Pump Primer

A series of articles on pump operations, applications, and environment

By John H. Horwath
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Company

PUMP PRIMER

The Pump Primer is a series of 20 articles on pump operations, applications and environment. If you would like a copy, or copies, of any of these articles please contact our inside sales staff at (414) 643-1852

Pump Primer

Basic Information About Centrifugal Pumps

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by John H. Horwath
Chief Engineeer, Ampco Pumps

The Pump Primer will be issued quarterly, beginning with this introductory issue, to provide engineering, maintenance and technical purchasing personnel with a basic understanding of pump performance and application.

Topics to be covered include:

1. Introduction — A Pump Odyssey
2. Pump Curves
3. NPSH
4. Cavitation
5. System Curves
6. Practical Limits
7. Energy Savings
8. Dynamic Sealing
9. Maintenance
10. Corrosion
11. Noise
12. Motors
13. Variable Speed
14. Viscosity
15. Friction Tables
16. Erosion
17. Pump Curves II
18. Bearing Loads
19. Electrical Wiring
20. Instrumentation

The intent of these articles is to develop a single, concise source of information combining theoretical and state-of-the-art approaches to typical pump problems. The articles are intended to provide an alternative to the typical ambiguous handbook route commonly used as a first source by personnel not in everyday contact with centrifugal pump applications.

Background

Essentially, there are two basic types of pumps... positive displacement and kinetic. Reciprocating and rotary are the two most common styles of positive displacement pumps while centrifugal and regenerative turbine are the most common in the kinetic group.

Centrifugal pumps may be further classified as: overhung impeller, impeller between bearings or turbine types.

A centrifugal pump is defined as a kinetic machine converting mechanical energy into hydraulic energy through centrifugal action.

While the primary pump under discussion throughout this series will be an over-hung, end-suction, volute casing, single-stage unit, the basic concepts are applicable to most units in the centrifugal category.

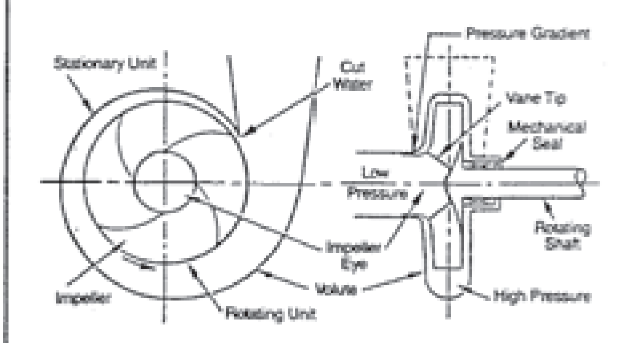
To acquaint you with the basic workings of a centrifugal pump, we invite you to accompany us on a "Pump Odyssey."

A Pump Odyssey

Before beginning this journey, it may be desirable to briefly discuss the major parts and specific functions of a typical centrifugal pump.

Basically, there are two sections to a centrifugal pump...the rotating unit, which develops the necessary liquid velocity and the stationary unit, which directs and contains the flow while converting the developed velocity energy to pressure energy.

The rotating arrangement consists of a shaft, the rotating portion of a shaft sealing device, and an impeller, which is the "heart" of the pump. An outside mechanical energy source (usually delivered from an electrical motor) is used to rotate the pump shaft. The impeller, firmly positioned on the shaft, is equipped with blades which move the liquid, thereby converting the rotating mechanical energy to liquid energy.



Schematic views of pump components and pertinent regions.

The stationary portion of a pump provides an entry path into the eye of the rotating impeller. On leaving the impeller, the liquid enters a spiral chamber known as a volute. It is in this chamber that the transformation from velocity to pressure energy occurs. The volute also provides a specific exit path for the fluid.

Once developed, fluid pressure must be contained and delivered to the pump discharge. To prevent liquid from returning to the low pressure region in the suction area, the impeller suction hub runs in close proximity to a stationary bore over a depth which will effectively hold a pressure gradient and reduce leakage to no more than a small percent of the total flow.

