

TUTORIAL ON SPECIAL PURPOSE PUMPS— PITOT; PROGRESSING CAVITY; AIR OPERATED DIAPHRAGM; AND HYDRAULICALLY ACTUATED DIAPHRAGM

by

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INTRODUCTION



Bruno Schiavello has been Director for Fluid Dynamics at Ingersoll-Dresser Pump Company, Corporate Technology, Phillipsburg, New Jersey, since 1993. He received a B.S. degree (Mechanical Engineering) from the University of Rome, Italy (1974), and an M.S. degree (Fluid Dynamics) from Von Karman Institute for Fluid Dynamics, Rhode St. Genese, Belgium (1975).

Mr. Schiavello was co-winner of the H. Worthington European Technical Award in 1979. He has written several significant papers and lectured at seminars in the area of pump recirculation, cavitation, and two-phase flow. He is a member of ASME, AIAA, Societe Hydrotechnique de France (SHF), and the International Association for Hydraulic Research

(IAHR). He has served on the International Pump Users Symposium Advisory Committee since 1983.

Mr. Schiavello started in the Research and Development Department of Worthington Nord (Italy). In 1982, he joined the Central Research and Development of Worthington, McGraw Edison Company (USA), and continued with Dresser Pump Division as Manager of Fluid Dynamics.

PUMPING SERVICES

There are pumping services characterized by peculiar conditions for which special purpose pumps (quite different from mostly known centrifugal pumps) represent the most effective selection in terms of performance, reliability, maintenance, and cost.

The requirements for pumping duties in the industrial processes and civil services can cover a very broad range of conditions in terms of: capacity, head, suction conditions, physical properties of the pumped media, and quality standards for the process products. In most cases, these requirements are mutually conflicting, and so

a compromise must be found in order to select the pump type/size that fits the end user needs, both efficiently and economically. In several circumstances, some requirements are critically overriding (special pump services) and consequently dictate the selection of pump types, which operate with pumping principle not quite common or popular (special purpose pumps). The classification of pumps into "common types" and "special purpose" introduced here is mainly in the light of the difficulty of the pumping service, especially as related to performance and/or pumped media. The classification has nothing to do with either the complexity of the pump design (hydraulic and mechanic) or the engineering level of the features incorporated in the pump. On the other hand, the rotordynamic pumps, which include centrifugal-, mixed-, and axial-flow pumps are here considered as "common type" pumps, because they represent the largest fraction of the pump population. Moreover, the rotordynamic pumps, especially centrifugal-flow pumps, are extensively covered in the open literature and so a large amount of technical information is available to the pump designer and end user as well. Sometimes the working physics and construction of the rotordynamic pumps are the best match for difficult pumping services and give the most satisfactory results for the end user.

Some very simple criteria are presented in Table 1 to help the engineer and/or end user in identifying, at least as a first preliminary idea, peculiar pumping services that may require special purpose pumps.

The fluid temperature is not listed as an independent parameter in Table 1, but it can have a key role for the pump selection through its influence on various factors (NPSHA, volatility, specific gravity, viscosity). Moreover, the corrosiveness of the pumped medium is important, more for proper selection of pump materials than pump type.

Table 1. Pumping Services: Parameters and Peculiarities.

Pumping Services: Parameters and Peculiarities	
Pumping Parameter/Aspect	Pecularity
Capacity + Head	Low + High (up to 600 to 800 gpm) (1000 to 5000 ft)
Specific Speed	50 to 350 US gpm
Suction Conditions: NPSHA (pressure/temperature) Gas content Vaporization/Flashing Volatility Volatility	Low/Variable Medium to High 10%-15% to 90%-100% (Volume) Possible with large volume High (depending on components)
Self-Priming	In dry conditions
Dry-running	For a long period of time
System head/capacity changes	Case by case (shape of H-Q curve; variable speed)
Allowable pressure pulsations	Low (suction and/or discharge)
Fluid specific gravity	Low (< 0.5) / High (> 1.3)
Fluid viscosity	Medium-High (3,300 cP to 4.5 Y 10 ⁶ cP) (15,000 SSU to 500 million SSU)
Fluid rheology property Viscosity vs shear rate Newtonian fluids (water, oils, ...) Non-Newtonian fluids Bingham plastic Thixotropic (plastisols, ketchup, ...) Dilatant (high concentrate slurries, ...)	Special pumping behavior: Related to viscosity value All situations
Abrasive media	Medium to high abrasiveness (particles/carrier fluid/velocity)
Slurries/Sludges (Waste treatment) Nonsettling Newtonian Nonsettling, non-Newtonian Settling Organic sludges (food, pharmaceutical, chemical, petrochemical, textile industry)	Pumping difficulty: Normal (derating per HIS) Special design Special care Hazardous waste (pathogen: bacteria, viruses; toxic)
Hazardous Media/Areas Toxic (poisons, allergens, ...) Flammable (organic solvents, ...) Dangerous (explosives, oxidizers, ...)	Excellent sealing by design (zero emission)
Paper stocks (consistency: percent fiber content) Low (1 to 7%) Medium (8 to 15%) High (> 15%, as bleaching operation)	Suitable pump selection: Modified centrifugal Special centrifugal Positive displacement
Fluidized solid (carrier liquid + solids) Granulated solids (mixture) Powder Lightweight bulk materials (chemical industry)	Minor product damage: Virtually no shearing action (Gentle, slow transport)

PUMP TYPES

The gamut of pumping services is very broad and heterogeneous, including special applications as determined by dominating parameters close to "extreme" values. Then it can be expected that no single class of pumps can be developed by the designer or selected by the user to handle the whole range of problems encountered in the pump world. Just for example, there are more than 30 different pump types used for slurry applications.

The pump classes include a wide variety of pumping conceptual approaches that have been elaborated, improved, and transformed into a series of pump types, presently available for the end user. All the existing pump types are grouped into two main categories:

- Kinetic (rotordynamic) pumps
- Positive displacement pumps

Kinetic (Rotordynamic) Pumps

The rotordynamic, or kinetic, pumps accelerate fluids by means of a rotating element (impeller) enclosed in a housing (casing). The pumping physical principle is that the mechanical energy available at the shaft coupling (brake horsepower) is converted into kinetic energy (change of the momentum of the flow stream), which is mainly transformed in the form of pressure energy (total dynamic head). This process of transfer and conversion of energy is realized through complex fluid-dynamic phenomena involving centrifugal forces action and blade fluid forces (aerodynamic lift and drag mechanisms), associated with high turbulence and strong velocity gradients. Thus, intense shear rates can be present, which broadly change by value in the rotor and in the casing.

Pump-discharge pressure varies with flowrate, pumped medium density, and rotational speed. The flow path through the pump is continuous from suction to discharge, giving a flow stream with relatively low pressure pulsations at pump suction and discharge. As a simple concept for user comprehension, the key working mechanism of rotordynamic pumps is pressure generation in a stream of flow. In general, rotordynamic pumps are more suitable to handle Newtonian fluids with low-medium viscosity and "pseudo-homogeneous" multiphase media (i.e., low gas content, low fiber consistency) as the various phases constituents are finely dispersed and mixed with the liquid phase (carrier). On the other hand, the handling of highly abrasive media and dense concentration slurries requires specific consideration, case by case.

The category of rotordynamic pumps includes these groups of pump types:

- Vaned impeller (centrifugal-, mixed-, axial-flow)
- Propellers
- Co-rotating discs (shear pumps) (special purpose)
- Archimedean screw (special purpose)

Overall, the rotordynamic pumps are capable (as single stage) of moderately high discharge heads and very high pumping capacities, covering a range of specific speed (Ns) from 50 to 15,000 US gpm.

The group of centrifugal pumps includes:

- Conventional design
- Regenerative turbines (special purpose)
- Pitot pump (rotating casing)

Positive Displacement Pumps

The positive displacement pumps isolate discrete volumes of fluid in cavities between the moving and stationary parts of the pump. The cavities are moved from suction to discharge parts by direct mechanical action. Thus, the pumping physical principle is

that the mechanical energy (brake horsepower) is converted directly into pressure and, unlike rotordynamic pumps, pressure is not a function of displacement flow. There are no high velocity gradients and, thus, the shear rates are relatively low.

The pressure, or head, developed is limited by the pump mechanical design and the power available. The flow path is discontinuous, resulting in pulsating flow, which can be smoothed by means of pulsation dampeners (suction and/or discharge). As a simple concept for user comprehension, the key working mechanism of the positive displacement pumps is transport (displacement) of a confined volume (mass) of medium from low to high pressure levels, which exist independently from the pump itself.

In general, positive displacement pumps are more suitable to handle highly viscous Newtonian fluids, non-Newtonian fluids, and “nonhomogeneous” multiphase media, with medium to high concentration of the nonliquid constituent phases (gas, wood fibers, solids, etc.). The pumping capability for abrasive media and slurries varies from type to type, requiring specific evaluation for each case.

The positive displacement pumps category includes two major classifications: rotary pumps and reciprocating pumps.

Rotary Pumps

A rotary pump consists of a housing containing screws, gears, sliding vanes, radial pistons, etc., which are actuated by relative rotation of the drive shaft to casing. The pump has no separate inlet and outlet valves. These pumps are characterized by their close-running clearances. Rotary pumps operate in generally the same pressure ranges as centrifugal pumps, except at lower capacities.

The screw pumps can be further classified as:

- Single-screw (progressive cavity).
- Two-screw (timed).
- Three-screw (untimed).

Reciprocating Pumps

A reciprocating pump is formed by one or more chambers in which a piston is moving with alternative motion (stroke: frequency, length, cycle). The alternative motion is derived from the rotational motion of the motor shaft by means of the drive train of the pump. The drive train consists of gearing and power mechanisms, which usually include some type of cam and follower or crank principle. The pump has suction and discharge valves in each chamber. Reciprocating pumps cover generally a range of pressure higher than centrifugal pumps, but at lower capacities.

The reciprocating displacement pumps include two subgroups:

- Piston pumps
- Diaphragm pumps

SPECIAL PURPOSE PUMPS

There are numerous pumping services that present special conditions (performance range, pumped medium) as shown in Table 1. On the other hand, the number and type of suitable pumps developed by the pump manufacturers and presented to the end user is quite extensive, in addition to the most common and usually applied pump group (i.e., rotordynamic, especially centrifugal types), as is readily evident from the above pump categories. Therefore, the pumping cases for which special purpose pumps may represent the best fit are quite frequent and extremely various.

Only four types of special purpose pumps are presented herein, and all special applications are not covered. However, the pump types discussed belong to three main groups (rotordynamic-centrifugal, rotary-positive displacement, and reciprocating-positive displacement). Then, some basic insights

with notable significance can be distilled by the end user, as guidance in the future to identify the best fit pump for his pumping requirements, peculiar or normal. Also, there is the need to stimulate more interest in special purpose pumps so that, in the future, other pump types will be presented for end user benefit.

Pitot Pump

The pitot pump is a specially designed rotordynamic centrifugal flow pump. It extends the specific speed domain of the centrifugal flow pumps to lower values of the range from 50 to 350 US, with flow up to 800 gpm and head up to 5500 ft in one stage configuration. The efficiency is good, considering capacity and specific speed. The rotating casing and high head coefficient are the design aspects that are most peculiar. The section is aimed at providing the pump user with a better understanding of how this type of pump can best be evaluated for, and applied to, high head low flow applications.

Progressing Cavity Pump

The progressing cavity pump is a unique type of rotary-positive displacement pump, characterized by a single screw design style. As typical of the rotary pumps, the progressing cavity pump is capable of handling a wide range of fluid viscosities, at medium pressures (up to 2000 psi) over a broad range of flowrates (up to 2000 gpm). However, while rotary pumps are not particularly suitable for handling severely abrasive fluids, the single-screw progressive cavity pump performs well in pumping abrasive slurries. Further, this pump has a very low shear rate and, so, is particularly suitable for pumping shear sensitive media. The section presents pros and cons for application of this pump, giving insights about the troubleshooting approach.

Air Operated Double Diaphragm Pump (AODD)

The air operated double diaphragm pump is a versatile type of reciprocating-positive displacement pump of very simple construction. The pump uses flat, flexible diaphragms that are hermetically sealed in the housing. The motive fluid is compressed air on the side of the diaphragm (pneumatic diaphragm pump) and impacts its pressure via the diaphragm to the pumped fluid on the other side.

A characteristic feature of these pumps is that they operate with virtually no shearing action as internal fluid velocity is typically three m/s (compared to 20 m/s or much higher within other types of pumps). In other words, the pump treats the pumped media with great care and so is a preferred selection in the chemical industry to transport fluidized solid and light-weight bulk material in solid. These pumps cover a flow range up to 750 gpm and differential pressures up to 125 psig. The section describes how the AODD pumps are excellently suited for transfer applications with “problem media” in relation to viscosity, abrasiveness, solid concentration, volatility, corrosiveness, and shear sensitivity.

Hydraulically Actuated Diaphragm Metering Pump

The hydraulically actuated diaphragm metering pump is a reciprocating displacement pump that uses a flat, hermetically sealed diaphragm. The motive fluid is a hydraulic and chemically stable fluid, which is displaced by a piston or plunger pumps and imparts its pressure to the process medium via the diaphragm. The pumping duties of these pumps embrace a coverage area with flows to 1500 gph (25 gpm) and discharge pressures up to 5000 psig with a simplex configuration of the pump. Metering pumps are also called “dosing pumps” and “proportioning pumps.” Metering pumps are used to add small quantities of liquid to other liquid streams or vessels. The basic requirements of a metering pump are to deliver measured volumes of liquids accurately without significant changes in volume as discharge pressure varies.

Typical accuracies would be ± 1.5 percent of the rated flow. The section shows how the hydraulically operated diaphragm pump represents a proper choice for special applications requiring metering and closing accuracy, leakfree chemical transfer, controlled handling of toxic, corrosive, or expensive fluids. Further, pump selection criteria are presented along with general application guidelines for pump control and operation.

APPLICATION OF THE PITOT PUMP



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Mr. Angle is a member of ASME, NACE, and ASM. He received a B.S. degree (1969) and an M.S. degree (1972) in Engineering from the University of California and is a registered Professional Engineer (Mechanical) in the State of California.



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EnviroTech in 1992, he held various positions from Research Engineer to Slurry Pumps Department Manager at NPO VNII Gidromash in Moscow.

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submersible pump product line.

Mr. Shaw received a B.S. degree (Mechanical Engineering) from the University of Cincinnati (1977).

ABSTRACT

The pitot pump is a little known variation of the centrifugal pump design. It has an impeller similar to a centrifugal pump, but instead of a volute or a diffuser, it incorporates a pitot tube to capture the discharge flow and a rotating case. This design obeys the centrifugal pump affinity laws, which govern the performance changes for variations in speed. The rotating case allows the pump to produce more head than a similar sized centrifugal pump operating at the same speed. The advantages and disadvantages of this design, as they relate to the application of the pump, are presented. The hydraulic design principles that characterize this pump are presented, so the pump user will have a complete understanding of the pump design, how the head is generated and how this pump can best be applied.

INTRODUCTION

History

The pitot pump is a relatively new pump design. It first appeared in U.S. patent history in the early 1900s. The original pumps had an open case design that resulted in severe performance limitations. In the 1920s, the closed rotating casing was first patented [1]. This design was studied in Germany and England during the 1940s for use in rockets and aircraft [2]. The study was dropped when problems with air locking were encountered, and at the same time the conclusion was drawn that the pump was limited to flows of less than five gpm. In the late 1950s and early 1960s, design improvements, such as the enclosed spin-up passages and other developments, increased the pump efficiency and capacity to levels that allowed it to be accepted by industry (Figure 1).

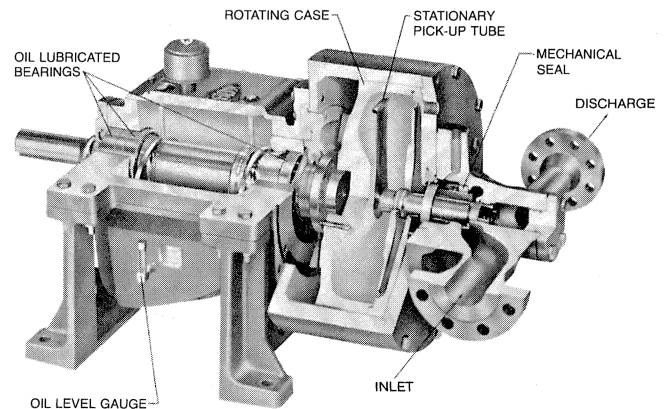


Figure 1. Pitot Pump.

Pump Description

The pitot pump comprises a closed rotating casing with a stationary pitot tube that extends into the rotating case along the axis of rotation with its inlet positioned near the maximum inner diameter of the casing as shown in Figure 2. The liquid enters the casing along the axis of rotation and picks up momentum as it passes through the enclosed radial vanes of the impeller into the rotating casing. The liquid is spun up to approximately the full rotational speed of the casing. It then impacts the inlet orifice of the pitot tube near the periphery of the rotating casing. (This is where the pressure and rotational velocity of the liquid mass are the greatest.) The liquid is discharged through the inner passageway of the pitot tube and out of the pump. The head developed is equal to the sum of both the static pressure head created by the centrifugal force and the velocity head. This sum is equal to approximately 1.6 times the head produced from a conventional centrifugal pump of the same size and speed.

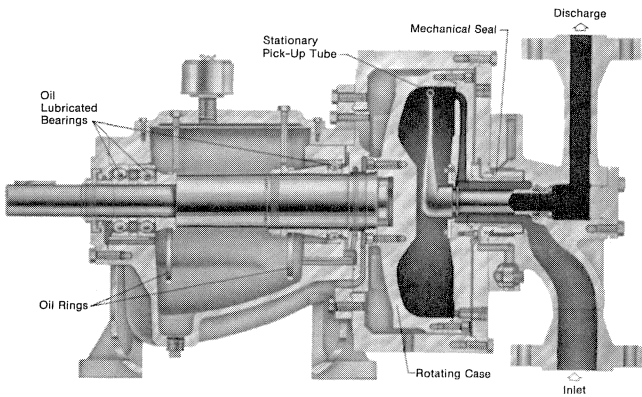


Figure 2. Pump Section.

