A COMPARISON OF THE APPLICATION OF CENTRIFUGAL AND POSITIVE DISPLACEMENT PUMPS

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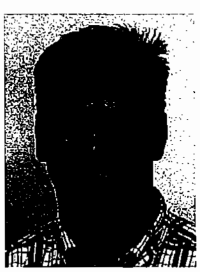

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ABSTRACT

There are very basic differences between the centrifugal and positive displacement pump. This paper discusses some of these aspects. It is not intended to be highly theoretical, but deals with the practical differences that affect the application of the two different types of pumps. Specific designs of centrifugal and positive displacement pumps are not emphasized, but the application traits common and different among the designs are covered. Discussed are theoretical differences in the way fluids are moved within the pump, the relationship between the pump type and the system, failures due to cavitation, performance with viscous liquids, minimum flows, and variable speed drives versus control valves for changing pump performance.

INTRODUCTION

For every centrifugal pump applied there are far fewer positive displacement pumps. Over the last few years this has begun to change and more displacement pumps are in service relative to centrifugal. Yet most individuals sizing and installing positive displacement equipment as yet do not apply positive displacement (PD) pumps often enough to see the difference between the two designs. Through the use of both theoretical and practical examples, this paper will help the reader to better understand the differences in application of the designs. It is an attempt to reduce the field problems seen for both positive displacement and centrifugal pumps.

CENTRIFUGAL AND POSITIVE DISPLACEMENT PERFORMANCE

Centrifugal radial pumps are inherently constant velocity devices. When the impeller and rpm are constant, they create a constant discharge velocity liquid at any flow. The liquid enters the pump at a specific velocity, the impeller accelerates the liquid, the high velocity liquid passes into the volute, then the liquid velocity reduces, and the pressure at the discharge increases. This is somewhat oversimplified, however for the general description it will be adequate. It can be graphically demonstrated in Figure 1. The actual pump curve is not horizontal; it has a constantly decreasing head because of other forces and inefficiency within the pump.

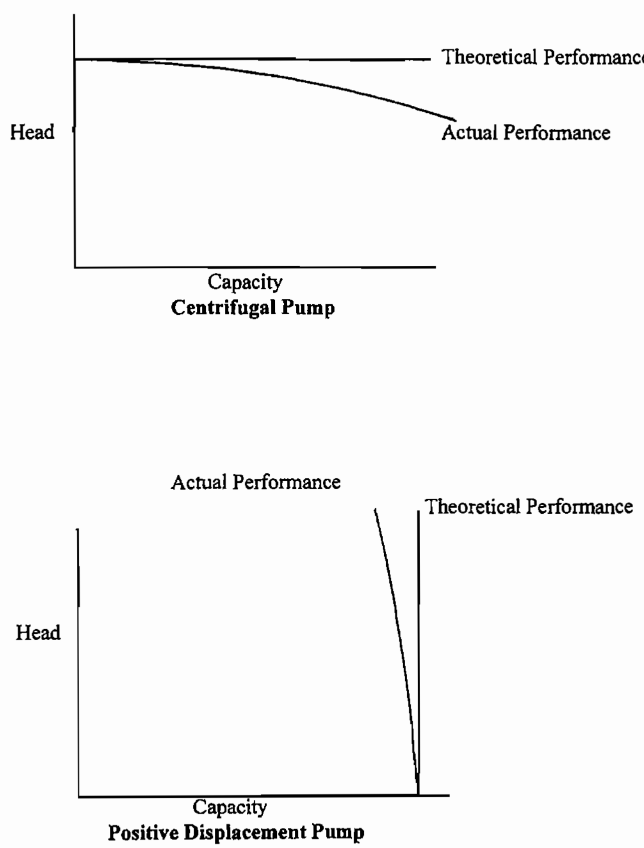

Figure 1. Theoretical Pump Performance.

A positive displacement pump is a constant volume device. The liquid flows into the inlet. It is captured within a confined space and then pushed into the discharge. There is no significant change in velocity with the pump. The pump will displace a given volume of liquid per rpm and is theoretically unaffected by pressure. It can be graphically demonstrated in Figure 1. The actual pump curve is not vertical. It slopes showing a decrease in flow with increasing pressure. This is caused by slip or inefficiency within the pump. The concept of a constant flow per revolution is the reason a positive displacement pump must always be applied in conjunction with a relief or bypass valve. The valve prevents over-pressurization of the system.

THE RELATIONSHIP OF SYSTEM HEAD CURVES TO CENTRIFUGAL AND POSITIVE DISPLACEMENT PUMPS

System head (Figure 2) curves are made up of three components:

- *Pressure head*—The energy (head) required at the discharge of the system. This is normally constant in the simple system; therefore it is a straight line in a graphic presentation.
- *Static head*—The elevation difference between the low level on the inlet of the system and the high level on the discharge of the system. In large systems, this may be a variable component but, in a small system, it is generally constant.
- *Friction head*—The energy (head) required to move the liquid through the piping, fittings, and system components. In many systems, the inlet and outlet may be variable. This example will use a single source for the inlet and a single source for the outlet. To establish the friction loss curve, find a single point on the curve and extrapolate the other points by the formula:

$$\frac{H_{f1}}{H_{f2}} = \left(\frac{GPM_1}{GPM_2}\right)^2 \qquad (1)$$

The three components are then plotted by adding values on the vertical axis. The pump curve will cross the system curve at only one point.

