscosity is expressed in centistokes, which is the kinematics viscosity or the absolute viscosity divided by the specific gravity. When a fluid's viscosity is constant as the rate of shear is increased, it is said to be a Newtonian fluid. Most fluids that are handled by the progressing cavity pump do not obey this law and are said to be non-Newtonian. With a non-Newtonian fluid the viscosity of the fluid changes as the rate of shear changes. Some fluids will show a decrease in viscosity (thixotropic) as the rate of shear increases. The following curves plot the viscosity in centipoises against the shear rate in inverse sec. on log-log coordinates.



Examples of thixotropic fluids are:
Adhesives, fruit juice concentrates, glues, animal oils, asphalts, lacquers, bentonite, lard, latex, cellulose compounds, waxes, syrups, fish oils, molasses, paints, soaps, paper size, starch, plastics, tar, rayon, printing inks, vanishes, resins, vegetable oil and shortenings.

An estimation of the “apparent viscosity” of the fluid can be made knowing several readings of the viscosity at known shear rates. The torque or power requirement for the pump selected can then be predicted at various speeds or shear rates.

Dilatant fluids are rather rare and are mostly high concentrated slurries. A dilatant fluid increases in viscosity as the shear rate is increased. Again, the power requirements can be determined knowing the apparent viscosity.

There are certain fluids or materials, which cannot be classified in the above categories and can be handled very nicely with the progressing cavity pump. These materials such as filter cake, dewatered slurries or sludge’s, paper stock are semi-dry and will not readily flow into the normal suction opening of the pump nor is it possible to obtain a viscosity measurement indicative of the thickness of the material. These applications are best handled with a hopper feed pump where the suction housing is replaced with flanged hopper and auger is attached to the connecting rod to assist movement of the material into the pump elements. Pump speed should be limited to 300 rpm and power requirements should be calculated as if the material were 10,000 cp. viscosity.

VOLUMETRIC EFFICIENCY

In the handling of viscous fluids it is important to know the effects of viscosity on the volumetric efficiency of the pump. Due to the inability of the fluid to flow into the open cavity of the pump elements there will be a reduction in the displacement. Curve S-0510 shows the relationship of volumetric efficiency at various viscosities and pump speeds. From this curve the pump speed required can be determined to deliver the desired flow rate at 0 psi. Slip is subtracted from this value to arrive at the flow rate under pressure.

Example: A 71244 pump at 300 rpm on a 2500 cp fluid will deliver 80% of the theoretical displacement or 106 gpm (132x.8) At 50 psi the slip for water is 20 gpm and for a 2500 cp fluid the slip would be 20/S.I. or 20/4.78 = 4.2 gpm. The flow rate at 50 psi would then be 106-4.2 or 102 gpm. Note the effects of volumetric efficiency are far greater than that of slip.

ABRASION

The progressing cavity pump is one of the best pumps for handling abrasive slurries, however there are some considerations in pump size that need to be made for maximum performance. It is necessary to minimize the slip and internal velocities to achieve good results. The chart in the data section and Table A on the performance curves is a guide of maximum pump speed and maximum pressure per stage.

Determining the degree of abrasion is mostly judgmental, however the make up of the particles will offer some clues as to how it is to be classified.

A deeper look into what causes abrasion may be helpful in determining its classification. The components of abrasivity are:

Particle

Size-wear increases with particle size
Hardness-wear increases rapidly with particle hardness when it exceeds the rotor surface hardness.
Concentration-the higher the concentration the more rapid wear.
Density-heavier particles will not pass thru the pump easily.

Relating the material hardness to some common materials on a 1 to 15 scale the following list can be used as a guide.

1. Talc slurry
2. Sodium sulfate
3. Drilling Mud
4. Kaolin Clay
5. Lime slurry
6. Toothpaste, potters glaze
7. Gypsum
8. Fly ash
9. Fine and slurry
10. Grout, plaster
11. Titanium dioxide
12. Ceramic slurry
13. Lapping compound
14. Emory dust slurry
15. Carborundum slurry

Carrier Liquid

Corrosivity-Surfaces attacked by corrosion will set up a corrosion erosion effect.
Viscosity-A high viscosity will tend to keep particles in suspension and not be as abrasive.

Velocity

A low fluid velocity or pump speed will minimize abrasive effects. For a heavy abrasive fluid it is recommended to keep the particle velocity between 3-5 ft/sec. A medium abrasive fluid should be limited to 10-15 ft/sec. These velocity limits are listed as rpm limits for the various size pumps in the data section and also in Table A on the individual pump curves.