PERFORMANCE

Hydraulic Theory

The pitot pump develops its head in two mechanisms, by means of both a centrifugal (static) and a velocity head. The combination of these two produces the total head developed by this pump [3].

Centrifugal Head

If we consider the rotating casing as a cylinder filled with liquid, as it rotates there will be an internal pressure developed by the centrifugal force field according to the following equation:

$$P_c = \frac{\rho \times \omega^2 \times r^2}{2g} \quad (1)$$

where:

P_c = Centrifugal pressure (lb/in²)

ρ = Liquid density (lb/in³)

ω = Rotating casing angular velocity (rad/sec)

r = centerline radius of the pitot tube inlet (in) (Figure 3)

g = 386.4 in/sec²

In terms of head (ft) of liquid (for water at SG = 1.0), this equation can be expressed as follows:

$$H_c = \frac{0.036 \times (N)^2 \times 4\pi^2 \times r^2 \times 2.31}{2g \times 3600} \quad (2)$$

where:

N = rotational speed (rpm)

H_c = Head (ft)

Velocity Head

The maximum velocity head developed can be calculated by the familiar expression:

$$H_v = \frac{v^2}{2g} \quad (3)$$

where:

H_v = Maximum velocity head (ft)

V = Liquid velocity (ft/sec)

g = 32.2 ft/sec²

because:

$V = r \times \omega$

$\omega = N \times 2\pi/60$

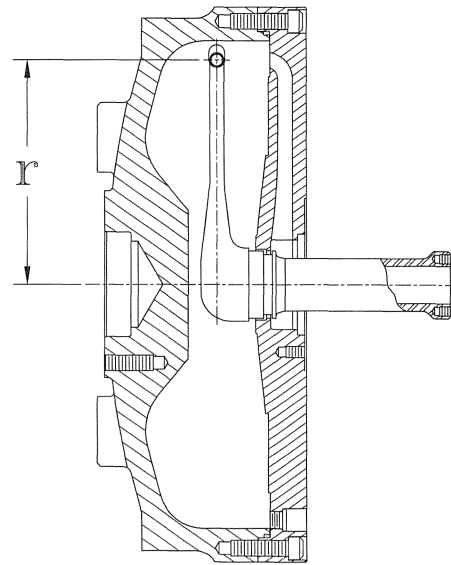


Figure 3. Rotating Casing and Pitot Tube.

H_v can be expressed as follows

$$H_v = \frac{(r \times N \times 2\pi)^2}{2 \times 3600 \times g \times 144} \quad (4)$$

Total Head

The total maximum head (H_t) developed by a pitot pump (assuming no hydraulic losses) is approximated by the following equation:

$$H_t = H_v + H_c \quad (5)$$

Substituting g with an appropriate value:

$$H_c = \frac{r^2 \times N^2 \times 3.28}{2782080} \text{ (ft)}$$

$$H_v = \frac{r^2 \times N^2 \times 3.29}{2782080} \text{ (ft)} \quad (6)$$

$$H_t = \frac{r^2 \times N^2}{423452} \text{ (ft)} \quad (7)$$

The equations above are not significantly different from the theoretical (virtual) head produced by a conventional centrifugal pump, if operating close to zero flow, and in the absence of slip and losses. However, the conversion of velocity to pressure head by the pitot tube is much more efficient than in a conventional centrifugal pump. Moreover, the disc friction is much higher in a centrifugal pump. A conventional centrifugal pump may, in practice, produce a maximum head ideal (no loss) in the range of 60 percent to the theoretical (virtual) value. In contrast, the pitot pump, as shown in Table 2, can produce an actual maximum ideal head very close to the theoretical value. The actual head (at the pump discharge flange) is lower than the maximum ideal head because of the hydraulic losses.

There are some additional observations regarding the development of head by a pitot pump.

- First, from Equation (6) it is apparent that the theoretical centrifugal and velocity heads are practically equal. This has been

shown by numerous observations in the field on startup when the motor has been wired to allow the pump to rotate backwards. In these situations, it has been observed that the head developed by the pump is approximately 50 percent of what is shown on the performance curve. It has also been observed that when the pitot tube is worn off due to abrasive service and no velocity head can be utilized, the observed head is 50 percent of the performance curve head.

- Second, according to Equation (7), the value of H_t is determined using the rotating casing speed N to calculate the actual fluid velocity. In fact, this is not strictly correct in that there is always some slippage and the mass of the fluid will rotate at an angular velocity of something less than the rotating casing speed. How much slippage takes place is a function of the hydraulic friction and turbulent exchange within the rotating casing itself [4]. The friction is affected by both the size of the pitot tube and the number of pitot tube heads. For the smallest size of the pitot tube used, the actual maximum head developed is very close to the theoretical head. For larger and double headed pitot tubes, the observed maximum head decreases significantly from the calculated value.

For example: A pitot pump operating at 4380 rpm and having a pitot tube radius of 7.7 in will theoretically develop a maximum total head of 2690 ft. The actual maximum head, as shown in Table 2, decreases as the pitot tube diameter (and its associated housing and support wing) increase and also decreases as the number of heads on the pitot tube increase from one to two.

Table 2. Actual vs Theoretical Maximum Head ($N = 4380$ rpm, $r = 7.7$ in).

Pitot Tube Diameter (in)	Number of Heads	Maximum Head from Test (ft)	Percent of Theoretical Head
0.266	1	2605	97
0.375	1	2460	92
0.375	2	2477	92
0.484	1	2363	88
0.484	2	2334	87
0.6	1	2318	86
0.6	2	2250	84
0.85	2	2081	77

- Third, it is important to note the very significant difference between a rotating casing pitot pump and a conventional centrifugal pump from the standpoint of internal friction. Conventional centrifugal pumps having an impeller diameter similar to that of a pitot pump are limited in their practical rotational speeds due to disc friction effects. These effects can be high because: a) the fluid between the impeller shrouds and the stationary casing rotates at approximately one-half of the impeller angular velocity, and b) the impeller shroud surface area can be quite high if the impeller diameter is too large. This is why centrifugal pumps with the same flow and head characteristics of a pitot pump have very small diameter impellers that operate at very high speeds, often above 25,000 rpm.

In contrast, the friction between the mass of fluid and the walls of the rotating casing is not a function of the speed of the casing but of the relatively low slippage velocity differential between the two, because the fluid and casing rotate at nearly the same angular velocity. The only frictional effects taking place at the full fluid velocity are those between the fluid and the relatively small pitot tube assembly, which presents a comparatively small friction surface.

It is the combination of more effective conversion of centrifugal plus velocity heads developed by the pitot pump and the minimal friction that allows this type of pump to develop high heads at relatively modest speeds and at good overall efficiencies.

Typical Performance

There are two methods of changing the performance characteristics of the pump. The first method is to change the size of the pitot tube in the pump. This changes the capacity characteristics of the pump. The second method is to change the speed at which the pump is operated. In order to represent both of these methods, a typical performance curve is generated for each size pitot tube with a series of head and horsepower curves for each of the common operating speeds. The typical performance curves for a single size pitot tube are shown in Figures 4 and 5.

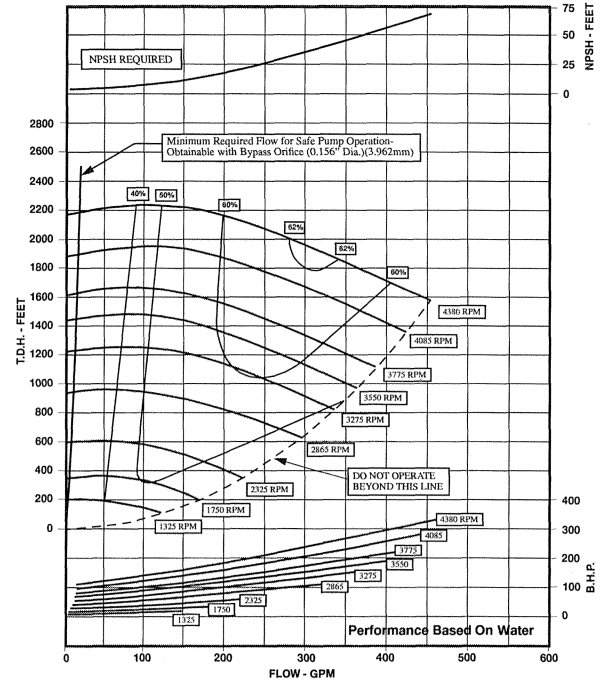


Figure 4. Performance Curve, 1325 RPM to 4380 RPM.

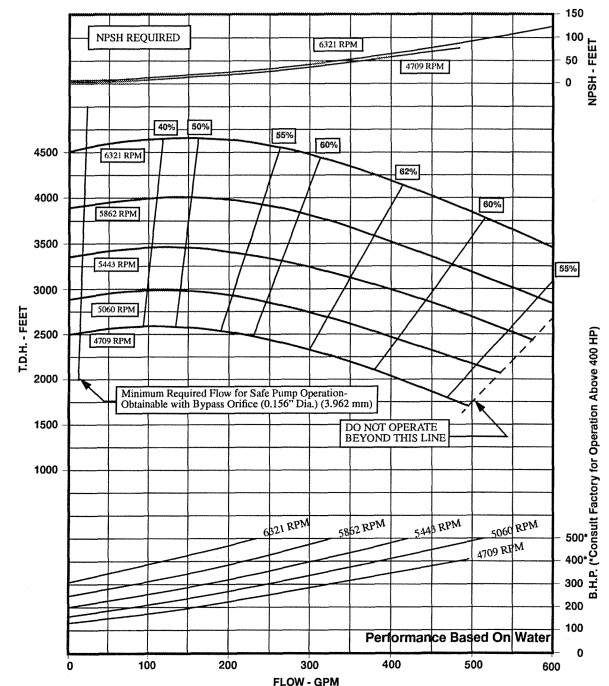


Figure 5. Performance Curve, 4709 RPM to 6321 RPM.

Performance Range

The performance range of this pump with the current commercially available pitot pump technology is shown in Figure 6. Recent developments have allowed the maximum capacity of the pump to increase from 400 gpm to 800 gpm. The maximum published head is 5500 ft, obtained at a pump speed of 6321 rpm. While 6321 rpm is the published maximum speed of the pump, it is capable of higher speeds and higher heads. Efficiencies of up to 62 percent have been reached with this pump design.

Specific speed and suction specific speed are often used to describe the hydraulic features of a pump.

Specific speed:

$$N_s = \frac{N \times Q^{0.5}}{H^{0.75}} \text{ (rpm, gpm, ft)} \quad (8)$$

Suction specific speed:

$$S = \frac{N \times Q^{0.5}}{NPSH^{0.75}} \text{ (rpm, gpm, ft)} \quad (9)$$

Typically, the specific speed for this type pump ranges from 60 to 360. The suction specific speed ranges from 5000 to 9000.

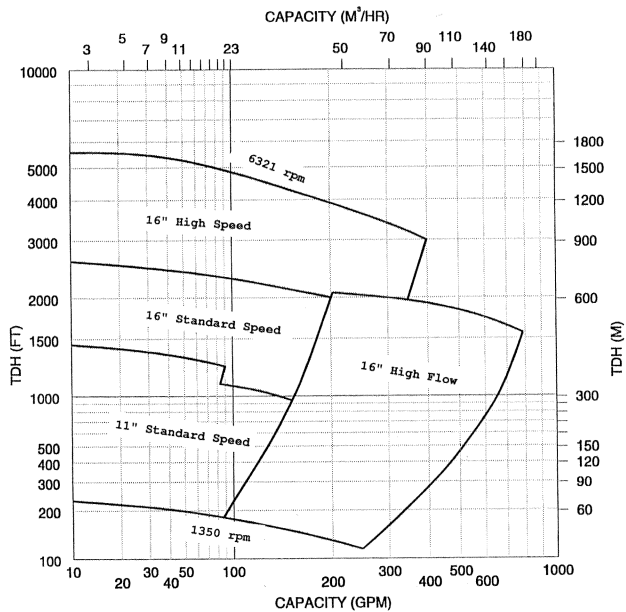


Figure 6. Family Performance Range.

KEY FEATURES OF THE PITOT PUMP

Safe Operation

A pitot pump can be safely operated at any point on its head curve, from full flow to shutoff. The pump can be operated at minimum flow indefinitely, with no wear or damage to the pump. The minimum flow is required only to keep the seal faces cooled and to keep the fluid in the pump from overheating and vaporizing. For continuous operation, minimum flow is in the order of 10 percent of the best efficiency point capacity.

Performance Adjustment

The capacity characteristics of the pump can be easily modified by changing only one component of the pump, the pitot tube, shown in Figure 1. This feature maximizes the standardization of pumps at a facility, thus reducing spare parts requirements. The pitot tube is manufactured in several inlet opening sizes and with either one or two inlets. These features determine the characteristics of the head/capacity curve for the pump.

Axial and Radial Load

The axial load on a pitot pump is relatively low and not affected by changes in pump discharge pressure, flowrate, or rpm. The radial load is relatively independent of flowrate and is primarily a function of pump speed and pitot tube size. These load characteristics allow the pump to be operated at any point on the capacity curve without harmful affects to bearing life.

Head Generation

The hydraulic mechanisms of the pitot pump allow it to produce very high heads at relatively low rotational speeds. It produces approximately 1.6 times the head as that of a similar sized conventional centrifugal pump operating at the same speed.

Dry Run

A pitot pump may incur only minor damage if run dry by loss of suction. A single seal pump will have damage at the seal faces.

Mechanical Seal Life

The mechanical seal is located on the suction side of the pump, shown in Figure 2. Thus, the seal operates on the low pressure side of the pump resulting in increased seal life.

APPLICATION INFORMATION

Materials of Construction

The pitot tube is the most critical and highly stressed component in the pump. Materials with both high strength and good corrosion resistance characteristics must be used in this application. Some of the materials being presently used are: 17-4PH stainless steel, 718 Inconel, and Hastelloy C.

The wetted pump components are available in a variety of materials. A few of the more common materials are: ductile iron, WCB steel, 316 and CA6NM stainless steel, CD4MCu duplex stainless steel, and Hastelloy C.

Wide Rangeability and Stable Low Flow Operation

Hydraulic stability and the pitot pump's ability to operate at minimum flow indefinitely makes it well suited for many applications. A pump with these operational characteristics works well in systems where the flow requirement continuously varies, or the pump must operate for extended periods of time against a closed valve. Some examples of these applications are:

- *Boiler Feed*—This application generally requires a pump to operate along its head curve from its maximum flowrate to its minimum flowrate, as the demands from the boiler change during the course of the day.
- *Central Cleaning Systems*—Typical applications use a multitude of spray guns or wands, either manual or automatic, to deliver high pressure water or other fluids upon demand. For these systems, it is essential that the pump be able to operate against a closed valve for extended periods of time, when there is no demand. The pump must also be able to meet the flow demand requirement for any variety of operation sequences. These same requirements are also found in gun drilling and hydraulic services.
- *NO_x Control*—This application uses a high pressure water spray to control the combustion temperature in a gas turbine to reduce the emissions of NO_x. This is a byproduct of the high temperature combustion of fossil fuels. The careful control of the temperature of combustion will reduce the amount of NO_x emissions during this process. A pump in this service must be able to operate at any point on its head curve as the demands on the gas turbine change.
- *Shower Service / Paper Industry*—In the making of paper, it is essential that the paper maker have complete control of the

condition of the screens or felt. High pressure showers are used to keep the felt and wires clean. If, during the manufacturing process, the paper maker decides that the felt is too wet, he may elect to turn off some or all of the showers, thus resulting in the pump operating at minimum bypass flow or at any point on its performance curve. This system demands a pump that can operate at any point on its performance curve.

Speed Control of Performance

The head (pressure) produced by the pump can be easily adjusted by changing the speed at which the pump is operated. These pumps are commonly driven either by electric motors controlled by a VFD, through a gearbox used as a speed increaser, or by V-belts and sheaves.

Deep Well Disposal

Deep well disposal is one of many services that require that a pump flowrate be maintained as the system pressure varies during a process. This can be easily accomplished with the combination of a VFD and a pitot pump. With its inherent hydraulic stability, the pitot pump offers the pump user a wide range of operating conditions.

Dry Run Capability

A pitot pump will not be damaged if run dry by loss of suction or when operated against a closed discharge valve, if it is equipped with a minimum continuous bypass flow orifice. The result from running a single seal pump dry will be the loss of its mechanical seal and so this will be the extent of the damage to the pitot pump.

Any service has the potential for the pump to be operated dry or to be operated against a closed discharge valve. A pump can lose suction for a variety of reasons. A valve may fail closed. A system unbalance or an obstruction in the suction line may cause a loss of $NPSH_A$. A pitot pump is unlikely to fail under these conditions, if operated for a short time. Operating against a closed discharge valve will not cause any harm to the pump, if a minimum bypass flow is maintained to cool the seal and the fluid in the pump. On the other hand, mechanical failure does not occur, because the loads on the pump bearings are relatively constant through the operating flow range of the pump. The pitot pump does not rely on the fluid being pumped for lubrication, except for the seal faces of a single seal.

High Pressure Processes

The ability of the pitot pump to produce high head at a relatively low rotational speed, coupled with a mechanical seal located on the suction side of the pump, and a high thrust capacity, make it particularly suitable for applications in high pressure processes, as pictured in Figure 7.

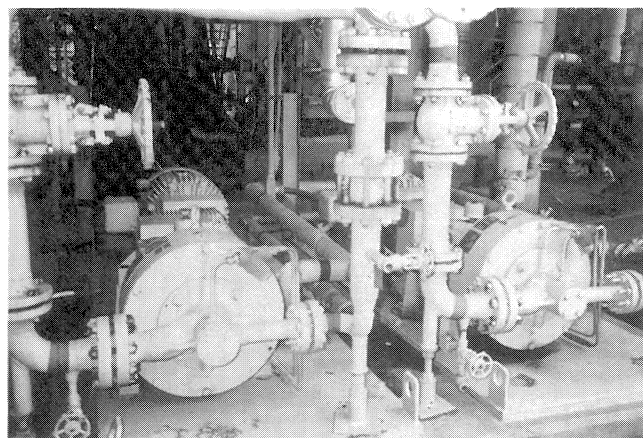


Figure 7. Pitot Pump Installation (High Pressure Process).