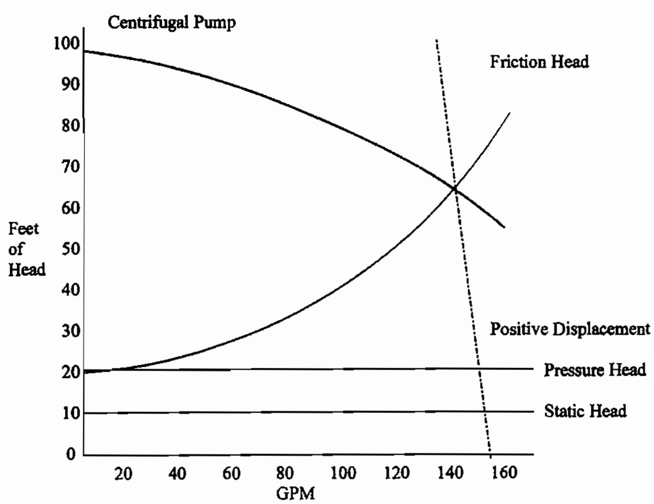

Figure 2. Simple System Head Curve.

Throttling the Pump

The difference between positive displacement and centrifugal pumps is easily demonstrated. Figure 3 shows the effects of a closing throttling valve and the resultant change in the system curve.

- The centrifugal pump can be directly throttled to its minimum flow; however care needs to be taken not to move below the minimum flow of the pump. Note that throttling a centrifugal pump does not always reduce its horsepower.

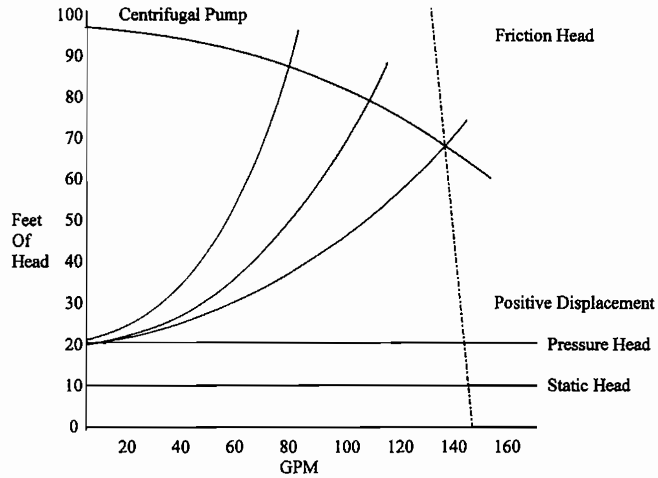

Figure 3. Throttling a Pump.

- A positive displacement pump requires a bypass valve arrangement or a variable speed drive. The bypass allows the pump to continue working at its design flow while returning some of the liquid to the suction of the pump or the storage tank. If the liquid is not bypassed or the pump speed is not reduced, damage will occur to the pump or the system because the PD pump is a constant volume machine.

Parallel Operation (Figure 4)

Centrifugal Pumps (Identical Pumps)

The flow of the two or more pumps is added together and overlaid on the system curve. Note that because there is an increase in friction loss within the system due to the higher flow, the combined flow is not doubled.

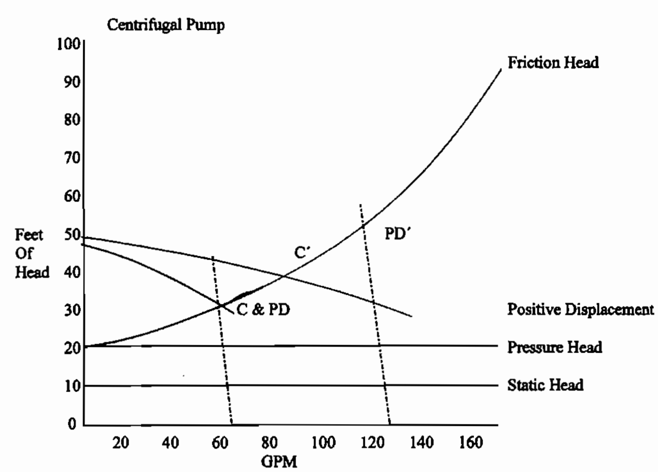

Figure 4. Parallel Pump Operation.

Centrifugal Pumps (Different Pumps)

The flow is again added together and laid on the system curve. Care needs to be taken to ensure that the smaller of the pumps does not run at shutoff.

Positive Displacement Pumps

The addition of the flows is similar to centrifugal pumps. However, PD pumps in parallel will give larger flows because they will inherently compensate for the higher system pressure.

Series Operation (Figure 5)

Centrifugal Pump

The head created by the pumps is added together and the resultant curve is overlaid on the system curve. Care needs to be taken not to exceed the maximum working pressure of the pumps, and the pumps downstream of the first stage must always have liquid flowing to them.

Positive Displacement

It is not necessary to run pumps in series because a PD pump will be self-compensating for system pressure and will only move a specific number at a given rpm. There is no advantage to placing positive displacement pumps in series when they are in close proximity to each other.