NET POSITIVE SUCTION HEAD

The Net Positive Suction Head (NPSH) calculations are routinely used in centrifugal or high velocity pump applications. In positive displacement pump applications where the pump velocities are usually low the calculation of NPSH has little significance, however, there are applications where this calculation becomes very important. These are suction lift, vacuum pot applications and an application where the fluid vapors pressure is low.

The Net Positive Suction Head Available (NPSHA) is the head available at the inlet of the pump and is the sum of the atmospheric pressure available, the fluid vapor pressure, the lift and suction line losses. The Net Positive Suction Head Required (NPSHR) is a function of the pump designs and pump speed. These values are shown on curve S-0508 and also on the individual performance curves and are expressed in Ft. of fluid. The following examples illustrate NPSH calculations.

Suction Lift

When the suction head is a negative value i.e., there is a suction lift, the amount of lift, and the suction line losses, in addition to the fluid vapor pressure have to be subtracted from the atmospheric pressure to determine the NPSHA. This should always be greater than the NPSHR.

Example: A 71205 running @ 900 RPM is required to lift 70 Degree F. water 10 ft. vertical thru a 3 in. line. The vapor pressure of the fluid is .3631 psia (.84 ft) and the suction line losses are .01 ft. of fluid.

Atm. Pres.		**33.9 ft**
Lift		**-.10 ft**
Losses	**-.01 ft**	
Vap. Pres.		**-.84 ft**
Total NPSHA		**23.05 ft**
Total NPSHR	**6.90 ft**	

There will be 16.15 ft of fluid over the required amount and will be acceptable.

Low Vapor Pressure Fluids

Another application where the NPSH becomes important is when the fluid’s vapor pressure is low. In the previous example 70 degree F. water was used which has a vapor pressure of .363 psia. If the temperature of the water was 190 degree F. the vapor pressure would be 9.34 psia or 21.6 ft. which would exceed the 16.15 ft. difference between

HPSHA and NPSHR and the fluid would vaporize or boil. To overcome this situation the amount of lift would have to be shortened. The vapor pressure of water can usually be found in most hydraulic books and can be estimated for other fluids if the boiling point is known i.e., a fluid will boil at atmospheric conditions when its vapor pressure reaches 14.7 psi. The calculations would now be as follows.

Atm. Pres	**33.9 ft**
Lift	**-.10 ft**
Line Loss	**-.01 ft**
Vapor Pres	**–21.6 ft**
NPSHA	**2.29 ft**
NPSHR	**6.9 ft**

Vacuum Pot Installations

In a vacuum pot application the fluid is in a vessel that is under a high or partial vacuum. This will affect the NPSHA to the pump.

Example: A 71205 pump running at 900 rpm is pumping water out of a vessel that is under 20 inches of mercury vacuum. There is 10 ft of 3 in. horizontal line connecting the suction.

Atmos Pres.	**11.20 ft**
Line Loss	**-.01 ft**
Lift	**0**
Vapor Pres.	**-.84 ft**
NPSHA	**10.35 ft**
NPSHR	**6.90 ft**

On vacuum pot and suction lift applications it is necessary to fill or partially fill the suction lines with fluid to provide a lubricant for the elements during lift or until the pumping fluid reaches the elements. Since there are more sealing points on the suction side (packing, drive pins etc.) better operation can be achieved if the pump is operated in reverse i.e., the normal discharge port is used as the suction port. This would put the packing under pressure and caution should also be used not to overpressure the suction side.

Temperature Effects and Limit

The fluid temperature will affect the pump performance in two different ways. First, since the stator is an elastomeric material, the thermal expansion is roughly 10 time greater than that of steel. This will cause a tighter fit for the elements and higher starting and running torques. When the temperature reaches a certain limit it is then advisable to use an undersize rotor, which compensates for the difference in size. The following guidelines should be used.

Multiplier for starting and initial torque

Standard size rotor

Temp. Deg. F.	**70**	**100**	**125**	**150**	**175**
Multiplier	**1.0**	**1.1**	**1.3**	**1.6**	**1.8**

Undersize rotor

Temp. Deg. F.	**175**	**200**	**230**	**250**	**270**
Multiplier	**1.1**	**1.3**	**1.6**	**1.8**	**2.0**

Double undersize rotor

Temp. Deg. F.	**230**	**250**	**280**	**300**	**350**
Multiplier	**1.0**	**1.1**	**1.3**	**1.6**	**1.8**

Second, the life of the elastomer is greatly affected by heat. The following limits are for elastomers which are being worked such as in a stator and will differ from other published information on elastomers used in a static state such as O rings and gaskets. Therefore, when a stator is being applied at less than its maximum rating (75-psi/stage) this limit can be exceeded slightly, but cannot exceed the static limit rating.