Petrochemical

There are a multitude of high pressure applications within this industry, one of these is the processing of light hydrocarbons. These services are characterized by their low specific gravity/high vapor pressure fluids and high process pressure requirements. The high vapor pressure of the fluids require that they be maintained at high process pressure to keep them from vaporizing, thus the pump will see high suction pressure. The fact that the seal in the pitot pump is on the suction side means that the seal does not need to be capable of handling pressures higher than the pump suction pressure. The high thrust capacity allows it to handle the high suction pressure.

There are several additional factors that must also be considered when applying a pitot pump in light hydrocarbon:

- The large volume of liquid contained inside the rotor chamber allows the pump to handle short term zero flow conditions without vaporizing the liquid in the pump.
- The geometry of this pump requires the liquid to pass through the mechanical seal and along the pitot tube extension before it gets to the eye of the impeller, shown in Figure 2. This allows the liquid to absorb heat prior to entering the impeller. A light hydrocarbon (or other liquid) that has a high vapor pressure increase with temperature change can experience a significant increase in vapor pressure in the pump suction area. This will give a corresponding decrease in the $NPSH$ available at the pump suction. This may lead to cavitation problems when the $NPSH_A$ is close to the $NPSH_R$.
- As a result of the above, a high minimum flowrate and/or special seal design or cooling may be required.
- As stated later, starting a pitot pump requires consideration of the motor sizing requirements. Often a light hydrocarbon (or other very low specific gravity liquid) may require a motor for starting the unit that is larger than what is needed to pump the low specific gravity liquid. Generally, this will not require more than a single step in the horsepower rating of the motor.

CHARACTERISTICS OF PITOT PUMP OPERATION

Startup

The startup characteristics of a rotating case pitot pump are different than most single stage centrifugal pumps and this must be understood by the user.

Starting torque and *start time* are two factors to be considered when designing a rotating case pitot pump installation. Unlike all other centrifugal pumps the rotating case pitot pump rotates both the impeller and the pump case. While this feature increases the head that the pump can produce, it also requires more mass in the rotating components of the pump and this affects the starting characteristics of the pump. The start time for this type pump will vary depending on the size of the pump, the starting torque of the motor, and the ratio of the gearbox or belt drive system. While the motor speed ramps up to full speed, the motor draws locked rotor amps. Therefore, the following considerations should be made when applying this type of pump:

- On low horsepower services, the start time should be calculated or the minimum horsepower recommendations given by the pump manufacturer should be followed.
- The pump should always be started against a closed discharge valve. This will minimize the horsepower load on the pump and the time required to start the pump.
- When a long start time is unavoidable, the electrical safety devices must be properly adjusted for this type of start.
- The use of a soft start device works extremely well with this type pump.

Vibration

As was discussed earlier, a rotating case pitot pump typically rotates a much larger mass than a geometrically similar sized centrifugal pump. As a result of this, the pump will typically have a vibration peak at its operating speed. This peak is a result of an out of balance condition of the pump. The ratio of the mass of the rotating components of the pump to the total weight of the pump is much greater in a pitot pump than in a conventional centrifugal pump. Because of this fact, a precise dynamic balance of the rotating assembly is required. The magnitude of the vibration is related to the amount that the rotating components are out of balance. Good results have been obtained by balancing the entire rotating assembly to be below both API and ISO G 0.4 levels. The pump is balanced with the rotor assembly filled with a liquid similar in specific gravity to the liquid that the pump will see in operation.

SOLIDS HANDLING

A pitot pump can handle some solids in the fluid. However, careful consideration must be taken in the application of this type of pump when solids are present. The two major factors that must be considered are balance of the rotating assembly and abrasive wear of the components.

Balance

A potential problem may occur when a pump that has been pumping solids in suspension is stopped with fluid still in the pump. As long as the solids remain in suspension, there will be no increased vibration on startup. But if the solids come out of suspension and settle on the bottom of the rotating casing, the pump will be momentarily out of balance and will vibrate at an increased level for a short time. As soon as the solids are mixed back into suspension by the hydraulic action in the pump, the balance will return and the pump vibration will decrease to normal levels. The most serious problem will occur if the solids adhere to the inside wall of the rotating casing when the pump shuts down and are not mixed back into suspension on startup. If this occurs, the pump will have to be stopped and the rotating casing will need to be flushed out. In many solids applications of pitot pumps, this problem is avoided by setting up the system so that clean fluid is run through the pump for a short period of time, just prior to shutdown.

Abrasive Wear

The problem of abrasive wear in a pitot pump becomes apparent when considering the relative speed of the rotating fluid in relation to the pitot tube. As an example, a pitot pump operating at 5000 rpm having a pitot tube radius of 7.7 in has a relative velocity between the fluid and the pitot tube of slightly over 335 ft/sec.

Abrasive wear is a function of a number of factors, including the shape, specific gravity, size, and hardness of the particles present. Each situation must be evaluated based on these factors along with past experience. In general, when solids are present, a 100 mesh screen on the suction line of the pump is recommended when practical for the application. A number of materials have been used in the various components that can reduce the abrasive wear to acceptable levels. However, given the very high velocities present, the pitot pump must be applied with a great deal of caution when solids are present.

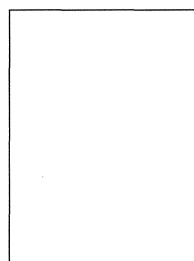
CONCLUSION

The pitot pump is a centrifugal type of pump in the low specific speed classification. It is suitable for applications requiring high heads and relatively low flows. The pump has good efficiency for its specific speed range 60 to 360 [3] and is suitable for operation

over a wide flow range. The pitot pump has been applied to heads over 5000 feet and to flows up to 750 gpm. The maintenance requirements are usually low due to the simplicity of the design [5].

The hydraulic design principles that characterize the pitot pump, the key features, and the strengths and weaknesses of the pump have been discussed in the text herein. It is hoped that this information will provide the pump user with a better understanding of how the head and flow are generated and how this type of pump can best be evaluated for, and applied to, high head low flow applications.

PROGRESSING CAVITY PUMP



Kent Whitmire is currently the Quality Assurance Manager for Roper Pump Company, in Commerce, Georgia, where he has been employed for the past 19 years. Prior to his latest assignment, Mr. Whitmire was a Project Engineer for Roper, responsible for product design involving progressing cavity pumps along with gear and air operated diaphragm pumps.

Mr. Whitmire has written several articles on pump selection and application that have appeared in various trade publications. With a background in computer science, he was instrumental in developing a computer-based selection program for progressing cavity pumps. He also has several years design experience with large centrifugal and axial flow pumps.

ABSTRACT

While not the most common of pump types, the progressing cavity pump is certainly one of the most versatile and interesting. Progressing cavity pumps transfer common fluids such as water and oil, and they also handle many difficult to pump fluids such as highly viscous or abrasive fluids very successfully. Progressing cavity pumps have found many applications in industrial processes, construction, waste water treatment, oil field service, pulp and paper processes, and food processing, to name a few. Operation and maintenance are simple and straightforward with no complicated valves or timing mechanisms to adjust or maintain. Understanding the capabilities and limitations of progressing cavity pumps can help in selecting the proper applications for their use.

INTRODUCTION

Background

Developed in the early part of this century, progressing cavity pumps have gone from an interesting oddity to a pumping workhorse handling many of today's more difficult to pump fluids. The progressing cavity principle was originally invented by a Frenchman, René Moineau, in 1929, and first introduced into North America in 1936 [6]. Progressing cavity pumps developed from this original Moineau concept are a unique type of positive displacement pump that can be applied over a wide variety of applications.

While the objective is to introduce the pump user to the basic design principles, applications, operation, and maintenance of progressing cavity pumps, the focus on the theoretical will be limited and the emphasis will be on more practical, everyday issues. However, to get started, it must be understood how the pump operates and that requires just a little theory.

Additional information may be found by referring to general references [7, 8, and 9].

Principle of Operation

The progressing cavity pump is a helical gear pump belonging to the rotary, positive displacement pump family. The pump consists 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 series of cavities 180 degrees apart, which progress along the axis of the assembly as the rotor is rotated. As one cavity is decreasing in volume, the opposing cavity is increasing at exactly the same rate. Therefore, the sum of the two discharges is a constant volume.

The cross section of the stator is two semicircles of diameter D separated by a rectangle with sides $4e$ and D (Figure 8). The cross section of the rotor is a circle of diameter D that is offset from the centerline by the eccentricity e (Figure 9). The lead of thread of the stator is P and is twice the lead of the rotor (Figure 10).

The dimensions of the cavity formed when the rotor and stator are meshed together is equal to the void of the cross section

$$\left(\frac{\pi D^2}{4} + 4eD\right) - \left(\frac{\pi D^2}{4}\right) \tag{10}$$

or $4eD$. This cross sectional void times the stator lead determines the cavity ($4eDP$) that is displaced upon each revolution of the rotor and can be expressed in various ways, one most popular being gallons per 100 revolutions.

This basically describes what is known as a 1:2 design, i.e., the rotor having one thread or lobe and the stator having two. Variations exist on this such as 2:3, 3:4, 4:5, etc. designs. The only requirement is for the stator to have one more thread or lobe than the rotor. Discussed only is the 1:2 design, as this is the most popular and commercially available design for progressing cavity pumps available today.

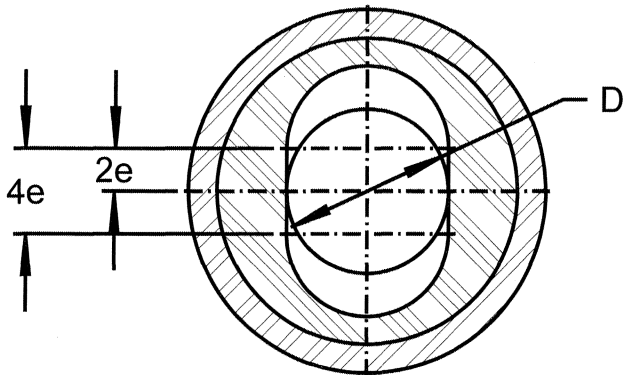


Figure 8. Stator Cross Section.

PUMP PERFORMANCE CONSIDERATIONS

Typical Pump Performance

Being a positive displacement pump, 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 deliver the same volume of fluid regardless of the pressure (resistance) that must be overcome in the discharge line. With a fluid such as water (1cP viscosity) and with zero discharge and suction pressures, the displacement, or quantity of fluid delivered, of the pump is only dependent on the revolutions per minute.

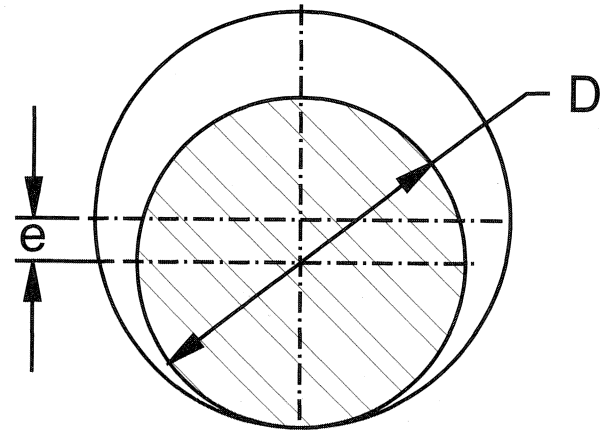


Figure 9. Rotor Cross Section.

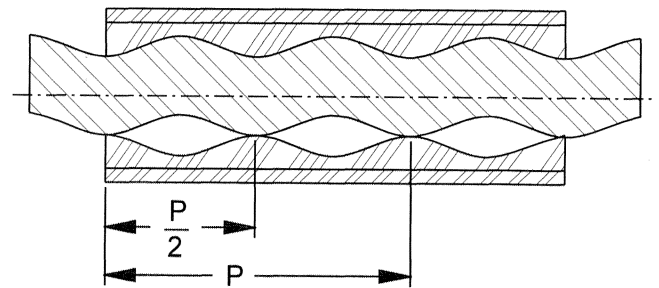


Figure 10. Axial Element Section.

As the pressure increases, a small amount of the fluid displaced slips back through the elements to the suction side. This slip is the fluid that leaks across the sealing lines of the cavities from the higher pressure discharge back to the lower pressure suction side of the elements. Slip is measured in units of flow such as gallons per minute (gpm). The amount of slip or leakage is greater as the discharge pressure increases. To minimize the amount of slip at high pressures, more cavities are added in series by lengthening the rotor and stator. This is called staging. Increasing the number of stages decreases the amount of slip experienced by the pump by distributing the pressure differential over a greater number of pump stages. Staging also is used to obtain higher differential pressures for a given flow design.

Catalog performance curves for progressing cavity pumps are usually for water and show the anticipated delivery of the pump vs differential pressures (discharge to suction) and speeds. Typical progressing cavity pump curves are shown in Figures 11 and 12. For thicker, more viscous fluids, the slip is significantly reduced and can be approximated by dividing the slip on water by the slip index value shown in Figure 13. The slip index is the ratio of slip on a viscous fluid divided by the slip on water at 70°F.

To demonstrate the effects of viscosity on slip, assume the pump curve shown in Figure 11 is being used to pump a fluid with a viscosity of 10,000 cP at a differential pressure of 150 psi. From the curve in Figure 11, the slip on water at 150 psi is nine gpm. This slip value is obtained by taking the flow at zero psi and then subtracting the flow at 150 psi. As the slip is basically independent of speed, this can be done at any speed as long as the speed is the same for both pressures in the formula. Looking at the slip index curve in Figure 13, the slip index for 10,000 cP is about 6.2. Therefore, the slip on the pumped fluid will be approximately:

$$\frac{\text{WaterSlip}}{\text{SlipIndex}} \tag{11-A}$$

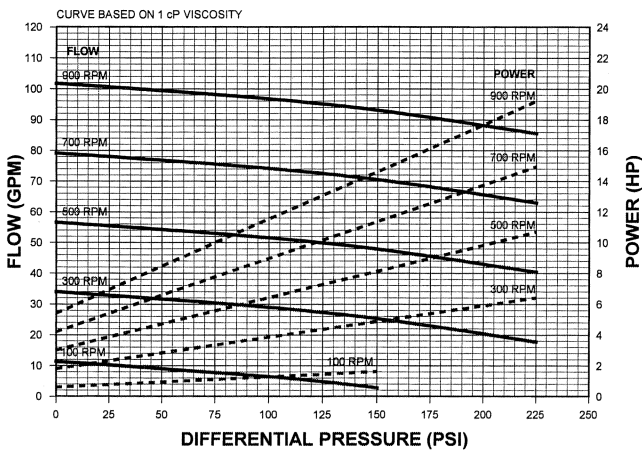


Figure 11. Typical Horsepower Curve.

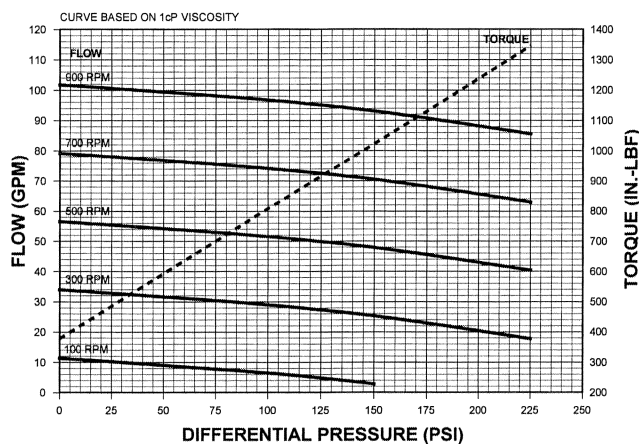


Figure 12. Typical Torque Curve.

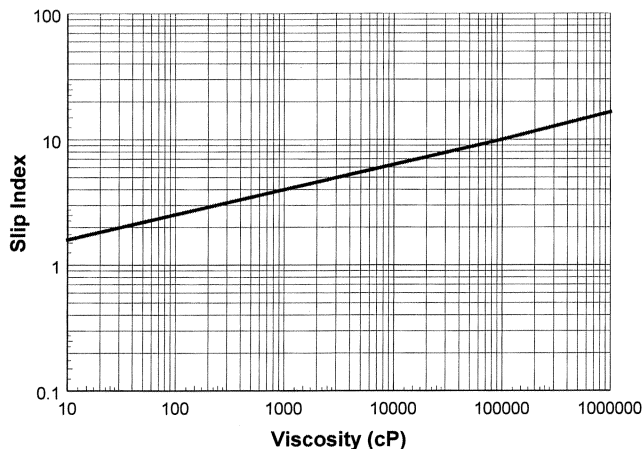


Figure 13. Slip Index Curve.

or:

$$\frac{9}{6.2} \quad (11-B)$$

≈ 1.45 gpm at this pressure.

Power Requirements

The power requirements for progressing cavity pumps is expressed differently by different manufacturers. Some

manufacturers express the power requirements simply as horsepower and others express them as torque. Since the progressing cavity pump is considered a constant torque device when operating at a given differential pressure, using torque to determine power requirements is a simplified and accurate method of determining the power requirements, and will allow better selection of drive components especially hydraulic, and electrical variable speed type drives.

The stator is usually made from an elastomeric material, although some manufacturers offer a metallic (or rigid) stator for specialized applications. This elastomeric stator allows the pump elements to have a compression fit and also offers a good abrasion resistant surface for handling particles in suspension. This compression fit, however, does cause a resistance to turning (torque) that is dependent on the element geometry and is shown on the performance curve as the initial torque required at zero psi. There is also a starting torque that must be overcome. As a rule of thumb, this value is roughly four times the initial torque. Horsepower can be derived from the torque value by the formula:

$$Hp = \frac{TN}{63025} \quad (12)$$

where *T* is torque in inch pounds and *N* is pump speed in rpm.