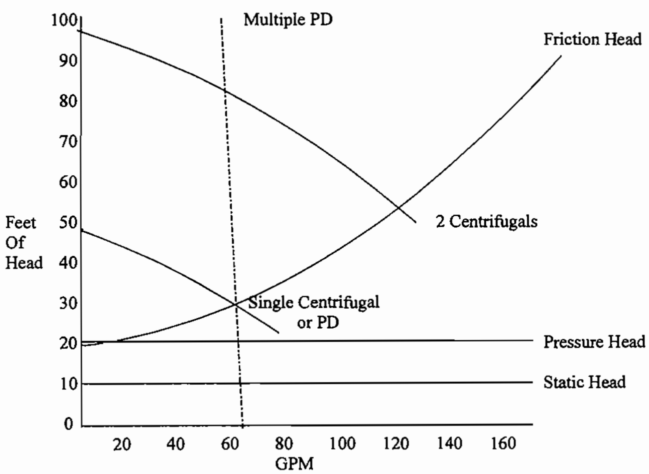

Figure 5. Series Operation.

CAVITATION IN CENTRIFUGAL AND POSITIVE DISPLACEMENT PUMPS

Cavitation is the imploding of a gas bubble within a pump. It manifests itself by creating a noise that sounds like gravel being pumped; there is also reduction in flow, increased vibration, and general failure. The problem can occur in several ways. The most common form of cavitation in both positive displacement and centrifugal pumps is caused by the rotating action of the pump reducing the pressure at the inlet area below the vapor pressure of the liquid. This causes vapor bubbles to form. They move through the pump to areas of higher pressure. The bubbles implode. The energy absorbed by the phase change causes the damage.

The centrifugal pump is attempting to accelerate the liquid through the impeller to some constant velocity. As the pressure within the fluid system decreases, the flow and the net positive suction head required (the inlet energy required by the machine) increase. The inlet and outlet velocity of the liquid increase. It simply takes more energy to push the liquid into the pump because the pump impeller is accelerating more liquid through the pump. When the inlet energy is not sufficient to maintain the liquid phase, vaporization occurs at the inlet of the impeller. As the liquid gas mixture moves through the impeller, the energy level of the mixture increases and the gas reverts to a liquid. This occurs between the impeller vanes but does not generally affect the inlet tip of the impeller vane or the outer tip of the vane, as seen in Figure 6. It is commonly known as a runout cavitation because the pump is running too far out on its curve.

Positive displacement pumps will also cavitate if the inlet flow does not have adequate $NPSH_A$ (energy available). However, because they are not a velocity increasing machine, and they move only a fixed volume of liquid per revolution, the $NPSH_R$ (the amount of energy required) is a single amount and will not vary unless the liquid or the pump speed change. It can be thought of as an $NPSH_R$ point rather than a curve. If this amount is exceeded, the pump will cavitate and there will be a decrease in flow with an increase in noise. The amount of damage is dependent on the specific design of the positive displacement pump. Some designs that have elastomers as major pumping elements and vane pumps will offer excellent life in applications where cavitation is present. An example is the pumping of liquefied gases. Figure 7 is an example of cavitation in a sliding vane pump. Gear pumps will have damage to both the housing and gear set.

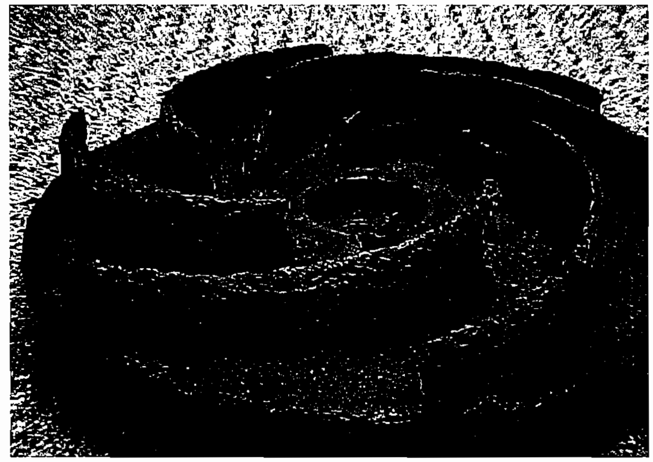

Figure 6. Runout Cavitation of a Centrifugal Pump.

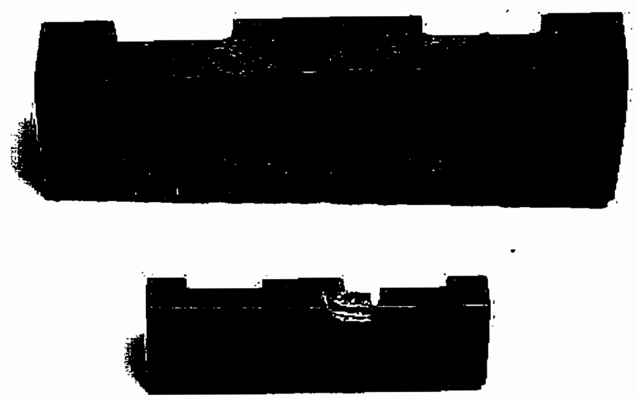

Figure 7. Cavitation of a Vane Positive Displacement Pump.

Some common corrections for cavitation caused by a lack of net positive suction head (available energy) are:

1. Reduce the pump speed,
2. Reduce the size of the centrifugal pump impeller,
3. Partially close the discharge of the centrifugal pump (Note: This will create other problems when using a positive displacement pump),
4. Increase the static elevation of the liquid on the inlet of the pump,
5. Increase the size of the inlet piping to the pump. (This is most effective if the liquid is more viscous than water.)

The first three are reducing the energy required by the pump from the inlet system, and the last two are increasing the energy the system can make available to the pump. In all cases, the available energy from the inlet system is being made greater than the energy required by the pump.