Temperature Rating (Deg. F.)

Material	**Stator Rating**	**Max Rating**
Buna N	**180**	**250**
Natural	**185**	**225**
EPDM	**300**	**350**
Viton	**300**	**400**
Nitrile AR	**200**	**250**

MECHANICAL SEALS

The standard method of sealing the shaft is packing which is universally accepted and understood by industry. There are certain instances where mechanical seals have become a viable alternative to packing both from a performance and economic standpoint.

The advantages of a mechanical seal are the lack of periodic maintenance requirements and the fluid leakage associated with packing. There are basically two types of arrangements available with mechanical seals, a single seal and a double seal. A single seal consists of an O ring static seal in the housing and on the shaft with a single set of dynamic sealing faces which are usually made of carbon and ceramic. In a single seal installation the fluid being handled acts as a lubricant for the faces.

In applications where the fluid being handled is abrasive there are two choices. A very hard face material such as silicone carbide can be selected for a single seal installation or a double seal can be installed. In a double seal installation two standard seals are mounted opposing each other and a clean flush fluid is re-circulated through the area between the seals acting as a coolant and lubricant for the seals.

Sketches S-0271, S-0272, S-0514 in the data section show the arrangements and dimensions for the various pump sizes using a single and double seal.

MOUNTING AND VIBRATION

The Progressing Cavity pump is inherently an unbalanced machine due to the eccentric rotation of the rotor. The vibration which occurs is dependent on the size of the element, the offset and the speed of rotation. For this reason the speed of the pump is limited.

The magnitude of the induced vibration is of a low frequency and a relative high amplitude and will not produce offensive noise, however, it should be a factor in mounting. It is recommended that the pump be mounted to a level hollow base plate, which is securely lagged down to a concrete foundation with grouting poured into the hollow portion of the base for rigidity and dampening. The drive for the pump should be mounted on the same surface as the pump either as a direct or angle connection. The coupling in a direct connection should have at least 1/8 inch spacing to avoid excessive shaft loading. Check the Installation Operating and Maintenance Manual for detailed instructions.

BEARINGS AND CONNECTING RODS

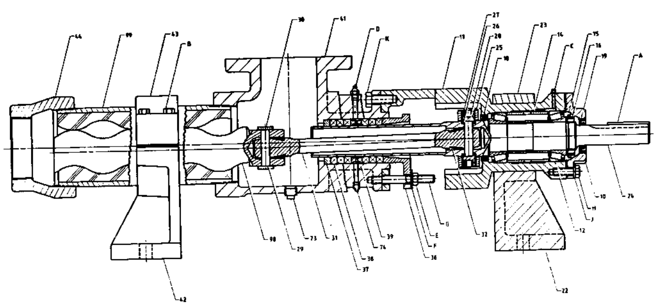
The Progressing Cavity elements are normally adapted to a drive end, which will provide an acceptable life span when properly applied and maintained. The AFBMA has developed a method of rating bearings which is called the “design life” or L10 life of the bearings when operated at a fixed load and rpm. This L10 bearing life for selected drive ends and Progressing Cavity elements at 100 rpm. To use the chart select the drive end or element and determine the thrust load in pounds by reading from the suction pressure or the differential pressure chart depending on the situation. Use this value to enter the curve at thrust-pounds and reading off the drive end line to find the L10 life in hours. This loading takes into consideration only the thrust load which is the major component of bearing load. An accurate analysis would also consider the radial load.

Example: A 72212 at 100 psi would have a thrust loading of 774 pounds. Entering the curve at 774 pounds and moving to the line for a 12 drive end we have an L10 life of 43,000,000 hours at 100 rpm. If the pump was running at 500 rpm the L10 life would be 43,000,000 x 100/500 or 8,600,000 hours. L10 life values in excess of 100,000 hrs are normally considered invalid because of other factors besides load which effect the bearing life.

If the pump were operated in reverse and the pressure was on the normal suction side the thrust loading would be 355 pounds instead of 774 pounds.