Progressing cavity pumps are rarely applied on fluids with a viscosity of one cP, and there is an added torque that is dependent on the viscosity of the fluid and the geometry of the elements. On slurries, the added torque required is dependent upon the particle size and concentration of solids. Most progressing cavity pump manufacturers include power adder information for viscosity and slurries with their standard technical literature.

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 along with the distance from the centerline. Internal velocity information is available from manufacturers and is important in ensuring an adequate supply of fluid to the elements. The section on inlet conditions will detail this aspect a bit more.

The shear rate of the pump can be determined from *D*, *e*, and *P*, and is available from pump manufacturers for their particular models of pumps. Shear rate is defined as the velocity difference between two layers of fluid divided by the distance between the layers, measured in inverse seconds (*s*⁻¹ or 1/*s*). In general, progressing cavity pumps have a very low shear rate. This is of particular importance when pumping shear sensitive fluids such as latex and some polymers that decompose if sheared too much during handling.

The shear rate of the pump can also be used to determine the apparent viscosity (the shear stress divided by the shear rate, measured in units of viscosity) of a non-Newtonian fluid inside the pump. This is useful in determining the power requirements of the pump as explained below.

Fluid Viscosity

The viscosity of a fluid is the property of a fluid that resists flow and is the 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 centipoise (cP). Sometimes, the fluid viscosity is expressed in centistokes, which is the kinematic 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 curves in Figure 14 plot the viscosity in centipoise against the shear rate in inverse seconds 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, tar, rayon, printing inks, varnishes, resins, vegetable oils, and shortenings.

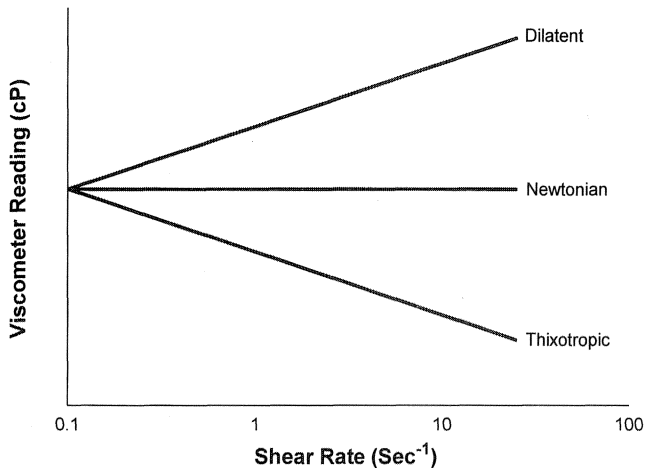


Figure 14. Comparison of Fluid Viscosities.

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 can then be predicted at various speeds or shear rates.

Dilatent fluids are rather uncommon and are mostly high concentrated slurries. A dilatent 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 that cannot be classified in the preceding categories and can be handled very nicely with the progressing cavity pump. These materials, such as filter cake, dewatered slurries or sludges, and paper stock are semidry 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 by a pump where the standard suction housing is replaced with a flanged hopper and an auger is attached to the connecting rod to assist movement of the material into the pumping elements.

Usually the viscosity of a fluid changes when there are changes in fluid temperature. Lower temperatures usually increase a fluid's viscosity, while higher temperatures reduce them. The temperature of the fluid must be considered in determining the viscosity of any fluid.

Inlet Conditions

The key to obtaining good performance from a progressing cavity pump, as with all other positive displacement pumps, lies in a complete understanding and control of inlet conditions and the closely related parameters of speed and viscosity. To ensure quiet, efficient operation, it is necessary to completely fill the cavities with fluid as they open to the inlet. This becomes more difficult as viscosity, speed, or suction lift increases. Basically, it can be said that if the fluid can be properly introduced into the pumping elements, the pump will perform satisfactorily. The problem lies in getting it in!

It must be remembered that a pump does not pull or lift liquid into itself. Some external force must be present to push the fluid

into the pumping elements initially. Normally, atmospheric pressure is the only force present, although there are some applications where a positive inlet pressure is available.

Naturally the more viscous the fluid, the greater the resistance to flow, therefore, the slower the rate of filling the moving cavities in the inlet. Conversely, low viscosity fluids will flow quite rapidly and will quickly fill the cavities. It is obvious that if the cavities are moving too fast, the filling will be incomplete and a reduction in output will result. The rate of fluid flow into the cavities should always be greater than the rate of cavity travel to obtain complete filling. Most progressing cavity pump manufacturers provide recommended maximum speeds for their pumps when handling viscous fluids that minimize the effects of incomplete filling.

The net positive suction head (NPSH) calculations are routinely used in centrifugal or high velocity pump applications. In many positive displacement pump applications where the pump velocities are usually low, the calculation of NPSH has little significance. There are applications, however, where this calculation becomes very important. These are suction lift, vacuum pot applications, and applications where the fluid vapor pressure is high.

The net positive suction head available ($NPSH_A$) is the head available at the inlet of the pump and is the atmospheric pressure available minus the fluid vapor pressure, the lift, and suction line losses. The net positive suction head required ($NPSH_R$) is a function of the pump design and pump speed. The $NPSH_R$ for a specific pump is available from the pump manufacturer, usually included with their standard performance information.

The following examples illustrate NPSH calculations:

Suction Lift

Example: A particular progressing cavity pump operating at 900 rpm is required to lift 70°F water 10 ft. vertically through a three in line. The vapor pressure of the fluid is 0.363 psia (0.84 ft) and the suction line losses are 0.01 ft. The pump is physically located at, or near, sea level. From the pump manufacturer's literature, the pump has an $NPSH_R$ of 6.9 ft.

Atmospheric pressure	33.90 ft
Lift	-10.00 ft
Line losses	-0.01 ft
Vapor pressure	-0.84 ft
Total $NPSH_A$	23.05 ft
Total $NPSH_R$	6.90 ft

From this example, there will be 16.15 ft of head over the required amount and this will be acceptable.

High Vapor Pressure Fluid

Another application where NPSH becomes important is when the fluid's vapor pressure is high. Vapor pressure is a key property of fluids that must always be recognized and considered. This is particularly true of volatile petroleum products such as gasoline, for example, which has a very high vapor pressure. The vapor pressure of a fluid is the absolute pressure at which the fluid will change to vapor (boil) at a given temperature. A common example is the vapor pressure of water at 212°F, which is 14.7 psia. For petroleum products, the Reid vapor pressure (absolute) is usually the only information available. This is vapor pressure as determined by the ASTM D323 procedure. Vapor pressure tables

for water can usually be found in most hydraulic books [10]. Vapor pressure estimates for other fluids can be made if the boiling point is known (i.e., a fluid will boil at atmospheric conditions when its vapor pressure reaches 14.7 psia). The absolute pressure must not be allowed to drop below the vapor pressure of the fluid. This will prevent boiling that will cause cavitation.

In the previous example, 70°F water was used that had a vapor pressure of 0.363 psia. If the temperature of the water was 190°F, the vapor pressure would be 9.34 psia or 21.6 ft, which would exceed the 16.15 ft difference between $NPSH_A$ and $NPSH_R$ and the fluid would vaporize or boil. The calculations would now be as follows:

Atmospheric pressure	33.90 ft
Lift	-10.00 ft
Line losses	-0.01 ft
Vapor pressure	-21.60 ft
Total $NPSH_A$	2.29 ft
Total $NPSH_R$	6.90 ft

In this example, there is not enough $NPSH_A$ to operate the pump. To overcome this situation, the amount of lift would have to be shortened or possibly a different sized pump, operating at a reduced speed, could be used.

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 $NPSH_A$ to the pump.

Example: A particular progressing cavity pump is pumping water out of a vessel that is under 20 in of mercury vacuum that corresponds to 11.20 ft absolute pressure. There is 10 ft of three in horizontal line connecting the suction of the pump to the vessel and the water is 70°F. As in the first example, the vapor pressure of the fluid is 0.363 psia (0.84 ft) and the suction line losses are 0.01 ft. From the pump manufacturer's literature, the pump has an $NPSH_R$ of 6.9 ft.

Vessel pressure	11.20 ft
Lift	-0.00 ft
Line losses	-0.01 ft
Vapor pressure	-0.84 ft
Total $NPSH_A$	10.35 ft
Total $NPSH_R$	6.90 ft

From this example, there will be 3.45 ft of head over the required amount and this will be acceptable.

On vacuum pot and suction lift applications, it is necessary to fill, or partially fill, the suction housing and 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 side that is normally considered the suction side of a progressing cavity pump, better operation can be achieved if the pump is operated in reverse. By doing that, the normal discharge port is used as the suction port. This would now put the packing or

seal under pressure from the discharge and caution should be used not to overpressure the suction housing. Progressing cavity pump manufacturers can provide helpful information on operating their pumps in reverse and should definitely be consulted if high discharge pressures are anticipated.

To allow pump manufacturers to offer the most economical selection and also assure a quiet installation, accurate inlet conditions should be clearly stated. Specifying a higher suction lift than actually exists may result in the selection of a pump at a lower speed than necessary. This means not only a larger and more expensive pump, but a costlier driver also. If the suction lift is higher than stated, the outcome could be a very noisy pump installation coupled with higher maintenance costs.

GUIDANCE FOR PROPER SELECTION AND INSTALLATION

Abrasion

The progressing cavity pump is one of the best pumps available 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. Most progressing cavity pump manufacturers limit these items by publishing reduced maximum speeds and pressures for pumps handling abrasive materials.

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 closer look into what causes abrasion may be helpful in determining its classification. The components of abrasivity are the particle, the carrier fluid, and the velocity.

- The 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 the wear.
- Density—heavier particles will not pass through the pump as easily as lighter ones.

Relating the material hardness to some common materials on a one to 15 scale, the following list can be used as a guide:

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

- The carrier fluid
- Corrosivity—surfaces attacked by corrosion will set up a corrosion-erosion effect.
- Viscosity—a high viscosity fluid will tend to keep particles in suspension and not be as abrasive.
- The velocity

A low fluid velocity, or pump speed, will minimize abrasive effects. For a heavy abrasive fluid, it is recommended to keep the average velocity in the elements between three to five ft/sec. A medium abrasive fluid should be limited to six to 10 ft/sec and a light abrasive should be limited to 10 to 15 ft/sec. These velocity limits are usually listed as pump rpm limits for various size pumps in conjunction with the abrasive nature of the fluid.

As mentioned previously, determining the degree of abrasion is mostly judgmental, however, the examples below will provide a simple rule of thumb guide to abrasive classification:

- No abrasives—clear water, gasoline, fuel oil, lubricating oil
- Light abrasives—dirty water containing silt or small amounts of sand or earth
- Medium abrasives—clay slurries, potters glazes, porcelain enamel, sludge
- Heavy abrasives—plaster, grout, emery dust, mill scale, lapping compounds, roof gypsum

As with the shear rate, the maximum particle size that can be handled by a progressing cavity pump can be determined from the element design and is available from pump manufacturers for their particular models of pumps. Depending on pump size, the maximum particle size that can be successfully handled without pump damage ranges up to about two inches.

Temperature Effects and Limits

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 times greater than that of the metal rotor that is usually steel or stainless 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 undersized rotor that compensates for the difference in size.

Second, the life of the elastomer is greatly affected by heat. The following limits are from one manufacturer for elastomers that 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 pressure rating, the operating limit can be exceeded slightly, but cannot exceed the static rating.

Table 3. Stator Temperature Ratings.

Temperature Rating (°F)		
Material	Stator Rating	Max. (Static) Rating
Nitrile	180	250
Natural rubber	185	225
EPDM	300	350
Fluoroelastomer	300	400

Mounting and Vibration

The progressing cavity pump is inherently an unbalanced machine due to the eccentric rotation of the rotor. The vibration

that 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 with typical limiting speeds reaching 1200 rpm for small pumps (0.02 gallon per revolution and lower) decreasing to limits of 300 rpm for large pumps (3.0 gallons per revolution and larger).

The magnitude of the induced vibration is of a low frequency and a relatively high amplitude. This will not produce offensive noise, however, it should be a factor in mounting. It is normally recommended that progressing cavity pumps be mounted on structural steel baseplates securely lagged down to concrete foundations. These baseplates should be provided with means to grout them into place on the foundation for rigidity and dampening. In all cases, the manufacturer's instructions and recommendations on mounting should be consulted to ensure an adequate installation.

The Drive Frame

The progressing cavity elements are normally adapted to a drive end that will provide an acceptable life span when properly applied and maintained. As with ANSI centrifugal pumps, the drive shaft is supported in a separate bearing housing, usually cast iron, by antifriction bearings, either deep groove ball or tapered roller bearings. These bearings are normally sealed from contamination by conventional lip type seals. The bearings in a progressing cavity pump are almost always lubricated by grease. Due to the relatively low speed of progressing cavity pump applications, this is quite acceptable.

Because of the rotor's eccentric motion, there must be some means of transferring the torque from the concentric rotating shaft to the rotor. The normal means of this is a connecting rod and universal joints. This combination allows the rotor to freely move through the eccentric circle and carries the torque and thrust created by the differential pressure on the elements. Several types of universal joints are used, with the two most popular being a simple pin and rod joint and the other a more elaborate crowned gear joint. Some manufacturers even offer a flexible shaft, eliminating the universal joints entirely.

The pin and rod pumps are considered the more traditional type pumps (Figure 15). Pins are installed at both ends of the connecting rod to secure it to the shaft and rotor (Figure 16). In operation, the connecting rod rocks on the pins from the eccentric motion of the rotor. Rubber O-rings, or a proprietary seal, keeps pumpage from the joint and retains lubricant. The pins and connecting rod are made from heat treated alloy steels to provide a long service life. When a stainless material is required for corrosion resistance in the pumping application, the pins and rod cannot normally be hardened. Most manufacturers can increase the drive frame size of pumps requiring these corrosion resistant materials to provide an acceptable service life. If differential pressures are relatively low (less than 75 psi), then this increase in frame size may not be necessary.

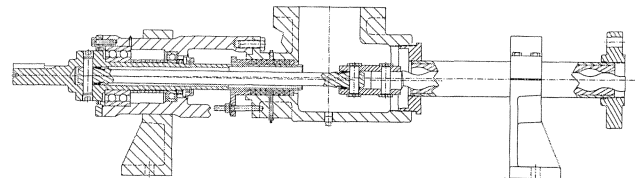


Figure 15. Typical Pin Joint Pump.

Unfortunately, the rocking between the pin and rod is a wearing action that eventually renders the pin unusable. To overcome this and prolong service life, progressing cavity pump manufacturers also employ a crowned gear type joint that has approximately five times the load carrying capacity of the conventional rod and pins (Figure 17).

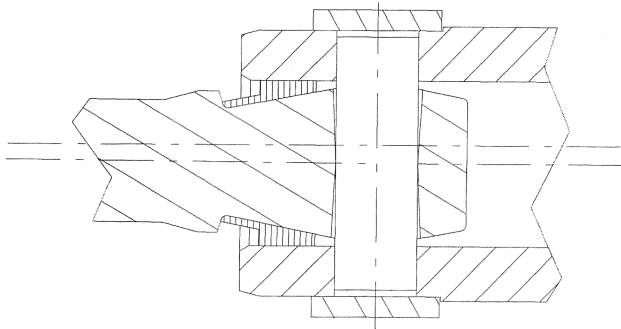


Figure 16. Pin Joint Detail.

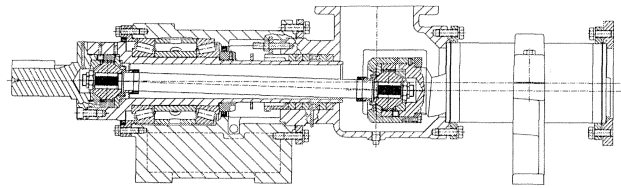


Figure 17. Typical Gear Joint Pump.

The gear joint is basically a modified ball and socket joint with gear teeth on the ball mating with a similar tooth profile in the socket (Figure 18). These joints are usually designed for the larger size pumps (≈ 0.1 gallon per revolution and larger), where the initial expense is outweighed by the maintenance and replacement costs of the rod and pin type.

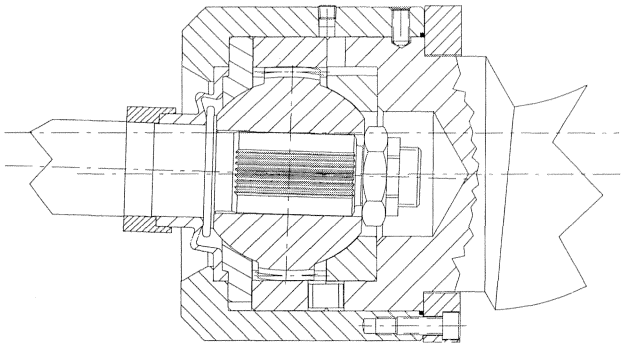


Figure 18. Gear Joint Detail.

PUMP APPLICATIONS

The number of applications for progressing cavity pumps are as many and as varied as there are progressing cavity pumps manufactured. As mentioned previously, progressing cavity pumps have found many applications in industrial processes, construction, waste water treatment, oil field service, pulp and paper processes and food processing, just to name a few. Flowrates available range from fractions of gpm to around 2000 gpm with differential pressures reaching upward to 2000 psi.

As with every pump type, there are certain advantages and disadvantages characteristic of the progressing cavity design. These should be recognized in selecting the best pump for a particular application [11].