Centrifugal pumps will also cavitate at low flows. This is caused by recirculation within the pump casing. The impeller is accelerating the liquid and, because of high system discharge pressure, the liquid cannot exit the pump casing. The cavitation is caused because the centrifugal pump impeller cannot accelerate the liquid sufficiently to create adequate flow into the discharge system. The energy (force generated by the impeller) is not high enough to allow the pump to move liquid out of the casing. The liquid is forced to recirculate across the cut water of the casing. A low pressure area is created behind the cut water and this causes some vaporization. As the liquid gas mixture continues to move around the casing, the mixture comes under increasing pressure and the gases implode, returning to their liquid state. Thus the cavitation occurs. This discharge cavitation, as it is commonly referred to, will sound like runout cavitation and will cause vibration and pump failure. The damage to the centrifugal pump impeller will occur at the tips of the vanes, as seen in Figure 8. It will also cause the clearance between the impeller and casing to enlarge and will cause excess stress on the shaft and bearings.

Figure 8. Discharge Cavitation of a Centrifugal Pump.

Because the discharge system and not the inlet system cause the problem, the corrective actions for discharge cavitation are the opposite of the corrections for inlet cavitation. The speed or impeller diameters are increased to create more flow. The discharge valves are opened to reduce system pressure. Increase the discharge piping to reduce friction loss and discharge head.

VISCOSITY CORRECTIONS FOR CENTRIFUGAL AND POSITIVE DISPLACEMENT PUMPS

Viscosity greatly affects the flow, head, $NPSH_R$, and horsepower of both centrifugal and positive displacement pumps. The effects are caused by the molecular drag and a velocity change within the centrifugal as the liquid is accelerated through the pump.

Figure 9 graphically demonstrates the effect of increased viscosity on a centrifugal pump. The head and capacity decrease, as does the efficiency. The horsepower increases because efficiency decreases.

In the positive displacement pump, the liquid velocity does not appreciably change. As with a centrifugal, there is an increase in molecular drag; however, it is offset by a reduction in slip (reverse flow of liquid through the clearances within the pump clearances). As an example of one specific PD design, the horsepower at a given rpm may decrease as the viscosity increases from 0.25 to 300 cp due to a reduction of slip. However, with liquids over 300 cp, the horsepower will begin to rise until the torque and pressure limits of the pump are reached. Some PD designs have low viscosity limits (less than 100 cp). The slip becomes so great, the movement of the liquid between the clearances damages internal parts. This is especially true of pumps with fixed clearances.

It is important to remember that the $NPSH_R$ of either a centrifugal or positive displacement pump will increase as the

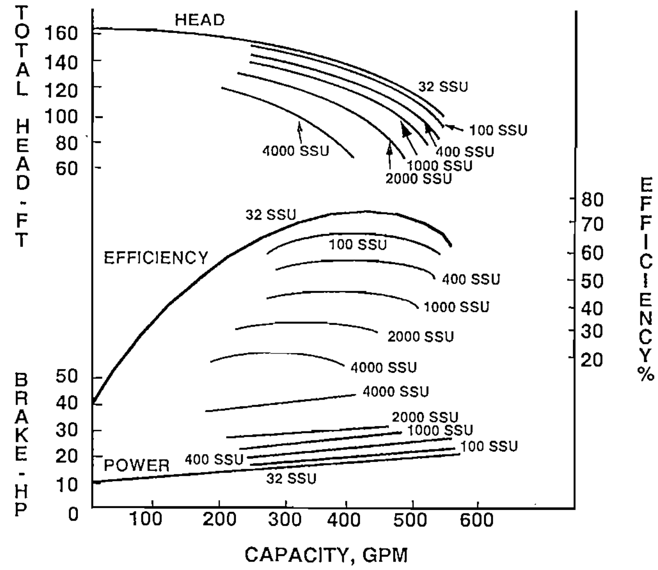

Figure 9. Performance of a Centrifugal Pump at Varying Viscosity.

viscosity increases. This is caused by the increase in resistance to flow by the liquid. The manufacturer of the pump can calculate this $NPSH_R$ increase. Centrifugal pump curves are normally based on water at 1 cp and 1 sp gr. This means that the $NPSH_R$ curve presented in the general pump curve is based on water, not on different viscosity fluids. Positive displacement pump curves are based on many different viscosities and represent a specific flowrate at a given viscosity; therefore, they will tend to state an $NPSH_R$ at a single condition point reflecting a single viscosity, rpm, flow, and pressure.

In centrifugal and nonpulsating positive displacement pumps, acceleration head is not a significant loss in the inlet system because the liquid is moving at a constant velocity within the piping. However, if a reciprocating style positive displacement pump is being applied, the $NPSH_A$ must be carefully considered. The acceleration head of the liquid in the inlet system becomes a significant loss. It can decrease the $NPSH_A$ by a factor of 10. The pulsating motion of the liquid caused by the motion of the plungers will decrease the available NPSH. This decrease varies with the number of plungers, piping configuration, and the type of liquid. For a more complete discussion of acceleration head, refer to Karassik, et al. (1985), or any other text dealing with reciprocating pumps.

Please note that for a specific flowrate, when the viscosity increases, the pump size increases and the rpm are decreased. This causes the torque on the shaft to increase because the hydraulic horsepower remains constant (it requires a given amount of energy to move a liquid within a system) and the rpm of the shaft is reduced. This torque increase needs to be balanced against the $NPSH_R$ of the pump and the $NPSH_A$ of the system and the torque carrying capacity of the shaft. The torque on the shaft can be calculated as follows:

$$T = \frac{(5280)HP}{RPM} \tag{2}$$

where:
T = Torque is in ft lb
rpm = Revolutions per minute
hp = Horsepower

The manufacturer of the pump needs to specify the maximum torque load for the pump shaft and the specification must exceed the torque generated by the application taken into account the $NPSH_R$ of the pump verses the $NPSH_A$ of the system. If these considerations are not made, the result is a broken shaft or cavitating pumps.