The shaft entry through a stationary wall in the pump is typically handled by a mechanical sealing arrangement. Consisting of a stationary element in direct contact with a rotating element, the seal assures a leakage rate of no more than a few drops per minute.

Taking a trip through a centrifugal pump may be considered synonomous to "riding the rapids." There are potential dangers, obstructions and possible setbacks to contend with throughout the journey.

Starting to Move

Our odyssey begins in the suction piping just outside the pump entrance. We are essentially in a motionless state until the pump drive is turned on. As the impeller begins to rotate, we find ourselves being "pulled" into the eye of the impeller. (Actually, we are being pushed in by an external force as the impeller creates a reduced pressure condition which induces continuous liquid flow into the impeller eye.)

Moving toward the eye, our speed remains relatively low (typically, under 10 feet per second). On approaching the eye, we begin to experience a gradual swirling motion as, following the laws of nature, we seek to move along the path of least resistance to and through the impeller passageways.

Danger Ahead

The primary cause of pumping problems occurs as we approach the impeller eye. All too often, owing to a local pressure drop, cavities filled with liquid vapor are formed. These conditions have a detrimental effect, restricting and varying flow, thereby causing the pump to vibrate and become noisy in operation. Elimination of this condition may be achieved in several ways. However, providing a higher absolute pressure at the impeller eye and a smooth approach free of obstructions in the most practical method.

Assuming that we are operating under a condition providing an adequate suction "push," our ride through the impeller will be relatively smooth. In the impeller eye, our basic direction begins to change from axial to radial. The transition, hardly noticeable at first, changes quite dramatically when we are suddenly picked up on the leading inlet side of one of the impeller vanes. We are now in a rather confined passageway, traveling at what appears to be approximately the same speed at which we entered the pump. Something is changing, however, as we radially slide outward on the backside of an impeller vane. Suddenly, we note the end of the impeller channel and a wider, deeper stationary passage beyond. On reaching the tip of the impeller blade, we are hurtled into the stationary passage at a high rate of speed (typically, 100 feet per second).

During this segment, we have seen mechanical energy transformed into hydraulic energy as the rotating impeller moves liquid at high absolute velocity.

Several other members of our group, who were in similar passageways of the impeller, have rejoined us. Our movement continues in an expanding circular direction in the progressively larger channel of the stationary unit.

Now, a noticeable reduction in our speed becomes apparent as we feel a squeeze caused by the increasing fluid pressure. In the course of less than one revolution, we find ourselves approaching the discharge end of the pump. We pass one last *major obstruction — the* cut-water, which separates most of the rotating flow away from the impeller and into the straight conical section of the volute. As in the spiral passageway, the conversion of velocity to pressure continues in the volute's conical section.

Again, we find ourselves separated from many of our companions. Some were recirculated back to the low-pressure section of the pump through the wearing ring clearance after having traveled through the impeller. Others, in close proximity to the rotating impeller, weren't in position to be diverted at the cut-water and circulated around again until they moved out further in the volute stream. Still others were diverted into the mechanical seal area where they helped lubricate and cool the mating sealing surfaces.

Into the System

Fortified with hydraulic energy, we are capable of continuing our journey to some point beyond the pump through a series of obstructions known as the "system" until our energy has been expended. There are an infinite number of system variations which a pump can handle within its defined parameters of pressure, flow and suction capabilities. The actual capacity delivered by the pump is as dependent on the system as on the pump itself.

Ultimately, the purpose of a centrifugal pump is to move a volume of liquid from its source to another location at the user's desired rate of flow.

The next Pump Primer topic will cover the development and scope of pump performance curves. . . and how they are used to correct for different speeds and impeller diameters. Also, how they provide an insight to the pump's capabilities and, yes, even to the impeller's hydraulic profile.

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Pump Curves

Nowhere will you find a more concise overview of an engineered product's capability than on a centrifugal pump's "oak tree" performance curve. The infinite amount of information in a single graph is astounding!