Advantages

- Wide range of flows and pressures
- Wide range of liquids and viscosities
- Low internal velocities

- Self-priming, with good suction characteristics
- High tolerance for entrained air and gases
- Minimum churning or foaming
- Pulsation-free flow and quiet operation
- Rugged design—easy to install and maintain
- High tolerance to contamination and abrasion

Disadvantages

- High pressure capability requires long length of pumping elements
- Fluid incompatibility with elastomers can cause problems.
- Not suited to high speed operation—requires gear reducer or belt reduction
- Cannot run dry

In oil field service, progressing cavity pumps are used in many applications in the production of oil, from polymer injection to crude oil transfer. Recently, China's Daqing Petroleum Administrative Bureau installed a battery of large progressing cavity pumps for polymer transfer in the Daqing oil field, located about 750 miles northeast of Beijing. These pumps are used to transfer 250 gpm each of 3000 to 5000 cP polymer at discharge pressures of 300 psi. The pumps selected were in the 1.15 gallon per revolution size and are operating at 250 rpm with 130 horsepower drivers.

Another interesting oil field application is the progressing cavity "down hole" pump. This is a progressing cavity pump suspended vertically below the surface into an oil well to pump fluid to the surface. Driven by a surface mounted driver, the pumps are suspended to depths of up to 4000 ft. Able to pump the entrained gases and sands, progressing cavity down hole pumps are an economical alternative to conventional rod pumps in many wells with difficult to pump fluids. They are proving to be particularly well suited to the heavy crudes in the oil fields of Canada.

Progressing cavity pumps are also useful in mining operations worldwide. In Australia, the gold mines use progressing cavity pumps for dewatering service in the mines. Located about 700 ft underground, these pumps deliver 560 gpm at a discharge pressure of 330 psi to keep the mines from flooding. The water contains about five percent earth solids with an average particle size of 0.040 in. The units selected for this service were in the 1.75 gallon per revolution size and are operating at 340 rpm using 175 hp drivers.

The construction industry is a large user of progressing cavity pumps. In Salt Lake City, Utah, spackling (wall joint compound) with an apparent viscosity of 30,000 cP is being pumped at one and one-half to two gpm against discharge pressures of 80 psi. The units selected for this service were in the 0.02 gallon per revolution size and are operating at 140 rpm, using three hp drivers.

Another construction application involves grout injection. In Southern California, portable units are made to inject grout under bridges to shore them up without closing them to traffic. These pumps deliver 35 gpm against 250 to 350 psi discharge pressures. The units selected for this application were in the 0.22 to 0.28 gallon per revolution size and operate at 175 rpm using 25 hp drivers. There are many manufacturers of grout equipment using progressing cavity pumps and this represents a large segment of their usage.

Waste water treatment is another area where progressing cavity pumps are widely used. Their use ranges from scum and sludge handling to polymer injection. In one Illinois correctional facility, units are installed in the 0.28 gallon per revolution size operating at 400 rpm using 10 hp drivers. These units deliver 100 gpm against a discharge pressure of 30 psi.

The pulp and paper industry uses a wide variety of progressing cavity pumps in the making of paper. Coating kitchen applications are a prime example of this. Both an Ohio and Mississippi paper plant use progressing cavity pumps to pump paper coatings in their coating kitchens. The units are 1.15 gallon per revolution size and operate at 235 rpm using 75 hp drivers. These units deliver 200 gpm against a 170 psi discharge pressure.

In food processing, many chicken processing plants use progressing cavity pumps to pump chicken parts (heads, feet, viscera, etc.) as waste. One Arkansas plant uses several units sized for 0.44 gallon per revolution operating at 200 rpm to deliver 80 gpm against a 100 psi discharge pressure. These units are driven by a 20 hp driver. These pumps use a specially enlarged suction housing and auger feed to ensure the material gets to the pumping elements.

Not only limited to handling waste in food processing plants, progressing cavity pumps are also manufactured in food grade designs to pump foods and pharmaceuticals. Pumps used for these services offer quick disassembly for cleaning and are made from food grade materials. Some manufacturers offer designs to meet 3-A sanitary standards. Applications for these units range from pumping toothpaste to maraschino cherries, the latter with less than a one percent damage rate!

TROUBLESHOOTING

The progressing cavity pump is usually forgiving in most installations, however, there are times when troubles occur. Accurate identification of the problem is usually 99 percent of the cure. The following is to assist in identifying the problem and offer some suggestions for rectifying the situation.

No Liquid Delivered

- Pump rotating in wrong direction
- Inlet lift too high. Check this with gauge at pump inlet.
- Clogged inlet line
- Inlet pipe not submerged
- Air leaks in inlet line
- Faulty pressure relief device in system
- Pump worn

Rapid Wear

- Excessive discharge pressure
- Pump runs dry
- Incompatibility of liquid and pump materials
- Pipe strain on pump
- Speed too high for abrasives present in liquid

Excessive Noise

- Starved pump
- Air leaks in inlet line
- Air or gases in liquid
- Pump speed too high
- Improper mounting. Check alignment thoroughly.

Pump Takes Too Much Power

- Speed too high
- Liquid more viscous than previously anticipated
- Operating pressure higher than specified. Check this with gauge at pump discharge.

- Discharge line obstructed
- Mechanical defect such as bent shaft
- Packing too tight
- Pipe strain on pump
- Incompatibility of liquid and pump material causing stator swell
- Pressure relief device in system not operating properly

Rotors

Other than normal wear on a rotor, there is little that can cause a problem. Occasionally a problem will arise with the chrome plate lifting. This is usually caused by corrosion of the base metal of the rotor. In these situations, a stainless steel rotor is usually the solution. On fluids of pH three or less, the fluid will remove the chrome and a nonplated rotor should be used.

Stators

Three modes of stator failure are bond failure, chemical attack, and high heat.

- *Bond failure*—In the stator manufacturing process, a special curing type adhesive is applied to a grit blasted tube prior to injection of the elastomer. The bond to the tube is stronger than the tensile strength of the elastomer and is sufficient to hold the rubber in place during operation. On fluids with a pH of 10 or greater, there exists the potential of chemical attack on the bond and it will eventually break loose from the inside of the tube. Very high heat (as from an acetylene torch) or cryogenic temperatures will also break the bond.

- *Chemical attack*—The effect of chemical attack is usually a swelling and softening of the elastomer. The pump will operate fine at first and then gradually increase in power requirements as swelling occurs until the elastomer fails. Analysis of this problem should be made immediately because after “drying out,” the elastomer will return to its original state and appear fine. When there is doubt about material compatibility, small test slugs of elastomer can be immersed in the fluid to study the effects before even purchasing the pump. These test slugs are available from many progressing cavity pump manufacturers.

- *High heat*—The enemy of rubber is heat and it will have a different effect on different materials. Nitrile and fluoroelastomer will harden to a hardness similar to steel and will have a glossy and crazed appearance. EPDM and natural rubber will have a tendency to get gummy and have a melted appearance. The source of excessive heat can come from several sources. The fluid temperature can go over the allowable limit or the heat can be internally generated. This internal heat can come from overrating the pump, pumping against a closed discharge, or running dry. In the first two cases, the stator will usually show signs of coming apart in chunks.

CONCLUSION

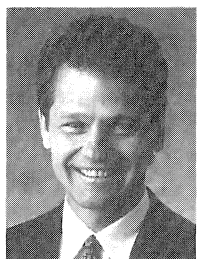
Progressing cavity pumps are widely used in handling many difficult to pump materials. In many cases, a progressing cavity pump is the only pump that can suitably handle these materials. Properly understanding how they work, and how to specify the operating characteristics, can help the user to assist progressing cavity pump manufacturers in selecting the most economical unit to solve many difficult pumping problems. When properly selected, progressing cavity pumps are quiet in operation and forgiving of many operational problems that arise in the system. At first an oddity, now progressing cavity pumps are gaining a foothold in the main stream of pumping applications.

AIR OPERATED DOUBLE DIAPHRAGM PUMPS



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Mr. Taylor received his education at Ohio University and Ohio State University and is a coinventor of the Elimi-matic™, an anti-icing, antistalling air valve system patent.



William F. Versaw is the Founder and President of Versa-Matic Pump Company, headquartered in Export, Pennsylvania. Mr. Versaw has patented several new designs that minimize downtime in manufacturing operations. Currently, Mr. Versaw directs Versa-Matic's product development activities and manages global sales and marketing programs.

Mr. Versaw is an expert in the area of fluid transfer and the application of air operated double diaphragm (AODD) pumps. He speaks at industry conferences and seminars, and has published a white paper on new technology in the AODD pump market.

ABSTRACT

The air operated double diaphragm pump [12] is a versatile type of reciprocating displacement pump. Discussed are the application uses of this type of pump; the performance range; pump selection; materials of construction; some examples of applications; recommended installation; limitations and advantages.

INTRODUCTION

The air operated double diaphragm pump (AODD) is a pump in which two flexible diaphragms are mounted vertically, sealed around the perimeter, connected with a common shaft, and pumps fluids through a single ported manifold by reciprocating movement. Compressed air is valved alternately behind one diaphragm, then the other.

This type of pump has been used for many years in the mining and construction industries. However, its major introduction and subsequent contribution to industrial users started in the 1960s. The market that it has adapted to is constantly expanding worldwide. Primarily, the AODD is powered by compressed air. Compressed nitrogen has been successfully used. However, the cost factor has favored compressed air as the accepted power source.

These pumps are especially suited for liquid and slurry transfer of anything that flows ranging from water to 90 percent solids suspended in a fluid and highly viscous materials. They are also capable of handling a very wide range of corrosive and abrasive liquids, because the pumped liquid does not come into contact with most of the working parts of the pump. Because compressed air is inherently explosion proof and the AODD uses no electrical or heat generating components to spark to cause product flashing, they are suited for volatile fluids transfer.

The AODD can handle a wider range of pumped fluids than any other type of pump.

APPLICATIONS

The AODD pumps are industrially well suited because of their low initial cost, simple installation, exceptional tolerance to variation in operating conditions and their ability to handle "problem media." Because there are no mechanical seals, they have dry self-priming capabilities and inherently variable flow, by either controlling the air inlet flow or controlling the discharge flow of the pump. They offer an easy-to-use alternative to difficult pumping applications.

There are countless design configurations, materials, and sizes. The size of a pump is designated by connecting pipe size. As an example, a two in pump would have connections at the suction (inlet) and discharge (outlet) ports to accept standard two in pipe. Presently, the size range is 1/4 in to four inches. The flow characteristics would range over 260 gpm.

Because the pump can be adjusted to infinitely variable discharge flowrates, it is a perfect pump for batching applications. The pump can operate against a restricted discharge without pump damage that would make it an excellent choice for bulk dispensing, batching, or feeding a pressurized system. It can run dry without damage to the pump or piping and is, therefore, excellently suited for transfer applications, emptying tanks, and dewatering. The pump can also handle fluidized dry powder transfer applications and shear-sensitive or fragile liquids such as paint, ink, clay slips, and many food products such as milk, beer, wine, oils, and semisolids such as tomatoes, strawberries, olives, cherries, and grapes.

PERFORMANCE RANGE—PUMP SELECTION

The user should know:

- Available air supply, pressure, and volume
- Pumping flow required
- Head pressure requirements
- Characteristics of product being pumped, i.e., viscosity, specific gravity, temperature range, and solids in suspension
- General pumping system layout

Pump manufacturers publish performance curves for each of their pump sizes (Figures 19 and 20). The vertical scale is the pumping head or pressure; the horizontal scale is the flow. Each of the curves touching the vertical (head) and horizontal (flow) scales represent a constant air inlet pressure to the pump. The curve that cuts through the performance curves is the scfm, (air consumption in standard cubic feet per minute) and represents the volume of air used. For the performance curve illustrated, the results plotted are of a pump pumping clear water at sea level, 68°F, with the static water level at the centerline of the pump, in a test laboratory.

Other performance curves are available using other pumping media and will be noted as such. However, the interpretation of the performance curves will be the same. The user calculates the piping system and flow requirements and by comparing the various sized pumps, the user can determine the size and most economical performance range best suited for that particular application. To do this, the user, having determined the total head requirement (vertical scale) and the flow required (horizontal scale), will draw lines of intercept. This will denote the required air inlet to the pump and the amount of air consumed. Because the performance curve is produced under laboratory conditions, this point on the performance curve is usually a "best case" range.

For example: the users basic requirement is to pump dirty water from a shallow pit area to a tank car. He determines that there is 35 ft of vertical distance from the pump to the top of the tank car and

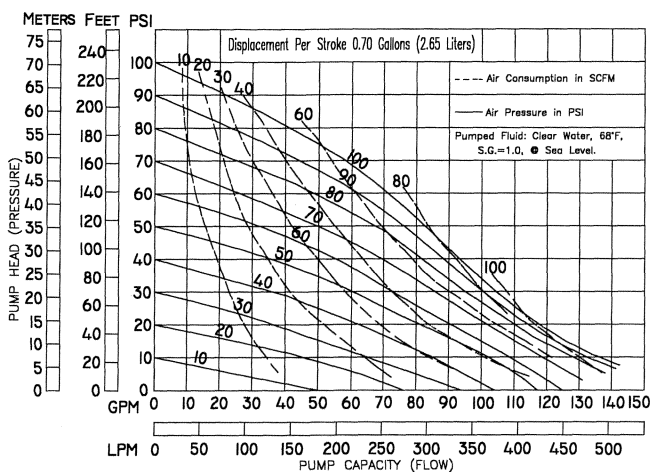


Figure 19. Two Inch AODD Performance Curve.

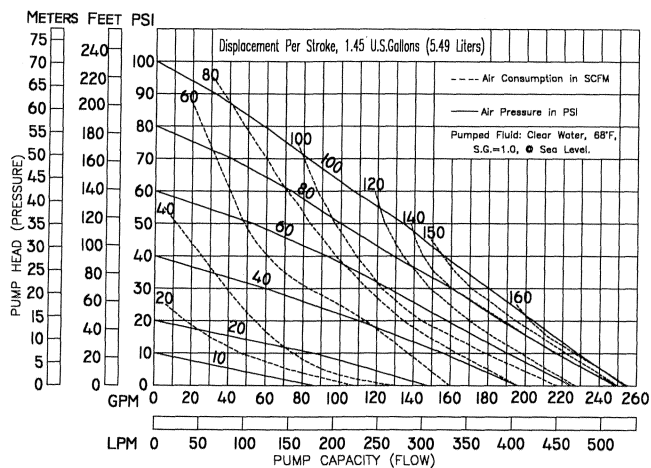


Figure 20. Three Inch AODD Performance Curve.

three ft of vertical depth in the shallow pit and wants 100 gallons per minute (gpm). His air compressor can deliver 60 scfm of air at 100 psi. Looking at several performance curves of different sizes of pumps (Figures 19 and 20), he finds that there are two sizes available to him; two inch and three inch that will deliver 100 gpm at approximately 40 ft of head. The two inch would consume approximately 52 scfm at 60 psi air inlet pressure; the three inch would consume approximately 53 scfm at 38 psi. Since the cost of operation (scfm) is approximately the same, the larger three inch sized pump should be the choice for several reasons:

- The displacement of fluid per stroke on the three inch pump is nearly two times that of the two inch pump. Therefore, it would be cycling slower causing less wear and tear on pump components.
- There is more operational range left on the performance curve within the capacity of the compressor.

If the installation has other factors that could take away from the laboratory testing environment, the user must build in a safety factor. Many times simply raising the air inlet pressure or jumping up a curve or two, on a specific size pump, will account for a safety margin. That is one of the many advantages of the AODD in industrial systems.

Many manufacturers publish a "Viscosity Correction Factor Chart" (Figure 21). The correction factor number, which is found by using this chart, will be used in conjunction with the performance curves located in the sales literature of the operating

manual for a particular pump model. Multiply the correction factor number by the gpm that is desired. Locate the new gpm number on the performance curve to determine the air inlet pressure and scfm needed to deliver the required gpm.

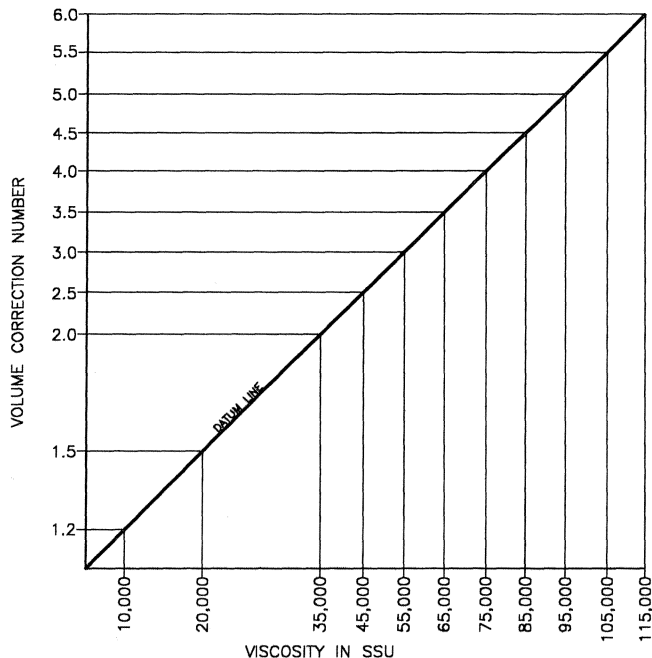


Figure 21. Viscosity Correction Factor Chart.

Example: Required is 20 gpm at a discharge head of 50 psig. The viscosity is 35,000 SSU. Locate 35,000 SSU on the chart, follow the line vertically to the datum line. Follow the horizontal line to find the correction factor number. In this example the number is "2.0." Multiply this number, "2.0," by the required gpm, which is 20, $20 \times 2 = 40$ gpm. Now locate 40 gpm at 50 psig head on a particular performance curve to determine air inlet pressure and scfm that is required. Take note this correction factor is to be used as an aid to approximate performance of an AODD pump in an uncomplicated installation. Suction requirements must always be investigated when pumping high viscosity fluids [13].

MATERIALS OF CONSTRUCTION

The materials of choice should be divided into areas of the pump. The "wetted side" would be all components that come in contact with the pumping media. The "air side" would be all components that would be within the air distribution system and not in direct contact with the pumping media (Figure 22).

The "wetted side" components include the chambers, elbows, manifolds, outer diaphragm plates, check valves, check valve seats, and the flexible diaphragms. AODD pumps are available in a wide range of corrosion resistant metals and plastics. Diaphragms, check valves, and check valve seats are available in a variety of chemically resistant polymers.