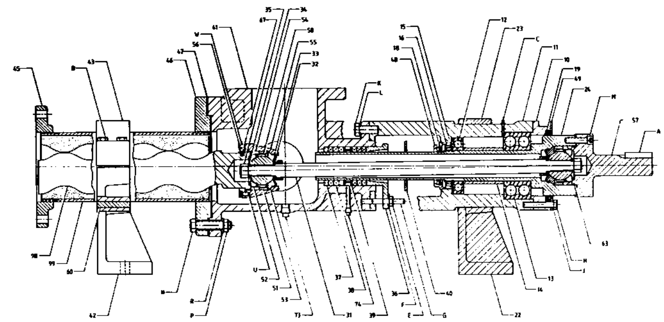
The normal connection between the drive shaft and the rotor is a connecting rod and pin. The slanted hole drilled in the connecting rod allows the rotor to freely move through the eccentric circle and carries the torque and thrust created by the pressure on the pump elements. When a carbon steel pump is used the pins and pinholes are hardened for maximum life.

When a stainless material is required for corrosion resistance the pinholes and pins cannot be hardened and the next largest drive end should be used when the differential pressure exceeds 75 psi or the speed exceeds 600 rpm.



Pin Joint Drive End

The heavy-duty pumps employ a gear type ball and socket joint, which has approximately 5 times the load carrying capacity of the conventional rod and pins. It is designed for only the larger size pumps where the initial expense is outweighed by the maintenance and replacement of a rod and pin.



Gear Joint Drive End

MATERIALS OF CONSTRUCTION

The Roper fluid list can be used as a guide in selecting materials of construction. There are however, several factors to consider which are unique to the progressing cavity pump and will influence the material selection. The stator bond s attacked if the pH of the fluid is greater than 10 and may cause bond failure. Applications where the pH is above 10 should be referred to the factory. When the pH is below 3.5 the chrome on the rotor is attacked and will lift off causing extreme stator wear. A non-plated rotor is recommended for such applications. The pH rating table in the data section and the Roper fluid list will act as a guide for pump selections. For fluids not covered by the fluid list a sample can be sent to the factory for analysis or small rubber slugs can be immersed to determine the correct elastomer.

Non-Abrasive Low Viscosity

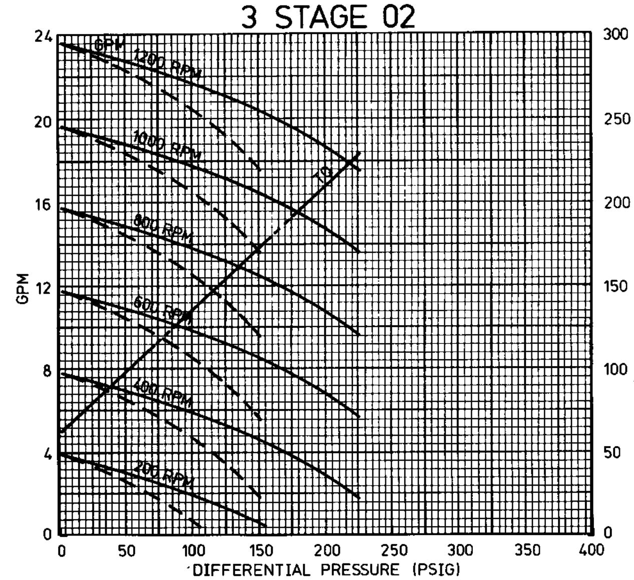
Example: Select a pump to delivery 15 gpm at 225 psi of 2% calcium carbonate water mixture. The operating temperature is 100 degrees F. and the suction is flooded. The particle size is 500microns.
Step A: This fluid can be classified as non-abrasive and the pump selected could be run at the maximum rpm. From the chart in the data section on 02 element can be selected which would have a displacement of 24 gpm at 1200 rpm.
Step B: A particle size of 500 microns is equal to 197 micro inches or .0197 inch and will pass through a 02 element, which will accept a .3-inch particle.
Step C: A discharge pressure of 225 psi will require 3 stages or the maximum of 75 psi per stage as listed in the data section.
Step D: So far we have selected a 3 stage 02 size pump. From the performance curve the slip at 225 psi is 6 gpm. A flow rate of 15 gpm is required at 225 psi, therefore a flow rate of 21 gpm (6 + 15) will be required at 0 psi. The displacement of the 02 element is 2 gpm per 100 rpm, therefore the pump speed will be 1050 rpm (21/2 x 100). The torque at 225 psi is 230 in lbs and the horsepower is 3.83
(230 x 1050 / 63025). From the torque chart the starting torque for 5 hp 1800 rpm design B motor is 324 in lbs. By belt driving the pump at 1050 rpm the torque at the pump will be
1800 / 1050 or 1.71 x 324 = 554 in lbs. From the curve the normal starting torque required by the pump is 168 in lbs. and at the operating temperature of 100 degrees F. it is 168 x 1.1 or 185 in lbs. A 5 hp motor will be adequate to start the pump.
Step F: The materials of construction can be cast iron housings with carbon steel internals and a Buna N stator or GHL construction

A 73202GHL pump driven at 1050 rpm by a 5 hp motor is selected.