MINIMUM FLOW

There are several aspects to minimum flow for both centrifugal and positive displacement pumps. Because centrifugal pumps are dynamic velocity machines, they are subject to radial load increases in single volute designs. The standard (single) volutes are normally used below 400 gpm in standard designs. Typical of these type pumps are end suction ANSI B73.1 pumps and small water pumps. When they create flows other than at their best efficiency point (BEP), the radial loads on the shaft rise, as seen in Figure 10. As the pump flow is reduced (the pump moves to shutoff), the radial load increases and the $NPSH_R$ is reduced until the pump begins to recirculate liquid. The inlet characteristic then becomes unstable. These phenomena can break shafts, and cause seal and bearing failure. Using a pump at shutoff can also cause a significant temperature rise in the liquid, especially liquid with a low specific heat.

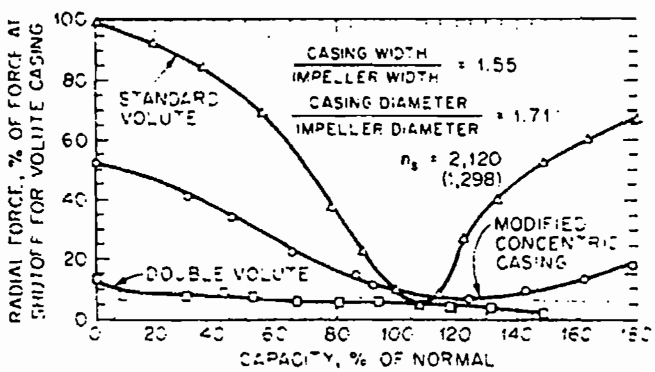

Figure 10. Radial Load on a Centrifugal Pump Shaft.

Mechanical Minimum Flow

For single-stage centrifugal pumps with less than three inches discharge, the suggested mechanical minimum flow is 10 percent of best efficiency point. For single-stage centrifugal pumps with more than four inches discharge, the suggested mechanical minimum flow is 25 percent of best efficiency point. For all other designs, please refer to the manufacturer. They need to supply both the thermal and mechanical minimum flow characteristic for the individual pump.

Positive displacement pumps are constant flow devices and require system pressure relief, bypass valves, or some sort of torque limiting equipment. As the flow requirement of the system is reduced, the liquid being pumped by the PD must be recirculated or the pump rpm must be reduced. If these measures are not taken, damage will occur to the system. If a bypass is used, temperature rise needs to be taken into consideration and an adequate heatsink needs to be available.

Temperature Rise

Under steady-state conditions, friction and the work of compression increase the temperature of the liquid as it flows from suction to discharge. A further temperature increase may arise from liquid returned to the pump suction through wearing rings, a balancing device, or a minimum-flow bypass line that protects the pump when operating at or near shutoff.

Assuming that all heat generated remains in the liquid, the temperature rise, ΔT, in °F is Karassik, et al. (1985):

$$\Delta T = \frac{H}{788 C_p \eta} \tag{3}$$

where:

H = Total head, ft
C_p = Specific heat of the liquid, Btu/lb °F
η = Pump efficiency, decimal value

PIPING AND MOUNTING OF CENTRIFUGAL AND POSITIVE DISPLACEMENT PUMPS

The general method for piping centrifugal and positive displacement pumps is the same. However, care needs to be taken in the calculation of friction losses as the range of viscosity increases. There are several methods of calculating friction loss. One of the most widely used is Darcy-Weisbach (Fanning formula) and is stated as:

$$h_f = f \frac{L}{D} \frac{V^2}{2g} \tag{4}$$

where:

h_f = Friction loss, ft of liquid
L = Pipe length with fittings, ft
V = Average pipe velocity, ft/sec
D = Average inside diameter of pipe, ft
g = Gravitational constant, 32 ft/sec^2
f = Friction factor, predetermined number

The Darcey-Weisbach formula takes into account the condition of the pipe, the inside diameter, the velocity of the liquid, and the viscosity. The friction factor can be calculated using the Moody Diagram, shown in Figure 11.

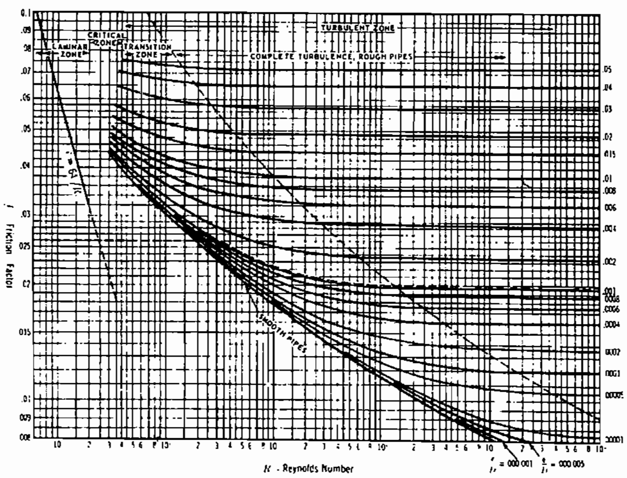

Figure 11. Friction Factors for Piping Line Losses. (Courtesy McGraw-Hill (Streeter, 1971))

There are standards that need to be followed when piping all pumps. Some suggestions are the Hydraulic Institute Standards as well as several American Petroleum Institute Standards. Care needs to be taken with all equipment to reduce nozzle (flange) loads to their minimum.