While the intent here is to focus on the pump's basic hydraulic capability, reference in several future articles to these curves will allow establishment of additional parameters essential to good pump application practices.

It must also be recognized that the centrifugal pump's point of operation at a particular speed is developed in conjunction with one or more factors outside of the pump's own performance capability. These factors play a role equal to that of the rotating pump in establishing the actual flow rate and include: (1) the physical suction conditions present, (2) the physical characteristics of the liquid being pumped, (3) chemical reactions occuring in the liquid being processed, (4) the actual head requirements of the system at specific flow rates and (5) the fluid temperature.

Pump Curve Development

Having learned how a centrifugal pump operates in the previous issue of "pump primer," we can now discuss a pump's hydraulic capabilities and how they are presented.

As defined previously, a centrifugal pump is basically a kinetic machine. The pressure it develops is primarily a function of peripheral velocity established by the rotating speed and diameter of the impeller.

Actually, a pump doesn't directly develop pressure but, based on a velocity relationship, produces a "head" in feet. The head remains the same for all liquids with the same viscosity. The head can be transformed into a pressure term utilizing the formula:

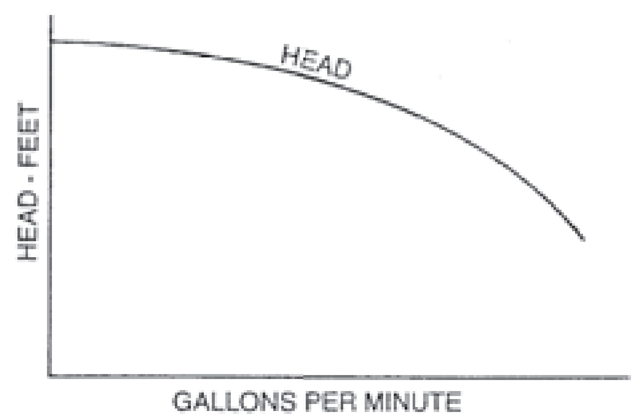
$$\text{Pressure (lbs./sq. in.)} = \frac{\text{Head} \times \text{Specific Gravity}}{2.31}$$

It becomes apparent that the pressure which a pump develops depends on the weight per unit volume of the liquid being pumped. The capacity delivered by a pump is directly proportional to the area of its passages and the velocity of the liquid through these passages.

A most important factor regarding capacity is a pump's suction capability usually expressed as NPSHR (Net Positive Suction Head Required) which will be covered in a future article.

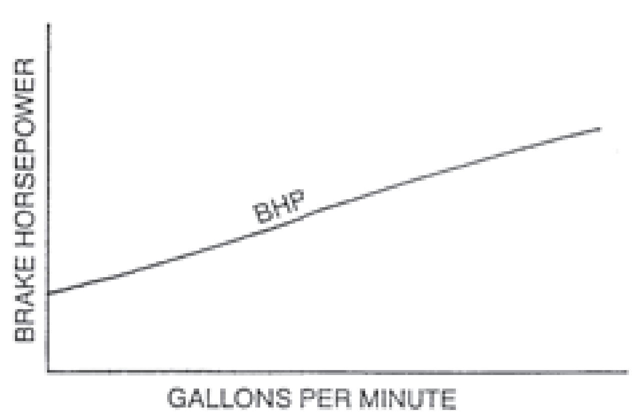
A pump curve graphically defines the pump's hydraulic capabilities in terms of capacity (usually gallons per minute) and developed head (usually expressed in feet of liquid). The graph also indicates the pump's drive requirements relative to speed (revolutions per minute) and power (brake horsepower) based on handling water. In addition, a complete pump curve will indicate efficiency of the unit.

The most common shape of the total developed head versus capacity characteristics curve for a conventional design volute centrifugal pump is the constant rising characteristic shown below.



The maximum head is usually developed at zero capacity — commonly referred to as "shut-off." As the capacity increases, the head decreases. The factors responsible for this are: (1) the direct proportion energy gradient drop necessary to produce flow and (2) the friction loss through the pump passageways.