PUMP ELEMENT DATA CHART

Element Size	Theo. Max. GPM (1 cPs)	Theo. GPM per 100 RPM	Max. Particle Size	Velocity per 100 RPM (ft/sec)	Shear Rate per 100 RPM (sec^{-1})	Initial Torque (lbf-in. / stage)	Hyd Torque (lbf-in. / stage)	Maximum Speed and Pressure for Abrasive Classes: None		Light		Medium		Heavy	
								RPM	PSI/Stg	RPM	PSI/Stg	RPM	PSI/Stg	RPM	PSI/Stg
006	.67	.056	.08	.41	148.5	6.8	.02	1200	60	900	50	600	30	300	15
025	3	.26	.15	.58	92.9	6.3	.1	1200	60	900	50	600	30	300	15
01	10	.86	.2	.87	93.0	15.7	.32	1200	75	900	60	600	35	300	15
02	24	2.02	.3	1.17	93.0	21	.74	1200	75	900	60	600	35	300	15
05	47	5.2	.4	1.55	78.2	41.4	1.91	900	75	675	60	450	35	225	15
12	105	11.7	.6	2.01	76.3	126	4.30	900	75	675	60	450	35	225	15
6-12	108	12	.7	2.03	63.2	101	4.4	900	75	675	60	450	35	225	15
19	141	18.8	.8	2.37	71.3	180	6.77	750	75	565	60	375	35	190	15
22	165	22	.85	2.47	63.5	172	8.1	750	75	565	60	375	35	190	15
28	208	27.7	.8	2.79	84.2	210	10.08	750	75	565	60	375	35	190	15
36	216	36	1.1	2.87	56.4	146	13.2	600	75	450	60	300	35	150	15
44	261	43.05	1.0	3.15	79.9	504	15.44	600	75	450	60	300	35	150	15
65	391	65.2	1.0	3.84	97.5	630	23	600	75	450	60	300	35	150	15
065	390	65	1.4	3.53	58.0	367	23.4	600	87	450	70	300	40	150	15
115	518	115	1.5	4.31	65.6	758	42.3	450	87	350	70	225	40	125	15
175	788	175	1.75	4.96	63.9	1370	58.8	450	87	350	70	225	40	125	15
335	1005	335	1.8	8.2	105.9	3420	117.5	300	87	225	70	150	40	75	15

3 STAGE 02



Abrasive Low Viscosity

Example: Select a pump to handle 10 gpm at 40 psi of lapping compound slurry of 30% concentration of fine (.01-.04) particles. The slurry temperature is 70 degrees F.
Step A: From the description of the material and the section in the technical manual this slurry would be judged as heavy abrasive.
Step B: Using the element data section chart a 05 element would be limited to 225 rpm and would deliver approximately the 10 gpm required.
Step C: From the data section for a heavy abrasive the maximum pressure per stage is 15 psi therefore 3 stages of element would be required.
Step D: The 05 element will pass a .4 inch particle and will be suitable.
Step E: From the curve for a 3 stage 05 pump the slip at 40 psi is 1 gpm. The desired flow rate is 10 gpm, therefore, the flow rate at 0 psi should be 11 gpm. A 05 element would have to run at 225 rpm (11/5 x 100). The torque required at 40 psi is 195 in lbs and the torque added from Table C on the performance curve is 338 in lbs. The total torque is 536 in lbs and the hp at 220 rpm would be 1.87 hp.
Step F: Materials of construction would be cast iron, carbon steel internals and Buna N stator.

The pump selection is 73205 running at 225 rpm driven by a 2 hp motor.