There are some different considerations for the design of coupling and baseplates used for the mounting of centrifugal and positive displacement pumps. PD pumps do not use more horsepower than centrifugals; however, they do turn at lower rpms. Torque is a function of speed (rpm). The lower the rpm at a given horsepower, the higher the torque. Therefore there are increased stresses transmitted to the slow speed coupling and the base. In some applications, the foundation and base will need reinforcement between the low speed shaft of the drive and the pump. A weak base and foundation will cause misalignment of the pump and drive components as well as failure of couplings and accessories. There are several excellent references on base design: the Hydraulic Institute, API, and Blodgett (1963). Care needs to be taken when sizing the low speed coupling. Simply sizing it for the horsepower will cause the selection of too small a coupling. It needs to be selected for the torque being transmitted.

SAMPLE CALCULATIONS

Variable Speed Pump with Constant Impeller Diameter, Constant Speed Pump with Control Valve, and Positive Displacement Pump

Different types and styles of pumps will react differently when controlled by valves or variable speed drives. In the case of a centrifugal pump with a very steep performance curve, typical of a vertical turbine, the horsepower saving will tend to be much greater using a variable speed drive rather than a control valve. For a standard end suction pump, the savings will still be significant but not as great. For a positive displacement pump, there is only a horsepower savings when a variable speed is used. If the system is controlled by a valve, there is zero savings because the excess discharge liquid must be bypassed back to the inlet of the pump. The formula for the brake horsepower of a centrifugal pump is:

$$\text{Brake Hp} = \frac{(\text{gpm})(\text{Ft of Hd})(\text{S.G.})}{3960(\text{Efficiency})} \tag{5}$$

The formula for the brake horsepower of a positive displacement pump is:

$$\text{Brake Hp} = \frac{(\text{gpm})(\text{psi})}{(1714)\text{Efficiency}} \tag{6}$$

- Example 1: 6 × 8 Vertical turbine three-stage pump
  - Given: 1. Liquid: Solvent @ std. conditions
    2. Pump gpm: Max = 1600, min = 500
    3. TDH in ft: Max = 450, min = 150
  - Find: 1. Variable speed pump
    Bhp for 1600 @ 450
    Bhp for 500 @ 150
    2. Control valve
    Bhp for 1600 @ 450
    Bhp for 500 @ 150
    3. Bhp difference
  - Solution:
    Variable speed drive
    Curve 1: 3450 rpm — Bhp for 1600 @ 450 = 242
    Eff. = 75%
    Curve 2: 1760 rpm — Bhp for 500 @ 150 = 31
    Eff. = 61%

    Control valve
    Curve 1: 8.25 inch impeller @ 3450 rpm
    Bhp for 1600 @ 450 = 242
    Eff. = 75%
    Bhp for 500 @ 150 = 271
    Eff. = assumed 30

    Variable speed Bhp savings: 242 − 31= 211 hp savings
    Control valve Bhp savings: 242 − 271= −29 hp (this is an increase)

    Please note: In Example 1, the efficiency increase for three bowls was not taken into effect.

- Example 2: 1 × 1.5 × 8 End suction, single-stage pump
  - Given: 1. Liquid: Solvent @ std. conditions
    2. Pump gpm: Max = 160, min = 40
    3. TDH in ft: Max = 282, min = 76
  - Find: 1. Variable speed pump
    Bhp for 160 @ 282
    Bhp for 40 @ 76
    2. Control valve
    Bhp for 160 @ 282
    Bhp for 40 @ 76
  - Solution:
    Variable speed drive
    Curve 3: 3500 rpm — Bhp for 160 @ 282 = 16.75
    Eff. = 68%
    1750 rpm — Bhp for 40 @ 76 = 1.7
    Eff. = 45%

    Control valve
    Curve 4: 8 inch impeller @ 3500 rpm
    Bhp for 3500 @ 160 = 16.75
    Eff. = 68%
    Bhp for 40 @ 325 = 9.1
    Eff. = 36%

    Variable speed Bhp savings: 16.75 − 1.7 = 15 hp savings
    Control valve Bhp savings: 16.75 − 9.1 = 7.6 hp savings

    The primary difference between the two examples is that Example 1 is a low specific speed impeller and Example 2 is a high specific speed impeller.

- Example 3: Positive displacement pump
  - Given: 1. Liquid: Solvent @ 500 cp
    2. Pump gpm: Max = 160, min = 40
    3. TDH in ft: Max = 282, min = 76
  - Find: 1. Variable speed pump
    Bhp for 160 @ 282
    Bhp for 40 @ 76
    2. Control valve
    Bhp for 160 @ 282
    Bhp for 40 @ 76
    3. Bhp difference
  - Solution:
    Variable speed drive
    Curve 5-P: 463 rpm — Bhp for 160 @ 282 (122 psi) = 14
    Curve 5-1: 100 rpm — Bhp for 40 @ 76 (33 psi) = .2

    Control valve with bypass Bhp for 160 @ 282 (122 psi) = 14
    Curve 5-P: 463 rpm — Bhp for 40 @ 76 (33 psi) = 14

    Variable speed Bhp savings: 14 − .2 = 13.8
    Control valve Bhp savings: 0

In the positive displacement example, the control valve horsepower never changes. All positive displacement pumps move a fixed volume of liquid per revolution. A bypass system is required to use a control valve; therefore the pump is always pumping the same volume, pressure, and the same horsepower.