Each pump manufacturer provides a list of available materials of construction and their most suited application.

When choosing the proper materials for an application, the user must take into consideration the following:

- Chemicals being pumped
- Temperature range
- Abrasiveness of pumping media
- Environment of pump exposure

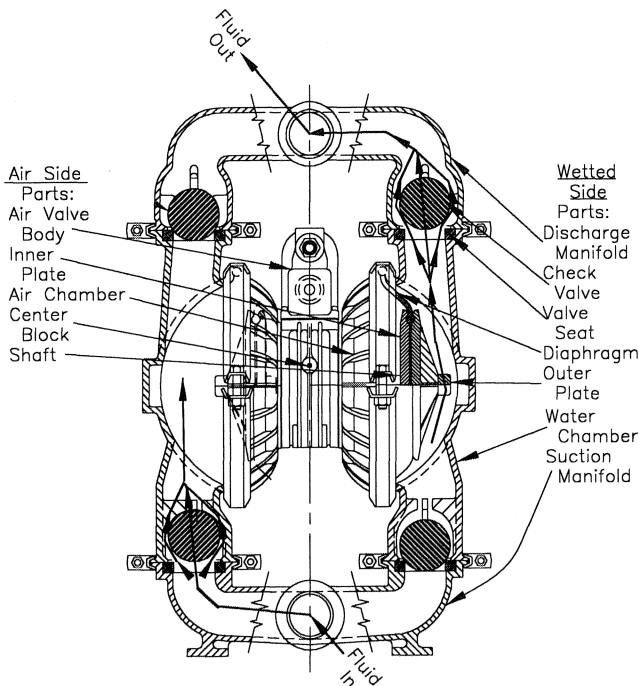


Figure 22. Pump Component Description.

The wide range of materials of construction and the various elastomer materials available should offer a suitable solution to a particular fluid transfer application.

SUITABLE APPLICATIONS

The ceramic industry uses the AODD pump to circulate an abrasive and viscous ceramic glaze that is sprayed on plumbing fixtures before they are kiln fired [14]. This material is circulated continuously, 24 hr/day, seven days/wk. These pumps must be nonicing, self-lubricating, and nonstalling.

A latex producing company is using AODD pumps to circulate and transfer latex emulsion. These pumps run constantly at full line pressure. They were also pumping titanium dioxide, which is very abrasive and extremely dilatant under pressure. The pumps run continuously, 30 percent of the time at a discharge pressure of 80 psi and 70 percent of the time at 40 psi. Fluid temperature was relatively constant at 60°F to 70°F. These pumps must be nonicing, nonstalling, and self-lubricating.

AODD pumps used to pump affluent into filter presses [15] is a typical application. The pumps are run at full line pressure. As the filters fill up, the head pressure builds until the pump simply quits pumping while holding the full line pressure to further squeeze the cake. These pumps must be anti-icing, nonstalling, and self-lubricating.

A grinding wheel manufacturer uses the AODD pump in a sump area application. The pumps are pumping many daily on/off cycles of carborundum slurry and waste water, which are very abrasive, from a shallow collection pit into a holding tank for further recycling processing.

A cosmetic manufacturer is using AODD pumps to recirculate mixing tanks for various cosmetics and lotion products. These pumps must be self-lubricating, so as not to contaminate batching.

Because the pumps operate on compressed air, there has been an inherent problem of icing and stalling. Until recently, the only solutions being utilized to solve this problem included compressed air dryer, heaters, steam systems, and anti-icing additives. Unfortunately, these solutions are not practical or efficient. An additional component or product becomes a necessity to ensure a consistent production run.

Currently, several AODD pump manufacturers are addressing alternate solutions to those mentioned. The most viable being an antistalling, nonicing air valve system. The theory behind these air valves is to minimize the sudden expansion of air to the atmosphere. When the compressed air is released from the air chamber and allowed to rapidly expand, a phenomenon occurs that will cause any moisture in the air to freeze instantly. This rapid drop in temperature, that happens with each exhaust, will eventually cause any moving valve parts to stall, i.e., not move, resulting in "freeze-up."

Through testing and actual use, this antistalling, nonicing air valve system is solving the problem of freeze-up and eventual stalling that results in expensive downtime. This antistalling, nonicing air valve system is a standard component on several of the AODD pumps in the market today.

Positive reaction is being heard as many industries are replacing their pumps utilizing standard air valves with pumps utilizing nonicing, antistalling air valve systems.

INSTALLATION

The AODD pump is a reciprocating pump and, therefore, can produce vibrations and surges that can produce problems on piping systems. There are basic installation guidelines offered by the manufacturer (Figure 23) that could minimize some of the vibrations. Some pump manufacturers offer pulsation dampeners that can be installed in the outlet and/or inlet piping in certain applications. Correctly installed dampeners provide a smoother continuous flow that is critical for proper performance of batching controllers or flow meters. It also protects piping and valves and increases the life of pump components. There is evidence that suggests the installation of a pulsation dampener on the inlet side of AODD pumps used in high flow rate applications may dramatically improve diaphragm life, especially in start-and-stop applications.

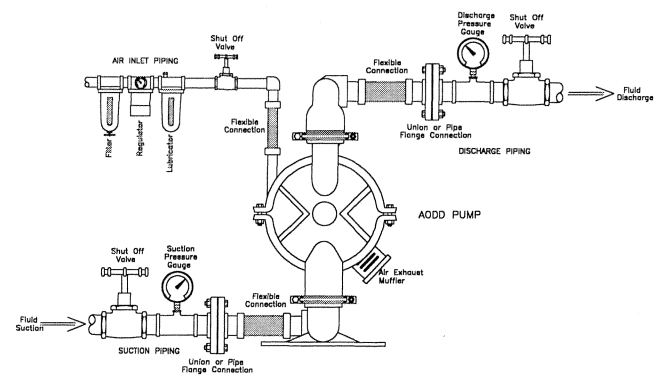


Figure 23. Recommended Installation Guidelines.

The AODD pump needs to be installed in a vertical position. For permanent installations where the pump is bolted to a rigid structure or the floor, vibration cushions should be placed between the bottom of the pump and mounting surface.

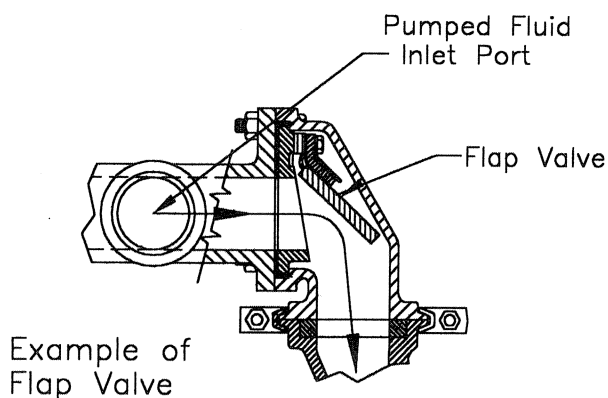
The AODD pump can be installed submerged. Some pump manufacturers offer screen mount pumps for this reason, such that the pump inlet is surrounded by a particular sized suction screen. The air exhaust with submerged pumps must be piped above the liquid level to prevent liquid from entering the exhaust port, when the pump is not operating.

CHECK VALVES

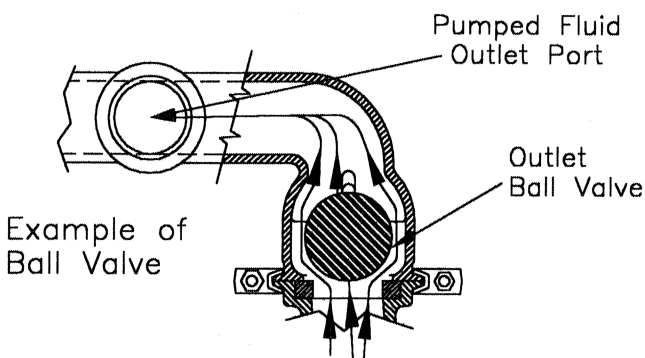
Because the AODD pump has one inlet port and one outlet port and two chambers dispensing product, there must be internal check

valves that control the direction of product flow. Each side of the pump has two check valves, one below the centerline of the pump and one above the centerline.

There are, basically, two types of check valves available in the standard AODD pump, ball check valves and flap valves (Figure 24). Each has distinct advantages.



Example of Flap Valve



Example of Ball Valve

Figure 24. Check Valves Used with AODD Pumps.

The ball check valve (Figure 25) offers a quicker, tighter seal for better priming characteristics. It operates well in thick, viscous materials. This valve is available in many different materials of construction including synthetic rubber, thermoplastic elastomers, and stainless steel.

The flap valve (Figure 26) allows the passage of solids. In some cases, semisolids as large as the inside diameter of the pipe size can be handled.

The AODD pump design is different with each valve style. In fact, in the ball valve pumps, the inlet porting is at the bottom of the pump. The outlet porting is at the top (Figure 25). In the flap valve pump, the porting is reversed (Figure 26). The porting on the flap valve pumps offers a distinct feature when pumping fluids with solids in suspension in that gravity helps the solids through the pump. When the pump is shut down, the solids settle to the bottom and are flushed out during the initial cycling of the pump, upon restart. The flap valve pump is very successful in pumping shear-sensitive semisolids such as tomatoes, grapes, and other food products. The materials of construction are somewhat limited, due to the design configuration of the flap valve. The flap valve seals with the help of gravity and pressure differential across the part.

The ball check valve uses the weight of the ball and differential pressure to seat tightly. They operate very well in fluids with suspended solids, within the size limits to a particular pump model,

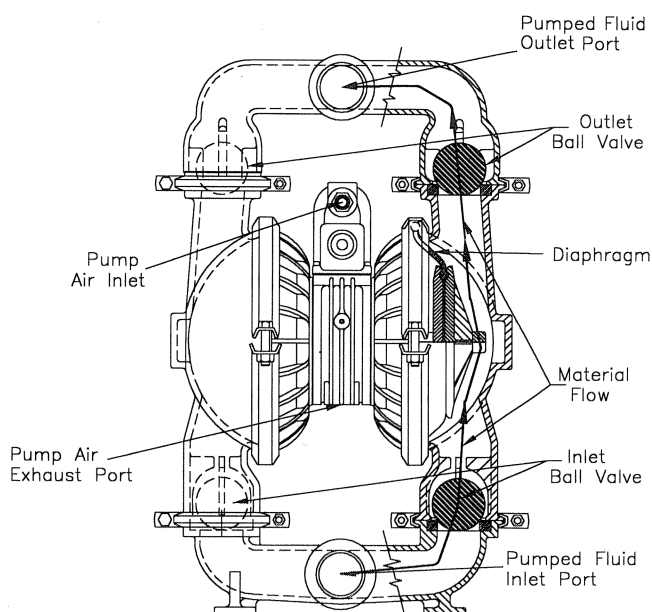


Figure 25. Two Inch Ball Valve Pump.

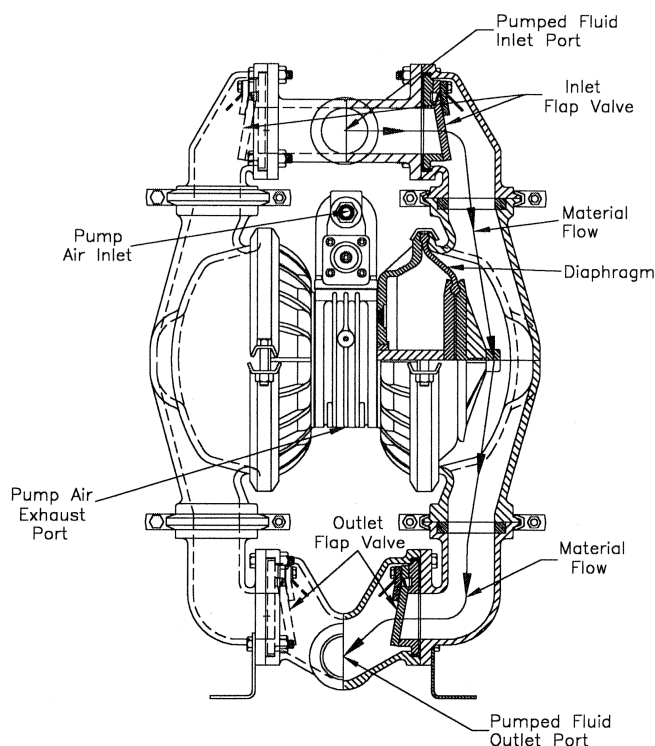


Figure 26. Two Inch Flap Valve Pump.

and very thick, viscous materials. Depending on the piping system setup, the AODD pump could economically handle a viscosity range to 100,000 SSU \approx 11,000 cPs).

DIAPHRAGM

The diaphragm is possibly the most important part of the pump. The diaphragm is a flexible membrane that separates the pumping media from the air pressure. It flexes with each pump stroke and is subject to the impact of the pumping media and any differential pressures. The materials of construction are many, from fabric

reinforced synthetic rubber to thermoplastic elastomers that will operate in temperatures of -40°F to $+350^{\circ}\text{F}$ and fluid solutions with numeric pH levels from zero pH to 14 pH. Diaphragm materials should be carefully selected.

LIMITATIONS OF THE AODD PUMP

The flap valve pump, generally, is not known for great dry priming ability. It is best suited for hopper feeding or the pump being below the level of the liquid (positive suction head). The suction side of the piping system must be looked at very carefully.

It may be difficult to get consistent and predictable performance characteristics from both the ball check and flap valve check pumps, with product that has entrapped air, such as concrete or whipped products. The pumps will handle the product, but may lose prime often, cycle very erratically, and splatter excessively at the discharge. This does not cause damage to the pump, just unpredictable performance.

The pump discharge pressure on an AODD pump is limited to the air inlet pressure used to power the pump.

The pumps do not operate well in series. There are limitations to positive suction pressure applied to the inlet. For the AODD pump during the suction stroke, this is the only time the diaphragm feels excessive differential pressure, because it is being pulled through a stroke by the connecting rod attached to the other side of the pump. Some pump manufacturers say that pressures exceeding 10 psi will greatly reduce the life of the diaphragms. The pumps, however, work very nicely in parallel.

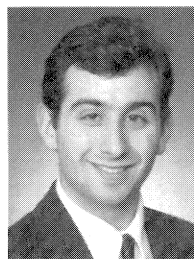
ADVANTAGES OF THE AODD PUMP [16]

- Can run dry for extended periods of time
- Can pump highly viscous materials
- Does not require complex controls to install and operate
- Exceptionally tolerant to variations in operating conditions
- Self priming dry or wet
- Infinitely variable pumping rate and pressure within the pressure and capacity range of the pump
- No mechanical seals or packing
- Explosion proof for use in hazardous environments
- Can be used in confined areas, no motor or electrical parts to cause heat buildup
- Can pump abrasive slurries and solids in suspension
- Simple to maintain and service
- Can be used in pumping aggressive chemical solutions
- Can pump dry powders in air suspension
- Gentle pumping of shear-sensitive materials
- Can pump materials in temperature range of -40°F to $+350^{\circ}\text{F}$ depending on choice of materials of construction
- Can pump against a shut off discharge head without damage
- Can pump a wider range of materials than any other type of pump

CONCLUSION

- The AODD pump is a very versatile pump.
- The user must be sure that there is an adequate and continuous supply of air.
- The user must know the installation and the requirements that the pump must meet such as: flow, head pressure, suction pressure, viscosity of materials being pumped, pipe size, temperature range of pumping media, and chemical makeup of pumping media.

THE APPLICATION OF HYDRAULICALLY ACTUATED DIAPHRAGM METERING PUMPS



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ABSTRACT

Hydraulically actuated diaphragm metering pumps find wide application in the water treatment and process industries. They afford the proper pump choice for applications requiring positive displacement metering and dosing accuracy, leakfree chemical transfer, high pressure delivery, and controlled handling of toxic, corrosive, or expensive fluids. Application of these metering pumps require that special consideration be given to the properties of the fluids that are pumped—including vapor pressure, viscosity, rheology, temperature extremes, corrosivity, solids content, and specific gravity.

Beyond the required consideration of the fluid's properties, the pump and system must be designed together to assure the specified fluid handling performance. In particular, suction side pressure requirements of net positive suction head and minimum suction head must be met, while on the discharge side, the system must be designed so as to control the back pressure. Pulsation dampeners may be employed to alter the available suction side pressure or back pressure conditions. Also, the proper method of flow control must be selected to meet the demand of the application. Flow may be controlled by adjusting the stroke speed (frequency), stroke length, or stroke cycle. These system requirements must be properly evaluated to obtain maximum benefit from the use of hydraulically actuated diaphragm metering pumps.

INTRODUCTION

Background

The pump universe consists of two main categories of pumps. Any one pump is either a dynamic pump or a displacement pump. There are also many subcategories beneath these two main families of pumps. A hydraulically actuated diaphragm pump can be found on the displacement side of the pump spectrum, under the subcategory of reciprocating. Because these pumps are positive displacement and more importantly because they are reciprocating type pumps, the user and/or the system designer must know the basics about how to select a pump for the application and the pumping system [17].

The selection of a hydraulically actuated diaphragm metering pump is dependent upon two separate criteria. First, the design of the chemical metering system must ensure that the fluid can reach the pump, and have no excess restrictions to the point of chemical

injection. Second, the pump must be capable of handling the transfer of the fluid once it is within the pump. In addition to explaining the criteria involved for selecting a pump and designing a system for the pump, general application guidelines for pump control and operation are included herein.