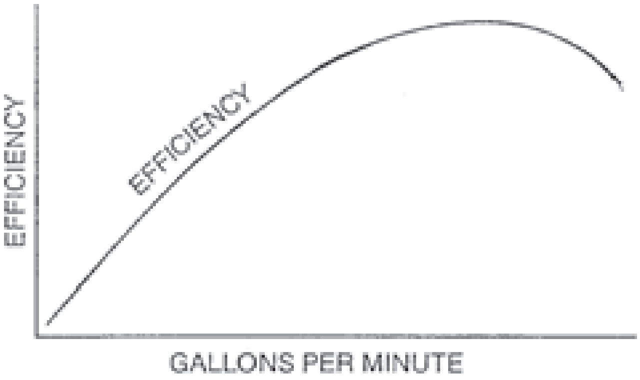
Another commonly provided characteristic is the power curve indicating the pump's actual brake horsepower (BHP) requirements relative to capacity.



From these two curves we can determine the pump's efficiency using the formula:

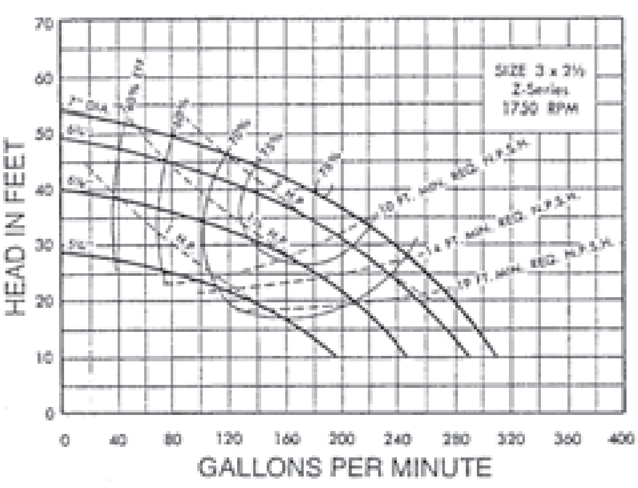
$$\text{\% Pump Efficiency} = \frac{\text{GPM} \times \text{HD} \times 100 \times \text{SP. GR.}}{3960 \times \text{BHP}}$$

This, in turn, allows us to develop an efficiency versus capacity curve.



After having graphically established the pump's performance for a maximum impeller diameter, the exercise is normally repeated for several smaller incremental impeller diameters, each utilizing a maximum brake horsepower requirement at 1.0 specific gravity (water) equivalent to nominal full-load motor horsepower. The resulting data is incorporated on a single graph to present a complete picture of a pump's hydraulic capability and power requirements.

We are now in a position to evaluate a pump's performance to meet specific hydraulic requirements.



To reduce the number of variables, each pump performance is corrected to its particular design speed. Characteristics are usually established based on handling cold, clear water. Other liquids with properties differing from 68° F water, such as specific gravity, viscosity, temperature and the presence of solids combined with abrasiveness, have varying effects on a pump's hydraulic performance and power requirements. Most effects can be reasonably predicted by theoretical or empirical means. Others must be sorted out by actual test of the fluid medium.

Test Specifications

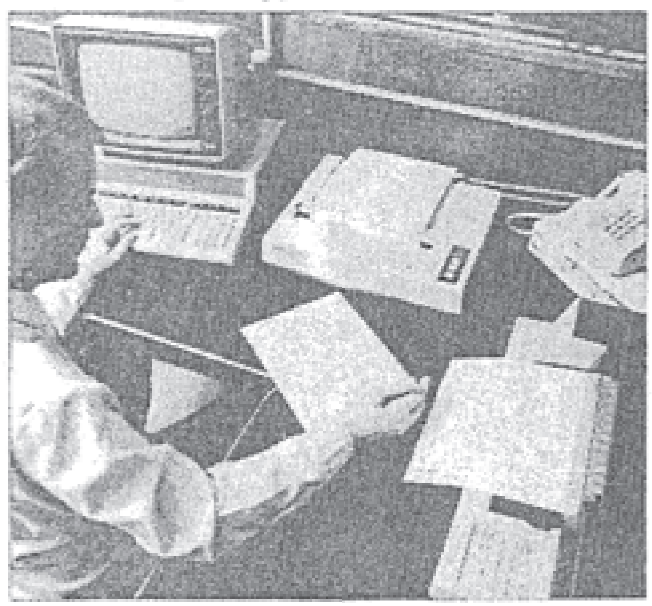
Several test standards are available for checking pump performance and operation. Currently, the Hydraulic Institute Standard is the most widely used general standard for centrifugal pumps. In fact, other specifications, in most cases, will invoke the Hydraulic Institute Standard for test purposes. Performance curve results are normally based on handling 68°F water at a specific weight of 62.3 pounds per cubic foot.