PUMP ELEMENT DATA CHART

Element Size	Theo. Max. GPM (1 cPs)	Theo. GPM per 100 RPM	Max. Particle Size	Velocity per 100 RPM (ft/sec)	Shear Rate per 100 RPM (sec^{-1})	Initial Torque (lbf-in. / stage)	Hyd Torque (lbf-in. / stage)	Maximum Speed and Pressure for Abrasive Classes: None		Light		Medium		Heavy	
								RPM	PSI/Stg	RPM	PSI/Stg	RPM	PSI/Stg	RPM	PSI/Stg
006	.67	.056	.08	.41	148.5	6.8	.02	1200	60	900	50	600	30	300	15
025	3	.26	.15	.58	92.9	6.3	.1	1200	60	900	50	600	30	300	15
01	10	.86	.2	.87	93.0	15.7	.32	1200	75	900	60	600	35	300	15
02	24	2.02	.3	1.17	93.0	21	.74	1200	75	900	60	600	35	300	15
05	47	5.2	.4	1.55	78.2	41.4	1.91	900	75	675	60	450	35	225	15
12	105	11.7	.6	2.01	76.3	126	4.30	900	75	675	60	450	35	225	15
6-12	108	12	.7	2.03	63.2	101	4.4	900	75	675	60	450	35	225	15
19	141	18.8	.8	2.37	71.3	180	6.77	750	75	565	60	375	35	190	15
22	165	22	.85	2.47	63.5	172	8.1	750	75	565	60	375	35	190	15
28	208	27.7	.8	2.79	84.2	210	10.08	750	75	565	60	375	35	190	15
36	216	36	1.1	2.87	56.4	146	13.2	600	75	450	60	300	35	150	15
44	261	43.05	1.0	3.15	79.9	504	15.44	600	75	450	60	300	35	150	15
65	391	65.2	1.0	3.84	97.5	630	23	600	75	450	60	300	35	150	15
065	390	65	1.4	3.53	58.0	367	23.4	600	87	450	70	300	40	150	15
115	518	115	1.5	4.31	65.6	758	42.3	450	87	350	70	225	40	125	15
175	788	175	1.75	4.96	63.9	1370	58.8	450	87	350	70	225	40	125	15
335	1005	335	1.8	8.2	105.9	3420	117.5	300	87	225	70	150	40	75	15

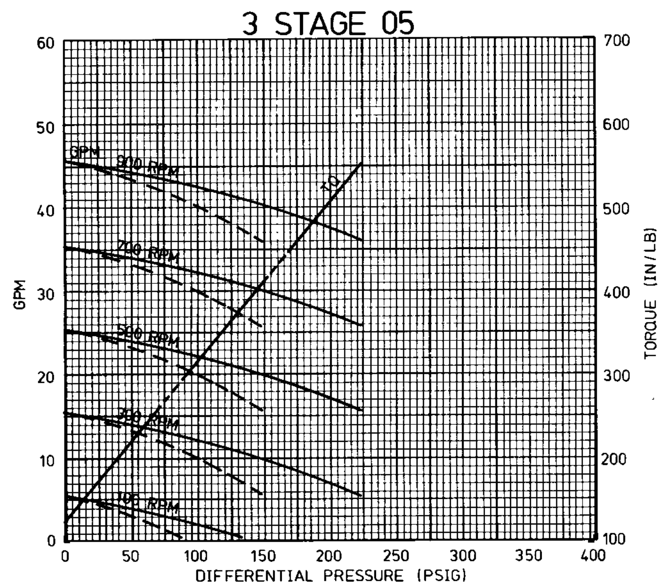


TABLE C WATER BASE SLURRY TORQUE ADDITIVE (IN/LB)