Principle of Operation

The principle of operation of a hydraulically actuated diaphragm metering pump is described. A prime mover, such as, but not limited to, an electric motor, is used to transmit power to the drive train of the pump. The drive train consists of gearing and power mechanisms that usually include some type of cam and follower or crank principle. A measuring piston or plunger reciprocates within a precisely sized cylinder at an established stroke length, displacing a volume of hydraulic fluid, not the process liquid. The hydraulic fluid is chemically stable and has a relatively low vapor pressure. The displacement of this hydraulic fluid, in turn, deflects the diaphragm, which then moves the process fluid in and out of the process reagent head of the pump. The diaphragm's only function is to separate the hydraulic fluid from the process liquid [17]. The pump's reagent head, process liquid side of the diaphragm, has two check valves, one on the discharge side of the head and one on the suction side of the head. When the piston or plunger is on its backward stroke, the suction side check valve opens up to let the process fluid in the reagent head. On the forward stroke of the piston, the hydraulic fluid is compressed to a pressure just greater than the discharge pressure exerted on the discharge check valve. Once the hydraulic fluid is compressed to this pressure, the process fluid is moved through the discharge check valve and into the discharge line towards the point of injection. The hydraulic diaphragm metering pump will always have some method for air bleeding from the hydraulic fluid, hydraulic fluid makeup, and hydraulic fluid bypass for system over pressures.

Cross section drawings of two types of hydraulic diaphragm metering pumps are shown in Figures 27 and 28. A typical "lost motion" type of metering pump is depicted in Figure 27, a typical "full motion" type of metering pump is depicted in Figure 28. The difference between full and lost motion is explained in *Flow Control Methods*, in the DISCUSSION section.

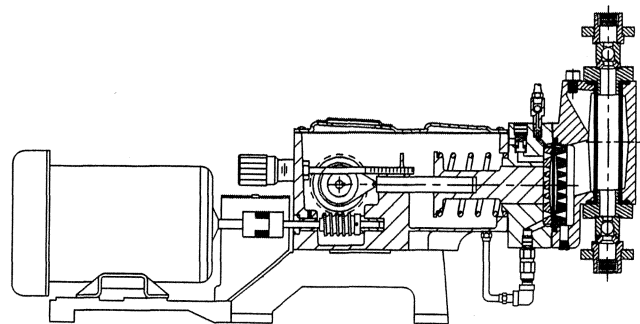


Figure 27. Illustration of a Typical Lost Motion Type Hydraulic Diaphragm Metering Pump.

Typical Performance

The typical flow and pressure range of a hydraulically actuated metering pump varies by manufacturer, but, in general, the majority of the market resides in an envelope with flows to 1500 gph and discharge pressures as high as 5000 psig with a simplex configuration of the pump. Typically, low flows are coupled with high pressures and higher flows are usually at lower pressures. Applications outside this envelope can be covered using special adaptations and multiplex configurations.

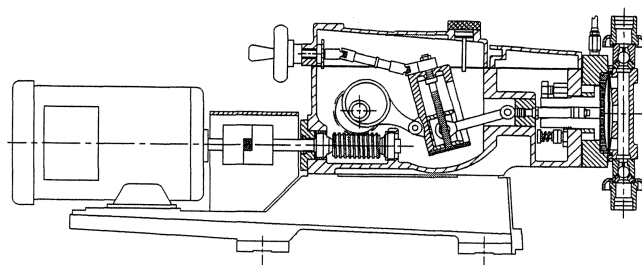


Figure 28. Illustration of a Typical Full Motion Type Hydraulic Diaphragm Metering Pump.

The turndown and accuracy of a pump is dependent upon the manufacturer and the model of pump selected. The typical turndown for a pump with standard manual stroke length control is 10:1, the stroke length is adjustable from 100 percent down to 10 percent with the accuracy stated. Other turndown ratios are available and are discussed in the next section, *Flow Control Methods*, and in the DISCUSSION section, *Flow Control Methods*. The typical accuracy of these pumps, with manual stroke length adjustment, is shown in Table 4.

Table 4. Typical Accuracy Statements for Hydraulic Diaphragm Metering Pumps.

Accuracy Type	Typical Value	API 675 Required Value	Capacity Reference	Turndown Ratio
Steady State	± 1.0%	± 1.0%	Full Flow	10:1
Repeatability	± 1.0%	± 3.0%	Full Flow	10:1
Linearity	± 1.0%	± 3.0%	Full Flow	10:1

Steady state accuracy is defined as the variation in capacity over a specified period of time, under fixed pump and system conditions expressed as a percentage of either set point flow or full rated pump flow. *Repeatability* is defined as the variation in capacity resulting from a specified excursion from a capacity set point, followed by a return to that set point, expressed as a percentage of the full rated pump flow. *Linearity* is perhaps the most commonly mentioned accuracy. It is defined as the relationship between actual capacity and calibrated capacity. Calibrated capacity is a linear function of capacity setting based on a calibration test and the least squares linear regression method. Linearity is the maximum deviation from the calibration at any point, and is expressed as a percentage of the maximum rated capacity.

Flow Control Methods

The primary means of flow control in applications involving metering pumps are:

- Stroke length
- Stroke speed
- Stroke cycle

One or all of these means may be employed, and can be controlled either manually or automatically. The most commonly used method of flow control is via stroke length, as illustrated in Figure 29.

PUMP SELECTION CONSIDERATIONS

System Requirements

When a chemical metering system is designed for a hydraulically actuated diaphragm metering pump, there are several variables to consider in order to ensure proper system operation and sound pump hydraulics. Some system variables that must be considered for metering pump applications are system suction

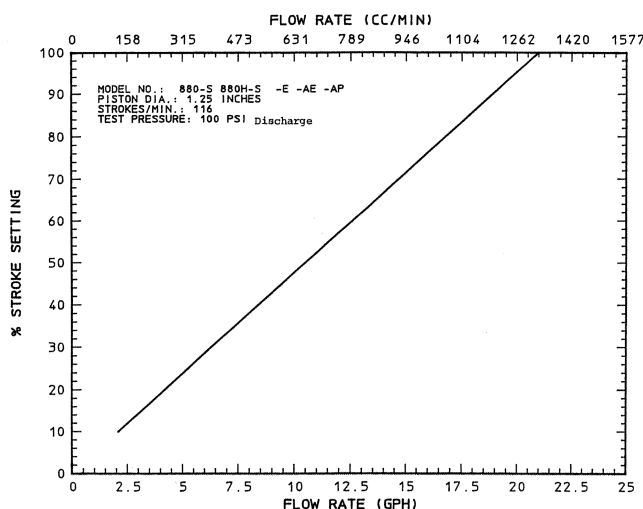


Figure 29. Typical Pump Flow Curve with Stroke Length Adjustment for Flow Control.

pressure at the pump inlet, system backpressure at the pump outlet, and length, shape, and diameter of system piping. All of these variables can be combined into formulas that act as tools used to easily determine if the pump and the system will provide the desired response. It is important to remember, also, that hydraulically actuated diaphragm metering pumps inherently involve reciprocating type flow and not steady state flow, therefore, steady state approximations and rules of thumb should not be used in these pumps' applications and installations. Every hydraulically actuated diaphragm metering pump has requirements that the system must provide in order to ensure optimum pump performance. A diagram of a typical chemical metering system is shown in Figure 30 that can be referenced when reading the following description of system requirements [18].

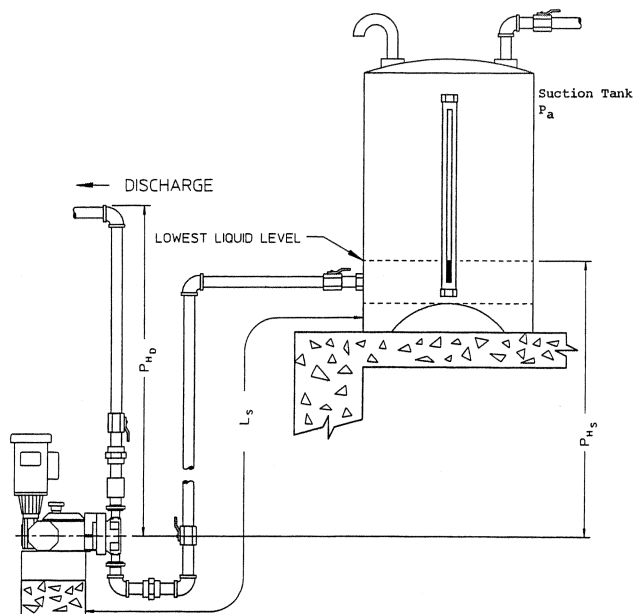


Figure 30. Diagram of a Typical Chemical Metering System.

Suction Head Requirements

• **Net Positive Suction Head Required (NPSH_R)**—The NPSH_R is defined as the pressure required above the absolute vapor pressure

of the process fluid at the pumping temperature. This pressure is required at the suction port of the pump throughout the entire pump stroking cycle, in order to prevent cavitation of the process fluid within the reagent head. The NPSH_R is derived from the following elements, with estimated values for waterlike process fluids, where pressure drop occurs:

- Suction check valve = 2 psi to 3 psi
- Reagent head dish plate (process side) = 1.5 psi to 2 psi
- Total NPSH_R = 3.5 to 5 psi (varies by manufacturer)

To determine the net positive suction head available (NPSH_A) in the pumping system, the following equations are used (Figure 30):

For fluid viscosity below 50 centipoise.

$$NPSH_A = P_A \pm P_{H_S} - P_V - \left(\frac{L_S R G Q}{C_1 d_S^2} \right) \quad (13)$$

For fluid viscosity above 50 centipoise.

$$NPSH_A = P_A \pm P_{H_S} - P_V - \sqrt{\left(\frac{L_S R G Q}{C_1 d_S^2} \right)^2 + \left(\frac{L_S \mu Q}{C_2 d_S^4} \right)^2} \quad (14)$$

The last term of each Equation (13) and (14) is referred to as the "acceleration loss" term.

• **Minimum Suction Head Required (MSH_R)**—Because hydraulically actuated diaphragm metering pumps are hydraulically balanced, the hydraulic side (not wetted with process fluid) of the diaphragm must also be kept rigid (not cavitating) by assuring that a minimum *absolute* pressure, minimum suction head (MSH_R), be maintained at the pump inlet throughout the pumping cycle to ensure a stable hydraulic system and proper pump operation. Minimum suction head available (MSH_A) is defined as the sum of the NPSH_A and the vapor pressure (P_V) of the process liquid and must be greater than the MSH_R.

Discharge Side Requirements

• **Minimum differential pressure**—Every hydraulically actuated diaphragm metering pump has a minimum differential pressure, difference between the discharge and suction pressure, required in order to properly seat the process side check valves. If this condition is not met, flowthrough can occur. Flowthrough can be defined as the process liquid flowing from a higher pressure to a lower pressure (downhill pumping), which attributes to pump failure and undesired flow at pump shutdown. The minimum differential pressure will depend upon pump design, but will typically range from as little as five psi to as much as 50 psi.

• **Maximum system discharge pressure**—Naturally, there are limitations imposed on the components of the pump. All hydraulically actuated diaphragm metering pumps provide for some means of bypassing the hydraulic fluid in the event of an over pressure in the discharge, however, all pumps also have their pressure limits. The hydraulic bypass system must be used only to protect the pump from discharge over pressure and a separate pressure relief valve should be installed in the process piping to protect other pieces of process equipment or instrumentation. The total system backpressure as seen by the pump can be calculated as follows (Figure 30):

For fluid viscosity below 50 centipoise:

$$P_T = \left(\frac{L_D R G Q}{C_1 d_D^2} \right) + P_P \pm P_{H_D} \quad (15)$$

For fluid viscosity above 50 centipoise:

$$P_T = \sqrt{\left(\frac{L_D R G Q}{C_1 d_D^2}\right)^2 + \left(\frac{L_D \mu Q}{C_2 d_D^4}\right)^2} + P_P \pm P_{HD} \quad (16)$$

Notice that C_1 and C_2 are the same constants as in Equations (13) and (14). Actually, the whole acceleration term format is the same on the discharge side as it is on the suction side. This illustrates the similarity of the reciprocating action on both sides of the pump.

Fluid Velocity

Most centrifugal and rotary pumps have steady state flow characteristics. Under normal conditions, there is little change in the velocity of the fluid pumped. That is not the case with reciprocating metering pumps, which cause the fluid to experience peak velocities and accelerations in both the suction and discharge lines.

Fluid velocity is often used to determine the correct pipe size for an application. Since the fluid velocity generated by a metering pump varies throughout each stroke of the pump, piping must be sized based on the peak fluid velocity and not the average fluid velocity. The peak fluid velocity, V_p , generated by a positive displacement metering pump can be calculated using the following equation, where the C_3 values are determined by the pump manufacturer:

$$V_p = \frac{Q}{C_3 d^2} \quad (17)$$

Fluid Properties

The ability of a metering pump to handle a given fluid is dependent upon certain fluid properties, which include:

- Vapor pressure
- Viscosity
- Rheology
- Temperature
- Corrosiveness
- Solids content
- Specific gravity

Fluid properties may influence the chemical metering system configuration, as a result of the system requirements, in addition to playing their own role in the selection of a hydraulically actuated diaphragm metering pump.

DISCUSSION

Flow Control Methods

Stroke Length

The hydraulic piston that displaces the hydraulic fluid to deflect the diaphragm travels a certain distance within the cylinder, which has a fixed length and diameter, and therefore, a fixed volume. The majority of hydraulically actuated diaphragm metering pumps have some means of limiting the travel of the piston within the cylinder, which in turn varies the flow output of the pump. This flow control means usually comes standard with a pump as a manual adjustment controller that has a turndown as specified by the pump manufacturer, dependent upon the degree of accuracy stated.

As mentioned earlier, many hydraulically actuated diaphragm metering pumps employ a cam and follower principle in the drive train of the pump. Those that use a cam and follower for power

transmission are referred to as “lost motion” types of pumps, because the driving element is always driving, but the piston’s backward travel can be limited by the stroke controller, thus the name *lost motion* (Figure 27). Another version of lost motion is used on some hydraulically actuated metering pumps where an adjustable orifice controls the amount of bypass of the hydraulic fluid, which directly affects the amount of deflection translated to process fluid flow.

Pumps that employ cranks as opposed to cam and followers are referred to as “full motion” pumps. There is some type of swinging assembly that is always moving the piston forward or backward and the travel of the piston is limited only by the angle of the swing of the assembly, thus the name *full motion* (Figure 28).

Most metering pumps can be selected to have automatic control as an option to vary the stroke length. The most common automatic control signal is 4-20 mA, although some newer pumps have the capability of communicating directly with a facility’s control system via RS-232/485 serial port communication.

Stroke Frequency

The most common input speed to the pump from the drive motor is 1750 rpm in 60 Hz power supply. Many applications involve the need for a higher turndown than obtainable with stroke length adjustment alone. These applications employ variable speed drive motors of three main categories in order to vary the stroking speed and the stroke length. The most common variable speed packages available are:

- *Silicon controlled rectifier*—The more commonly known DC “SCR” controller is commercially available for a speed turndown of a DC motor to about 20:1 with a degree of accuracy compatible with the pump.
- *AC variable frequency drives*—The AC drive is commercially available in many different turndown ratios dependent on the combined selection of the motor and the drive. The range of speed turndown ratios available are as high as 2:1, or as low as 100:1.
- *Air driven motors*—Metering pumps can also be driven by an air driven motor. The air supply flow and pressure to the motor can be varied to affect the output speed of the motor.

No matter which method of variable input speed to the pump is used, all metering pumps require a constant torque input from the motor independent of motor speed. Constant torque is required because the pump is a constant load and requires the same amount of torque throughout the stroking cycle. Automatic control depends on the method, manufacturer, and model of the variable speed drive element selected.

Stroke Cycle

For extremely low flow ratios of additive stream to the primary fluid (beyond 100:1), flow may be controlled by cycling the pump on and off. This method of flow control, sometimes referred to as batching, is successful for processes that include a high residence time for the treated fluid.

System Requirements

Suction Head Requirements

If the chemical metering system does not afford adequate suction head conditions to the pump, one or more of the following recommendations may be made referencing Equation (13) and Figure 30:

- Increase supply tank blanket pressure
- Increase head pressure above pump (raise supply tank)
- Decrease length of suction pipe

- Increase diameter of suction pipe
- Slow pump stroking rate

If these recommendations are not suitable or feasible, the installation of a pulsation dampener in the suction line as close as possible to the pump's suction port will act as an accumulator and will effectively reduce the friction losses in the suction line. The application of pulsation dampeners is explained later in this section.

Discharge Side Requirements

If insufficient total system backpressure is present, or the minimum differential pressure is not met, the installation of a backpressure valve is required. Differential pressure refers to the difference between the discharge and suction pressures. This valve creates artificial pressure by means of restricting the flow on the discharge side. This allows the pump check valves to seat properly.

If the total system backpressure exceeds the capability of the pump, the following recommendations can be made referencing Equation (15) and Figure 30:

- Decrease length of discharge piping
- Eliminate any extraneous pipe fittings or process equipment
- Slow pump stroking rate
- Increase diameter of discharge piping

Pressure fluctuations in applications of a reciprocating hydraulically actuated diaphragm metering pump are inherent in the pump's design. In many instances, a pulsation dampener is installed in the discharge line of the pump in order to eliminate pressure surges and to protect sensitive instruments from these fluctuations. To some degree, the pulsation dampener can be used to effectively decrease the amount of total system backpressure in the same manner.

Metering Pump Accessories

• **Pulsation dampeners**—A pulsation dampener is a pneumatically charged diaphragm within a chamber that continuously stores energy carried in the acceleration of the pumped fluid. When a pulsation dampener is used on the suction side of a metering pump, it will improve suction pressure conditions by reducing the pressure losses associated with the acceleration of the fluid. On the discharge side, it will protect sensitive equipment from pulsating flow spikes, and will transform pulsating flow to nearly linear flow by reducing both the peak flow and the pressure generated by the metering pump. Regardless of which side of the pump the dampener is to be used on, it should always be installed as close to the pump as possible in order to reduce the length of pipe in which pulsating flow occurs.

In many installations, the diameter of costly piping can be reduced by incorporating a pulsation dampener on both suction and discharge lines. Installing a dampener can also reduce cost by allowing the use of a smaller faster stroking pump as opposed to a larger, more costly, slower stroking pump.