While new casting technology has improved the repeatability of hydraulic performance, discrepancies in general casting and machining practices result in sometimes significant variances in individual curves from the developed standard. Hence, the following statement is indicated on pump curves: "Curves show approximate characteristics - rated point is guaranteed."

Pump manufacturers are normally conservative in the performance rating provided on a general curve and will generally slightly oversize an impeller to make sure it falls within the tolerance requested by the customer for the specific point of operation. Other sections of a specification may even spell out a tolerance for the entire operating range - this is usually limited to Military Specs or unique specification contracts.

Calibration Testing

Ampco's pump testing facilities include a highly accurate, automated system for calibration testing to provide data for plotting performance curves.



The system is equipped with instrumentation traceable back to the National Bureau of Standards. Overall accuracy is ± 1%. Each transducer in the calibration testing equipment sends some form of electric input to a data logger for conversion into engineering units.

A computer is programmed to accept this data and calculate GPM, Total Head, Brake Horsepower and Efficiency which is then fed to a plotter for graphic presentation.

Pump Selection

Ideally, a pump selection should be made to utilize the B.E.P. (Best Efficiency Point) for the intended point of operation. In doing this, the following benefits are derived:

1. In utilizing the highest efficiency for a hydraulic requirement, a minimum amount of energy will be expended — a important plus in today's cost-conscious market.
2. Higher efficiency indicates smoother liquid flow through the pump. Lower efficiency usually indicates energy is being wasted in overcoming hydraulic obstructions, which can adversely affect pump life through erosive action.
3. The degree of mechanical loading imposed on a volute-type pump shaft and bearing is minimal at the pump's B.E.P. capacity. However, due to hydraulic inbalance in the volute, acting on the impeller profile, a substantial resultant radial force can be developed essentially from 50% of B.E.P. capacities back to shut-off. Capacities extending 25% beyond the B.E.P. capacity are also a source of concern as the developed loads can cause excessive shaft deflection and premature bearing failure. Secondary effects may include mechanical seal leakage and wear ring rubbing.

Affinity Laws

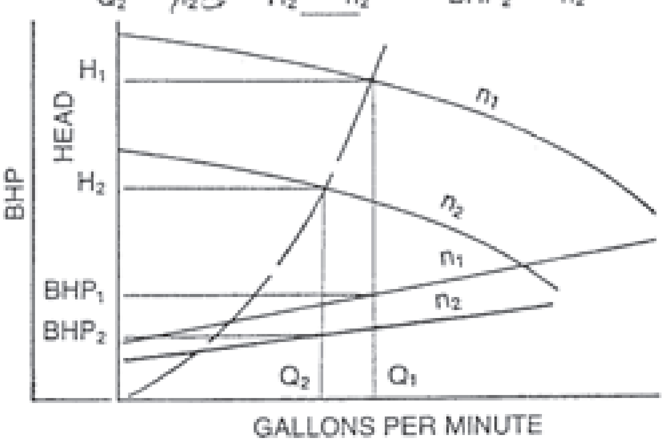
Pump head and capacity vary with speed in such a way that the performance curves retain their characteristic features. The variation of head, capacity and brake horsepower with speed follows definite rules known as affinity laws.