In all examples, the period of time for which the pumps will be running at specific condition points is not known. Nor is the cost of energy. Both these variables will affect the overall pumping cost and the selection of the drive type. The choices of a variable speed drive versus a control valve needs to be made on a case by case basis (Figures 12, 13, 14, 15, and 16).

SELECTION PROCEDURE FOR POSITIVE DISPLACEMENT AND CENTRIFUGAL PUMPS

There are several different diagrams for the general selection of pumps based on flow and discharge pressure. Figure 17 is typical of these.

Note that a majority of the flow can be done with a positive displacement (rotary), centrifugal, or reciprocating pump. Because of this, the selection process must take into consideration many different parameters. Certainly the physical properties of the system are important:

- Flow required
- Suction condition
- Discharge pressure
- Net inlet pressure or net positive head available
- Temperature

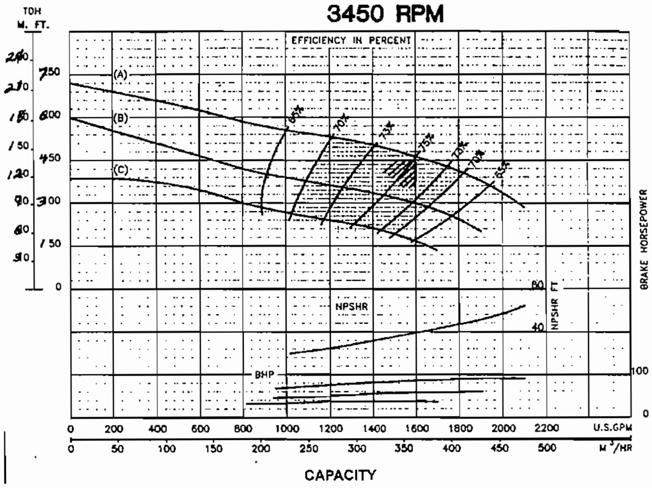

Figure 12. Curve 1 Turbine.

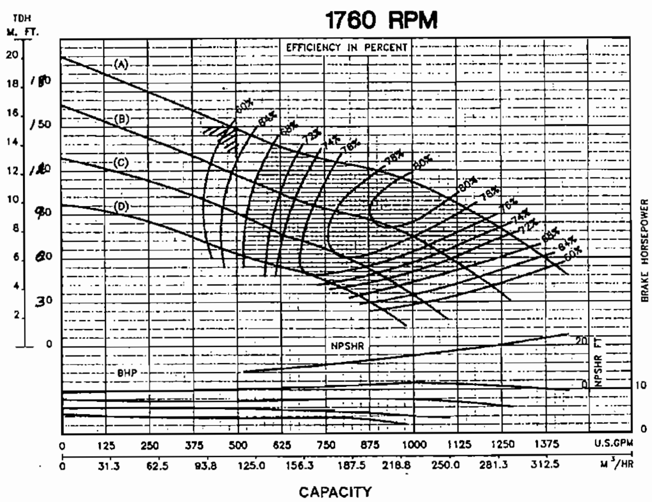

Figure 13. Curve 2 Turbine.

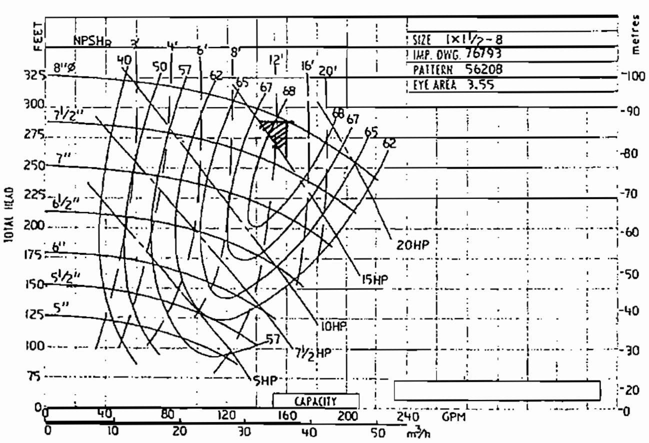

Figure 14. Curve 3 End Suction 3500 RPM.

- Liquid properties
  - Name
  - Vapor pressure
  - pH
  - Abrasive nature
  - Percent of solid by weight or volume

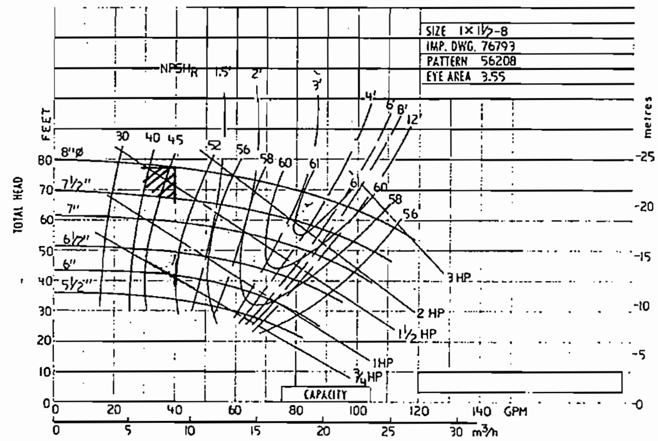

Figure 15. Curve 4 End Suction 1750 RPM.