When a pulsation dampener is used, the length of piping (L_S or L_D) in Equations 13, 14, 15, and 16 becomes the distance between the pump and the pulsation dampener. For the remainder of the piping, uniform flow friction losses can be used since the flow is now constant and not pulsating for this distance of piping [19]. These losses can be found in any standard friction loss table based on Darcy's formula. The standard friction losses must also be subtracted from the $NPSH_A$ or added to the backpressure calculations (Equations 13, 14, 15, and 16).

• **Backpressure valves**—As described earlier, the backpressure valve is installed in the discharge piping if the minimum

differential pressure is not met. The backpressure valve is usually a spring loaded diaphragm or poppet that creates artificial pressure by means of providing additional frictional losses in the discharge line. The valves are field adjustable so that the correct pressure resistance can be set.

• **Pressure relief valves**—Pressure relief valves external to the pump (on the discharge side) are always recommended to protect other process connections, instruments, and equipment from an over pressure in the system.

• **Calibration columns**—Metering pumps must always be calibrated in the system in which they will perform. The use of a calibration column, usually on the suction side, provides a simple indication of metering pump performance in that system.

• **System strainers**—For fluids with solids that are not integral to the process, a suction side strainer is always recommended. The required mesh size is available from the pump manufacturer.

The layout of an ideal metering pump system is illustrated in Figure 31.

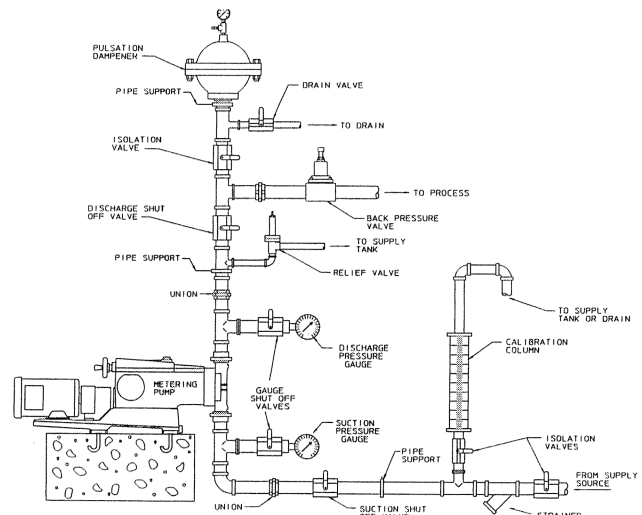


Figure 31. Layout of the Ideal Metering Pump System.

Fluid Properties

Vapor Pressure

When handling fluids with a substantial vapor pressure, generally above one psia (0.07 bar (a)), the result of an uncontrolled high vapor pressure is observed as the cavitation of the product pumped. To control high vapor pressures, the suction line must always be at a pressure greater than the vapor pressure of the fluid. This is usually accomplished by "blanketing" the supply vessel with an inert gas. By reviewing Equation 13, one can see the effects of a high vapor pressure on the $NPSH_A$. As with any pump, the $NPSH_A$ to the metering pump must be greater than, or equal to, the $NPSH_R$. Otherwise, cavitation would result in both the damage of internal components and the loss of metering accuracy, or the ability of the pump to move the fluid at all. Those fluids that would be suspect of this difficulty would include light hydrocarbons. Those fluids with a low vapor pressure include heavy hydrocarbons (high boilers), or aqueous solutions of ionic or molecular bonded constituents. To improve the ability to control a fluid with substantial vapor pressure, the fluid's temperature can be lowered, the system (Figure 30) could be changed to increase the suction head pressure (P_H), increase the supply tank blanket pressure (P_A), decrease the length of suction pipe (L_S), or the metering pump selected to provide minimum practical stroking

rate, (R). The selection of a pump with lower stroking rate will usually force the improvement of other parameters, for instance, a larger pump head, internal porting, and valves.

Viscosity

A fluid’s viscosity is the ratio of shear stress to shear rate as expressed below:

$$\mu = \frac{\tau}{\Gamma} \tag{18}$$

By nature of this definition, a fluid property called the shear stress, τ , is combined with a dynamic condition called shear rate, Γ , and thus allows some measure of control by the system designer [20]. For instance, the shear stress of a fluid is a function of temperature and can be reduced by increasing its temperature, thus decreasing its viscosity. Shear rate can be influenced by other system and pump factors, which will be discussed below and in the next section, *Rheology*.

Viscosity is reported as absolute viscosity, μ , when the fluid’s shear rate is measured using a rotating spindle. Kinematic viscosity, ν , is determined by measuring the time to pass a known amount of fluid through an orifice. These two physical properties are measured differently, but that does not affect the rules of application for metering pumps with viscous fluids. The following formula presents the conversion between absolute and kinematic viscosity:

$$\mu = \rho\nu \tag{19}$$

When selecting a pump to handle a viscous fluid, generally above 500 cP with a Newtonian fluid, those pumps that provide minimum internal restriction are the best choices. Tubular or conical diaphragms, which offer a straight-through flow path, or flat diaphragm design pumps without a front diaphragm support plate would all be suitable (Figure 32). Other application guidelines for reducing the shear rate of the fluid include:

- Over-sizing the pump for greater power capability and greater clearances.
- Over-sizing the pump’s check valves and openings.
- Selection of the slowest practical stroking rate.
- Over-sizing the process piping.

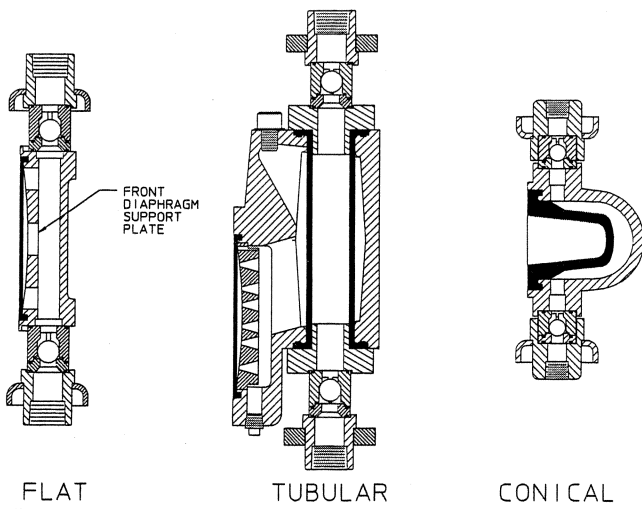


Figure 32. Diaphragm Styles Available, Flat, Tubular, Conical.

Rheology

The fluid’s rheology offers yet another consideration. Newtonian fluids are shear independent, and if the reported viscosity falls within the capability of the pump, they will certainly be handled as anticipated. Non-Newtonian fluids offer a different challenge. Any fluid rheology which results in increased viscosity with an increase in shear rate, dilatant fluids, would require testing to assure that the pump could handle the fluid. Some examples of dilatant fluids are clay slurries, various starches, and some paints. Fluids similar to dilatant fluids, rheopectic fluids, experience an increase in viscosity due to a maximum time of agitation. These fluids would also be subject to testing prior to final pump selection.

A fluid with any rheology that results in a viscosity decrease with an increase in shear rate, thixotropic fluids, and pseudoplastic fluids, can offer a successful application of a metering pump, even though the reported viscosity would seem to be too high. In this instance, the shear rate is increased by increasing the velocity of the fluid. As a rule of thumb, a pump may be applied to a fluid of this type of rheology for reported viscosities up to four times the pump’s normal Newtonian limit. Typical examples of thixotropic fluids are molasses and some types of paint. Some fluids known as pseudoplastic fluids are tomato catsup and many polymers.

Temperature Extremes

Fluid temperature requires multiple considerations when selecting and applying a metering pump. Consideration must be given to the design temperature, temperature control (via heating or cooling jackets and insulation), seasonal and daily cycles, and both vapor pressure and viscosity, which are directly related to temperature. Unique pump configurations that remove the “wet end” from the “working end” of the pump are available. One of the many possible remote head configurations is illustrated in Figure 33. With this configuration, temperature extremes of -100°F and $+700^{\circ}\text{F}$ may be handled. This capability is afforded because of the isolation of the temperature extreme from the pump. Hot fluids are pumped with the remote head at an elevation above the pump, and cold fluids are handled with the remote head at an elevation below the pump to limit the transmission of heat. Further, the transition pipe, is sized to capture a volume that is three times the displacement of the pump’s piston. By this design, three temperature “zones” will be approximated even though the fluid is continuously moved both fore and aft within the pipe.

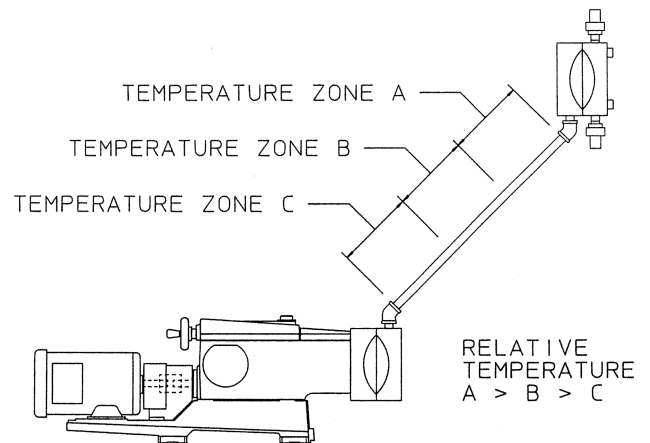


Figure 33. Extreme Temperature Configuration, Remote Head Arrangement.

Corrosivity

The corrosivity of fluids is well documented. A fluid’s ability to attack/degrade the wet end components is usually one of the first

considerations. Correct compatibility between the fluid being handled and all wetted material is important to assure the longest life of the equipment, product integrity, safety, and optimum pump performance. When a corrosion guide is consulted, please remember that it is just a "guide." A "second source" should always be consulted, and the best information is usually available onsite, where the fluid is currently handled. The selection of wet-end materials must be based on both the information from the corrosion guide and an understanding of the corrosive environment, including:

- The effect on corrosion rate of elastomers that are placed in a dynamic condition as opposed to static (for instance, diaphragms as opposed to O-rings).
- Static conditions allow an oxide layer to form on metallic surfaces, which then protects the metal substrate. With moving fluids, this oxide layer may not be allowed to form. It is best to conduct coupon tests with the coupons placed in a pipeline where the fluid is moving, as opposed to placing the coupon in a beaker under static conditions.
- Some chemicals, sulfuric acid for example, may have corrosion compatibility with certain materials, but can, at the same time, be extremely erosive when set in motion.
- Most chemicals become more corrosive at elevated temperatures.

Slurries

Slurries can be successfully handled using hydraulically balanced diaphragm metering pumps. Typically, the solids concentration range is 10 to 20 wt percent. However, colloidal suspensions behave differently and higher concentrations may be handled. The considerations for application of a metering pump include the following:

- A tubular style diaphragm offers the best option, since all areas that may entrap solids are eliminated. Flat diaphragm designs are usually successful as long as the solids do not accumulate inside the pumping chamber at the face of the diaphragm.
- Better metering accuracy can be obtained by using elastomeric seats within the valve assembly. These seats provide an improved sealing surface and better abrasion resistance for some slurries, but do require a limited discharge pressure.
- Moderate stroking rates (88 to 116 spm) should be used to avoid valve chatter and to limit the effects of erosion.
- An average velocity of four to five feet per second is required throughout the pumping system to prevent settling of most slurries. This threshold should be met over the entire planned flow range for the pump. Typically, the minimum required flow is set at 20 percent of maximum. While the velocity should be maintained at a high level, a balance must be given to assure that the high velocity does not reduce the $NPSH_A$ to below an acceptable limit.

Specific Gravity

Moderate stroking rates (88 to 116 spm) should be used for fluids with specific gravities above 1.5. When the fluid is accelerated on either the suction or discharge side, the momentum of the fluid (accounting for the entire mass of the fluid set into motion and the peak velocity of this mass) can be so great as to become irreversible as the pump reciprocates between its discharge and suction cycle. This effect, called "inertial flow-through," will cause greater pump through-put than anticipated and loss of accuracy.

Also, fluids with very high specific gravities may cause the ball in the check valve assembly to float. This often becomes a concern for nonmetallic ceramic or glass balls for fluids of specific

gravities greater than 2.0. In these cases, a heavier metallic ball would be recommended. In extreme cases, tungsten carbide balls may be the only satisfactory choice.

Common Industrial Applications

Amongst the municipal market applications, mainly water and wastewater treatment, a major percentage of the pumps sold in industrial applications are also for water treatment. The water treatment market can be broken down into five major areas of application:

- Process water
- Boiler water
- Cooling tower water
- Potable water
- Waste water treatment

In addition to water treatment applications, the following applications are a major portion of the hydraulic diaphragm metering pump business around the world.

- *Petroleum and Gas Industry*
 - Bacteria control of water for well flooding
 - Chemical desalting of crude oil
 - Caustic soda to crude oils or soured gasoline for rerun through cracking stills
 - Sampling feed stocks, jet fuels, gasolines, intermediates for analysis
 - Sludge inhibitors for fuel oils
 - Metal deactivators (antioxidants) to eliminate gum formations in gasolines
 - Caustic soda addition to adjust pH of sour gas or crude oil to hamper corrosion
 - Acid for polymerization
 - Addition of color to gasoline
 - Jet fuel anti-icing additives
 - Ratioing antifoam additives to lubricants
 - Addition of antiknock components to alcohol or gasoline
 - Pilot plant applications for high and low pressures
 - Addition of odorants (Mercaptan) to gas lines and storage tanks
 - Lubricant addition to gas lines
- *Pulp and Paper Industry*
 - pH control of pulp
 - Addition of titanium dioxide to pulp for opacity control
 - Kaolin slurries as paper filler
 - Automatic emulsification of rosin size
 - Resin addition for waterproofing
 - Ratioing all ingredients in preparation of chlorine dioxide bleach
 - Polymers for drainage and retention aid
 - Metering of colorants and dyes
 - Metering of wire life extenders to paper machine
 - Defoamer addition
 - Wastewater treatment

- *Pharmaceutical and Cosmetics Industry*
 - pH control of broths
 - Nutrient feed in antibiotic manufacturing
 - Sampling of product analysis
 - Ratioing of oils and perfumes for skin and body
 - Scenting powders (talcs)
 - Ratioing ingredients for hair dyes and dressings
 - Metering acetone for fats, oils, resins, and waxes
 - Manufacture of perfumes (Acrolein in alcohol)
 - Carbowax for hair preparations and ointments
 - Diatomaceous earth slurries for filtration
- *Food Industry*
 - Chemical treatment of food process oils prior to centrifuging
 - Blending ingredients in manufacture of mustard, mayonnaise, and salad dressings
 - Oil addition to peanuts
 - Metering vitamins to many food products including milk
 - Adding preservatives to citrus juices, jellies, jams, etc.
 - Addition of flavoring oils to flour, cake, and pie mixes
 - Spraying coatings and flavoring
 - Adding melted nutmeg butter to edible meat by-product
 - Water conditioning and pH control for many processes including wastewater
 - Metering diatomaceous earth slurries for filtration of process and wastewater
- *Other Industries*
 - Breweries
 - Distilleries
 - Wineries
 - Soft drink industry
 - Pet food and cattle industry
 - Mining industry
 - Metal plating and mirror coating
 - Metal recovery
 - Clay refining
 - Sugar refining
 - Textiles
 - Rubber manufacturing
 - Steel, aluminum, and copper mills
 - Glass, ceramics, refractories
 - Tobacco industry
 - Paint manufacturing
 - Printing (metering inks and dyes)
 - Nuclear facilities
 - Utilities (gas and electric)

CONCLUSION

Hydraulically actuated diaphragm metering pumps are usually selected for their capability in delivering fluids at medium to high

pressure, to provide high accuracy, and to transfer the fluid with a leakfree sealless design. These pumps are highly adaptable, but the basic properties of the fluid are of paramount importance in making the proper pump selection. Special consideration must be given to those fluid properties which will affect pump performance—including vapor pressure, viscosity, rheology, temperature extremes, corrosivity, solids content, and specific gravity.

In addition to pump selection dependent on fluid properties, the chemical metering system must be designed with system requirements in mind. Even if hard-to-handle fluid properties prevent the ease of selection of one particular metering pump, the system can be modified to make the application more favorable for the pump. Many metering pump accessories are recommended, and specific accessories are available, to help comply with system requirements. Flow may be controlled via stroke length, stroke speed, or stroke cycle.

With proper consideration given to these important fluid properties and system requirements, one can be assured of the proper selection and successful application of hydraulically balanced diaphragm metering pumps.

NOMENCLATURE

- $NPSH_R$ = Net positive suction head required (psi, bar)
- $NPSH_A$ = Net positive suction head available (psi, bar)
- P_A = Pressure at the surface of the liquid being pumped (atmospheric or supply tank blanket pressure) (psi(a), bar(a))
- P_H = Head pressure above (+) or below (−) the pump centerline (psi, bar) (convert from ft or m), subscripts of D and S stand for discharge and suction respectively
- P_V = Absolute vapor pressure at pumping temperature of the process liquid at pump inlet (psi(a), bar(a))
- L = Length of piping (actual, not equivalent) (ft, m), subscripts of D and S stand for discharge and suction respectively
- R = Pump stroking rate, strokes/min (spm)
- G = Specific gravity of process liquid (no units)
- Q = Pump average flowrate (gph, lph)
- d = Internal pipe diameter (in, mm), subscripts of D and S stand for discharge and suction respectively
- C_1, C_2, C_3 = Numeric constants as specified by pump manufacturer (no units)
- μ = Absolute viscosity of process liquid at pumping temperature, centipoise (cP)
- P_P = System discharge pressure (psi(g), bar(g))
- P_T = Peak pump discharge pressure at the discharge port (psi(g), bar(g))
- V_P = Peak liquid velocity generated by the pump (suction or discharge) (ft/s, m/s)
- ρ = Fluid density of process liquid at pumping temperature, grams per cubic centimeter (g/cm^3)
- ν = Kinematic viscosity of process liquid at pumping temperature, centistokes (cs)
- τ = Shear stress of the process liquid at the pumping temperature, pascals (Pa)
- Γ = Shear rate of the process liquid at pumping conditions, reciprocal seconds (s^{-1})

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