When applied to every point of the head capacity curve, these laws state:

> When speed is changed, capacity varies directly with speed; the head varies directly as the square of the speed; and the brake horsepower varies directly as the cube of the speed.

The affinity laws are expressed by the following formulas:

$$\frac{Q_1}{Q_2} = \frac{n_1}{n_2} \qquad \frac{H_1}{H_2} = \frac{n_1^2}{n_2^2} \qquad \frac{BHP_1}{BHP_2} = \frac{n_1^3}{n_2^3}$$



The affinity laws can also be applied to impeller diameter in the same manner. However, this is normally limited as, due to the cut impeller, some error is introduced, though the peripheral velocities developed would be the same. Normally, the performance reduction would be considered fairly accurate up to a 10% change in diameter from an actual test diameter.

Since the peripheral speed is the same whether achieved by reduced speed or reduced impeller diameter, the affinity laws will hold here as well. Therefore:

$$\frac{Q_1}{Q_2} = \frac{n_1}{n_2} = \frac{d_1}{d_2} \qquad \frac{H_1}{H_2} = \frac{n_1^2}{n_2^2} = \frac{d_1^2}{d_2^2} \qquad \frac{BPH_1}{BPH_2} = \frac{n_1^3}{n_2^3} = \frac{d_1^3}{d_2^3}$$

With these relationships, we are able to forecast pump performance quite accurately from established data over a limited range.

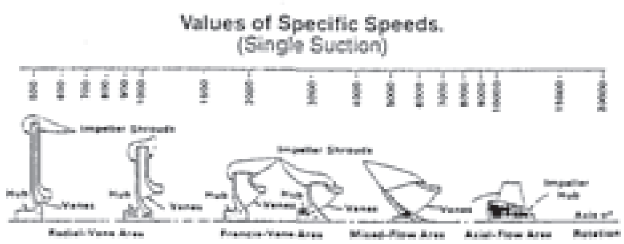
Specific Speed

Pumps are classified by a term referred to as specific speed:

$$N_S = \frac{RPM \times GPM^{1/2}}{HD^{3/4}}$$

Specific speed is the rotational speed at which a theoretical impeller will deliver 1 gpm at 1 foot head. All impellers with a homologous or identical shape, regardless of size, will have the same specific speed. The performance characteristics of any impeller can be applied to any other identically shaped impeller by use of the specific speed formula.

The general shape of head-capacity curves is affected by specific speed. Impeller profiles are defined by specific speed. N_S is independent of pump size but indicates the general shape of the impeller. Assuming proper design, hydraulic performance characteristics affecting pump applications fall into general patterns that can be indexed to specific speed as indicated below in the comparison of pump profiles.



COMPARISON OF PUMP PROFILES

Profiles of several pump impeller designs ranging from the Low Specific Speed Radial Flow on the left to a High Specific Speed Axial Flow on the right, placed according to where each design fits on the Specific Speed Scale.

From Hydraulic Institute Standards

A pump's probable efficiency range can also be related to pump capacity and specific speed.

Summarizing briefly, a pump characteristic curve provides a wealth of data based on actual tests relative to a pump's hydraulic capability and basic design. This data may be employed in determining its suitability for a specific application.

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Cavitation

In the late 19th Century, the development of significant rotating speeds as a means of transmitting energy resulted in a new phenomenon — CAVITATION.

Initial examples of cavitation were noted on marine propellers and water turbines as vane surfaces took on a pitted appearance similar to erosion. Closer observation revealed the formation of bubbles developing just prior to the flow reaching the affected areas.

In a centrifugal pump, the suction entry area of the impeller immediately beyond the vane tips is the most susceptible to this type of attack.

For the past six decades, cavitation has been, and continues to be, a popular and intriguing subject in the field of fluid rotating machinery. As demands for lower costs continue to grow, the design trend has been toward providing the same hydraulic capability with a physically smaller pump operating at higher speeds. The result is a pump requiring extensive state-of-the-art refinement in the suction area to overcome cavitation.