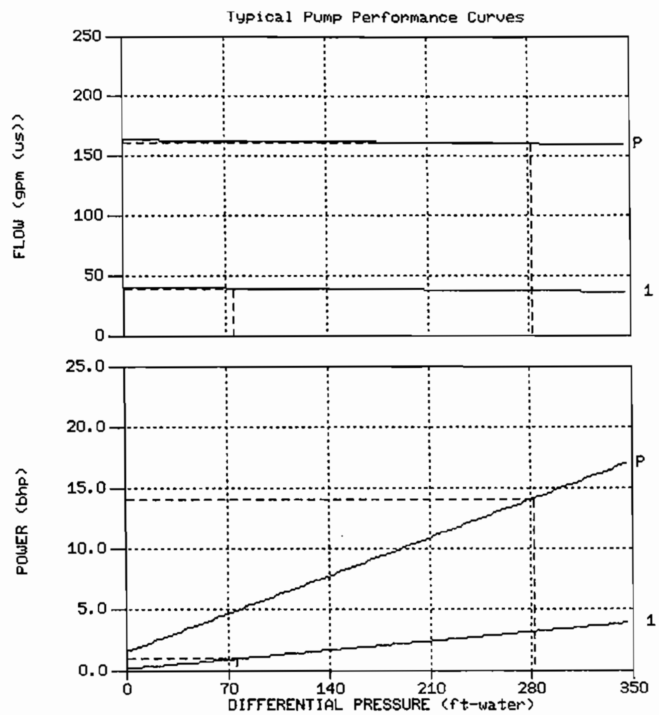

Figure 16. Curve 5 Positive Displacement.

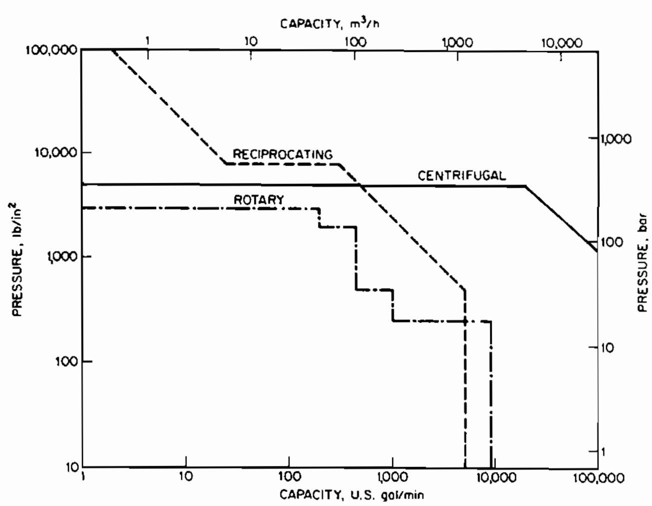

Figure 17. General Selection of Pumps.

- Specific gravity or weight per volume
- Viscosity
  - At pumping temperature
  - At rest
  - At flow condition
- Liquid description

- Duty cycle
- Operating environment
- A complete description of the drive required

The environment of the applications and the local support from supplier and engineering firms need to also be considered. Pump selection is not a matter of simply looking at a few application parameters. The total cost of ownership needs to be addressed. This includes total energy cost, capital cost, maintenance, and cost of money to name but a few. It is important for all groups involved, including the vendor, to work together to create the best overall system. Not necessarily the least costly system, but one with the lowest total cost over the life of not just the equipment, but of the application.

CONCLUSION

Many of the concerns are the same for either type. The inlet and piping consideration are similar. Their control is different. There are different considerations when mounting the pumps on baseplates. Some PD pumps are more tolerant of cavitation then centrifugal pumps. Although centrifugal and positive displacement pumps both have their peculiarities; they are both easily applied if the basic way they move liquid is always considered. Simply remembering that a centrifugal pump is a constant pressure device at any flow that accelerates liquid through an impeller, and that a positive displacement pump is a constant volume device that is always attempting to move the same amount of liquid at any pressure will help to eliminate most application errors.

REFERENCES

Blodgett, O. W., 1963, *Design of Weldments*, Cleveland, Ohio: James F. Lincoln Arc Welding Foundation.

Karassik, I. J., Krutzsch, W. C., Fraser, W. H., and Messina, J. P., 1985, *Pump Handbook*, Second Edition, New York, New York: McGraw Hill.

Streeter, V. L., 1971, *Fluid Mechanics*, Fifth Edition, New York, New York: McGraw-Hill.

BIBLIOGRAPHY

ANSI/HI 1.3.1, 1997, "Centrifugal Pumps—Horizontal Baseplate Design," Hydraulic Institute, Parsippany, New Jersey.

API Standard 610, 1995, "Centrifugal Pumps for Petroleum, Heavy Duty Chemical and Gas Industry Services," Eighth Edition, American Petroleum Institute, Washington, D.C.

API Standard 676, 1994, "Positive Displacement Pumps—Rotary," American Petroleum Institute, Washington, D.C.

Blackmer, 1987, "Hydraulic Data for Pump Applications Number 33," Grand Rapids, Michigan.

Hydraulic Institute, 1997, "Hydraulic Institute Standards," Parsippany, New Jersey.

Ingersoll-Rand, 1995, *Cameron Hydraulic Data Book*, Eighteenth Edition, Woodcliff Lake, New Jersey.

Ingersoll-Rand, 1962, "Cameron Pump Operators' Data," Woodcliff Lake, New Jersey.