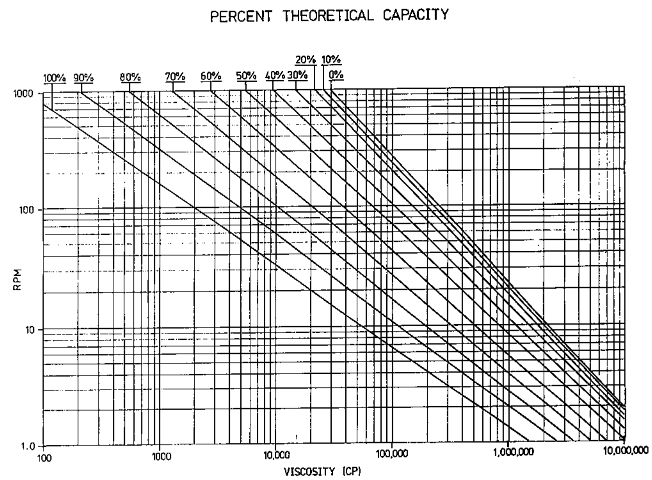
NOTE: MAXIMUM PARTICLE SIZE .4 INCH

% \ SIZE	FINE .01" TO .04"	MEDIUM .04" TO .08"	COARSE .08" & LARGER
10	113	150	228
30	338	449	683
50	563	749	1139

ABRASIVE HIGH VISCOSITY

Example: Select a pump for handling hand cleaner with fine abrasive grit at a rate of 20 gpm at 75 psi. The fluid has 20% solids and the carrier fluid has a measured viscosity of 5000 cps. at a shear rate of 100 inverse seconds. The temperature is 70 degrees F.
Step A: This material can be judged to be a medium abrasive fluid.
Step B: From the theoretical capacity chart a 5000 cps fluid would have a limiting speed of 600 rpm for 60% volumetric efficiency.
Step C: Since the required flow is 20 gpm an element is selected that has a theoretical displacement of at least 5.5 gpm/100 rpm (20/6.0/.6).
Step D: A medium abrasive fluid is limited to approximately 35 psi per stage. The requirement is for 75 psi, therefore, at minimum a 2-stage element is selected. The curve for a 2-stage 12 shows 3 gpm slip at 75 psi for water. The slip for a 5000 cps fluid would be approximately 3/5.5 or .5 gpm so the pump displacement of 0 psi would have to be 20.5 gpm (20 + .5). A 12 element would have to run at 170 rpm with 100% volumetric efficiency. Checking curve S0510 for a 5000 cps fluid shows 80% volumetric efficiency at 170 rpm therefore the pump speed should be 212 rpm (170/.8).
Step E: The torque from the performance curve is 580 in lbs at 75 psi. The torque adder for 5000 cps fluid is 640 in lbs while the torque adder for 20% solids is approximately 300 in lbs. The viscosity will have more influence on torque than the solids so only the torque adder for viscosity is added. The total torque is 1220 in lbs (580 + 640) and the horsepower is calculated as 4.1 hp.
Step F: Since the temperature is 70 degrees F the calculated values are acceptable. The materials of construction are cast iron, carbon steel internals, and Buna N.

The pump selected is a 72212GHL at 212 rpm with a 5 hp motor.



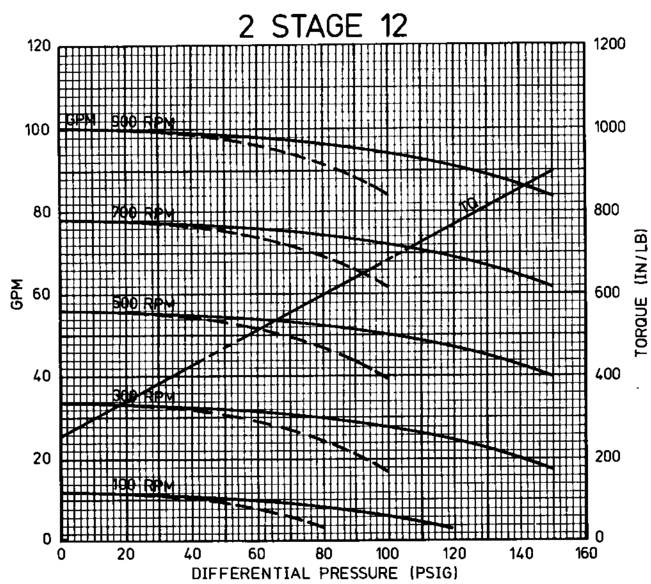


TABLE B APPARENT VISCOSITY - TORQUE ADDITIVE (IN/LB) & MAX. SPEED

CPS	100	1000	2500	5000	10,000	50,000	100,000	150,000	200,000
TQ	104	304	464	640	890	1860	2560	3090	3550
RPM	900	900	900	600	320	80	40	30	25

TABLE C WATER BASE SLURRY TORQUE ADDITIVE (IN/LB)

NOTE: MAXIMUM PARTICLE SIZE .6 INCH

SIZE / %	FINE .01" TO .04"	MEDIUM .04" TO .08"	COARSE .08" & LARGER
10	149	186	277
30	448	559	830
50	746	931	1384

Non-Abrasive High Viscosity

Example: Select a pump to handle a viscous caulking compound at a rate of 10 gpm against a 50-psi discharge pressure. The temperature is 70 degrees F.

Step A: The fluid viscosity measured 150,000 cps at a shear rate of .15 seconds, and 100,000 cps at a shear rate of .6 inverse seconds. Assuming that it is a thixotropic fluid estimate a viscosity of 10,000 cps at a shear rate of 100 inverse seconds.

Step B: From the theoretical capacity chart, using 10,000 cps, the maximum speed would be about 320 rpm for 65% volumetric efficiency.

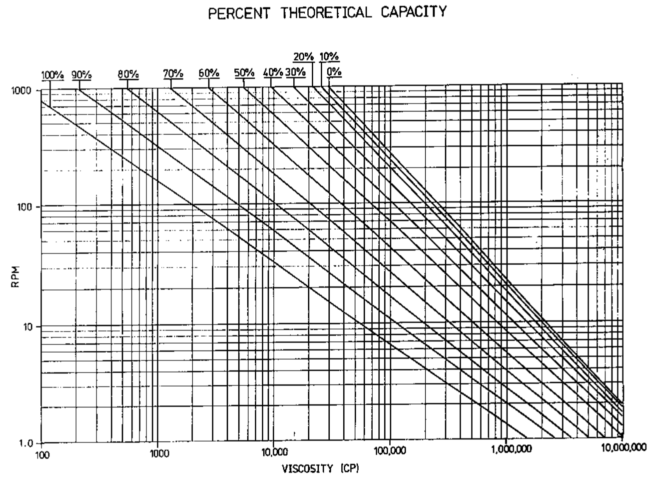
Step C: For a requirement of 10 gpm tentatively select a 12 element. From the chart in the data section a 12 element would have a shear rate of 77.2 inverse seconds at 100 rpm, which is close to the initial selected shear rate of 100 inverse seconds. The volumetric efficiency at 10,000 cps and 100 rpm would be 80%. Using a 12 element at theoretical displacement and an 80% volumetric efficiency the displacement would be 9.6 gpm (12 x 80%).

Step D: Since the discharge pressure is less than 75 psi a single stage pump can be used. From the curve for a single stage 12 (71x12) the lip for water at 50 psi is 8 gpm. The estimated slilp for 10,000 cps would be 8/S.I. or 8/6.15 1.3 gpm. The capacity is now 9.6 gpm / 100 rpm so the pump would have to run at 118 rpm (11.3 / 9.6 x 100).

Step E: The torque at 50 psi is 340 in lbs. The torque adder from Table B on the performance curve is 445 in lbs. The total torque in 785 in lbs (340 + 445). The horsepower required is 1.47 hp (785 x 118 / 63025). The starting torque necessary for a 71212 is 408 in lbs. Using a 2 hp motor and gearing down to 118 rpm the starting torque available would be 200 in lbs or more than enough to start the pump.

Step F: Since the operating temperature is 70 degrees F. there will not be any adjustment and the materials of construction can be cast iron, carbon steel and Buna N.

The pump selected is a 71212 GHL at 118 rpm with a 2 hp motor.



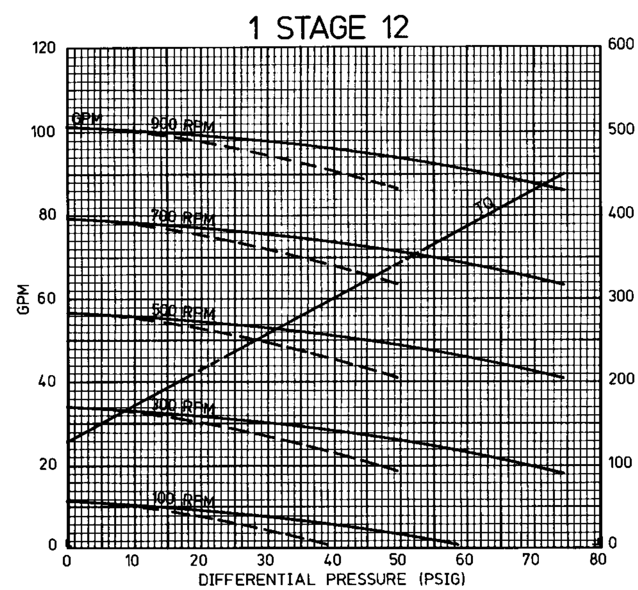


TABLE B APPARENT VISCOSITY - TORQUE ADDITIVE (IN/LB) & MAX. SPEED

CPS	100	1000	2500	5000	10,000	50,000	100,000	150,000	200,000
TQ	52	152	232	320	445	930	1280	1545	1775
RPM	900	900	900	600	320	80	40	30	25

PROGRESSING CAVITY PUMP DATA SHEET

Capacity Required ______________________________gpm

Differential Pressure _____________________________psi

Apparent Viscosity _______________________________cp

Particle Size _______________

% Solids ___________________

Abrasive Class
None ____________________
Light ____________________
Medium __________________
Heavy ___________________

Temperature _______________Deg. F

Rotor Standard_________________
Undersized_______________
Double Undersized_________

Temperature Multiplier_______________

Pump Selection ______________________

Materials of Construction ______________________

Slip on Water _________________

Corrected Slip _________________

Corrected RPM ________________

Initial Torque __________________ Corrected Initial Torque ________________(1)
Hydraulic Torque ________________(2)
Viscous Torque ________________(3)
Solids Torque ________________(4)

Corrected Starting Torque _____________________

(Add Lines 1,2,3,& 4) Total Torque _________________

$$HP = \frac{TQ \times RPM}{63025}$$

HP ________________