Absolute Pressure

Absolute pressure in a centrifugal pump is normally at its lowest in the suction area adjacent to the point at which fluid enters the rotating impeller blades. Absolute pressure is determined by liquid surface pressure, liquid lift or head, specific gravity, suction system line friction losses and surface tension characteristics.

When the absolute pressure becomes equal to or less than the vapor pressure of the liquid being pumped, the formation of bubbles consisting of dissolved gases begins to occur. The bubbles are carried by the liquid flow into an area beyond the leading edges of the impeller vanes where higher pressure causes the bubbles to condense and collapse, creating mechanical shock. The term used to indicate this condition is called "implosion."

It is important that we distinguish liquid separation from the just-described cavitation phenomenon.

Liquid separation occurs when the flow pattern deviates from the flow path provided by the pump, resulting in void areas in the intended flow path.

This can be brought about by operation at off-design flows as well as by poor pump design. The condition develops in the rotating impeller as the liquid flows past the impeller vanes at an angle not fully utilizing the impeller's flow passageways.

Cavitation, or the formation of a partical vacuum, is accompanied by a tremendous instantaneous increase in localized pressure with characteristics similar to water-hammer blows. The impacts follow each other in rapid succession, the vapor bubbles bursting both in the immediate vicinity of the affected surfaces and in the mainstream. Cavitation pitting is generally accentuated in cracks, scratches and other surface flaws.

The impact of the bursting bubbles also causes vibrations which are transmitted through the entire pump and may even set the foundation vibrating.

Extent of Damage

Physical damage will vary depending on the material being attacked and the stage of cavitation exposure. In a typical attack, the first phase may produce indentations but no metal removal. As cavitation exposure continues, actual metal removal begins to occur. The rate of damage increases exponentially as actual pitting begins.

Pitting is caused solely by the mechanical action of collapsing bubbles. Effect varies with impeller material and degree of cavitation. The instant stress developed at the precise point of collapse has been theoretically determined to be as high as 100,000 psi. Selected observations of certain types of cavitation has shown forming and collapsing rates of nearly 2,000,000 per second in a one-inch diameter area.

Material Selection

Material selection as a means of minimizing cavitation damage has been done in a empirical manner based as experience, lab testing and actual performance in the field. Results have shown both nickel-aluminum bronze and 316 stainless steel to have good resistance to cavitation erosion. Both are far superior to other materials such as gray iron, carbon steel, brass and 70-30 copper nickel. Surface hardening may also increase cavitation resistance although the hardness itself is not necessarily the deciding factor as far as pitting resistance is concerned.

Four common symptoms of cavitation are: noise, vibration, drop in efficiency and erratic flow. Briefly, these signs can be described as follows:

Noise - caused by the collapse of vapor bubbles as they enter the high pressure area. Typically identified as a light hissing and cracking sound at the onset of cavitation and a roaring noise when fully developed.

Vibration - caused by the impacting of the bursting bubbles on the surface. May also result in premature bearing and shaft seal failure.

Drop in Efficiency - indicates the onset of a cavitating condition. The degree of efficiency drop-off increases precipitously as cavitation increases. (It should be noted here that efficiency drop-off in a centrifugal pump is partially dependent on impeller design.)

Erratic Flow - usually results from a cavitating state. The severity of the fluctuating flow is determined by the degree of cavitation and, again, pump design.

To reduce or minimize the effects of cavitation, the following suggestions are made:

1. Become familiar with the cavitation problems that can occur in centrifugal pumps.
2. Review NPSH (Net Positive Suction Head) of the system and pump during the design stage of the application.
3. Consider modifications which might alleviate a cavitating condition:
   - A. Modification of suction system
     1. Suction piping
        - a. Increase suction line size to lower inlet velocity.
        - b. Shorten suction line length to reduce friction loss.
        - c. Remove any high points in the suction line where air pockets could form.
        - d. Provide smoother internal surfaces.
        - e. Avoid suction line valves if possible. Where absolutely necessary, use valves with minimal flow resistance.
        - f. Use large radius (sweeping) elbows.
        - g. Provide streamlined flow patterns with gradual transition.
        - h. In a closed system, provide a vent from the suction line back to the top of the liquid to provide an escape route for gas pockets.
     2. Reduce liquid temperature in an open suction system.
     3. Increase submergence depth.
   - B. Pump modification and substitution
     1. Modification
        - a. Replace impeller with one designed for low NPSH service.
        - b. Check with pump manufacturer for availability of an inducer.
        - c. Reduce excessive pre-rotation.
        - d. Substitute impeller material.
        - e. Check possibility of reducing rotating speed and increasing impeller O.D. to meet rating.
     2. Substitution
        - a. Replace pump with one suitable for intended service. Discuss specific requirements with pump manufacturer's application engineers. Determine their familiarity with low NPSH availability. Consider having a NPSH test run (this is a special option not covered by standard test procedures). Request certified test data covering the entire hydraulic range in which the pump may run.

Recent Findings

Until recently, similar destructive occurrences in other sectors of the pump impeller's hydraulic passageways were usually written off as erosive actions. Current thinking attributes these problems to internal recirculating flow patterns being developed within the impeller's passageways, particularly under low flow conditions.

It has been established that high vacuum conditions may exist in the "eye" of the recirculating flow pattern in what was previously believed to be strictly a high-pressure area. This can result in the same destructive cavitating condition previously attributed exclusively to the low-pressure suction area.

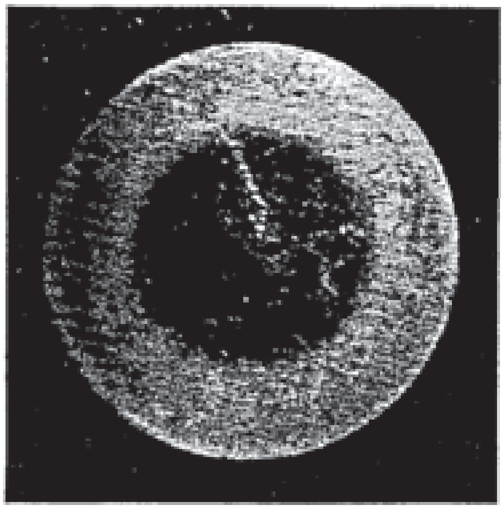
To alleviate this condition, it may be necessary to maintain a certain minimum flow rate through the pump. The excess flow could be by-passed back to the suction source when this temporary condition exists.

Conclusion

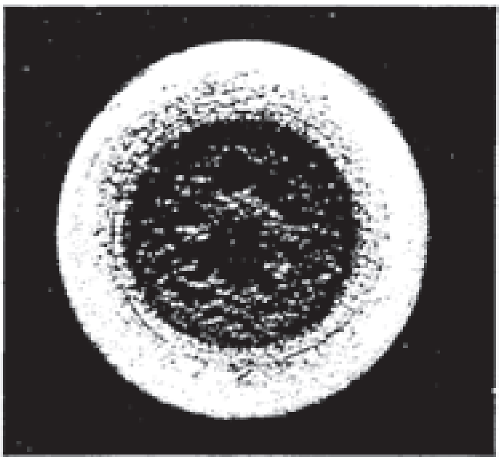
In essence, cavitation occurrence in centrifugal pumps remains a complex problem with many absolute answers yet to be established. The situation is extremely delicate relative to a pump's suction capabilities when energy levels required to cause flow may be low. It is necessary that the available absolute pressure at the pump suction exceeds the vapor pressure of the liquid pumped by an amount at least equal to the suction energy requirement of the pump.

When making basic preliminary decisions, weigh the factors of system construction vs. initial pump costs. Costly changes can be avoided by employing sound application engineering practices right from the start.

Our next quarterly article will cover NPSH (Net Positive Suction Head), a term used today to indicate the minimum absolute pressure that the user's suction system must provide at the pump impeller eye to produce a specific flow under non-cavitating (or controlled cavitating) conditions.

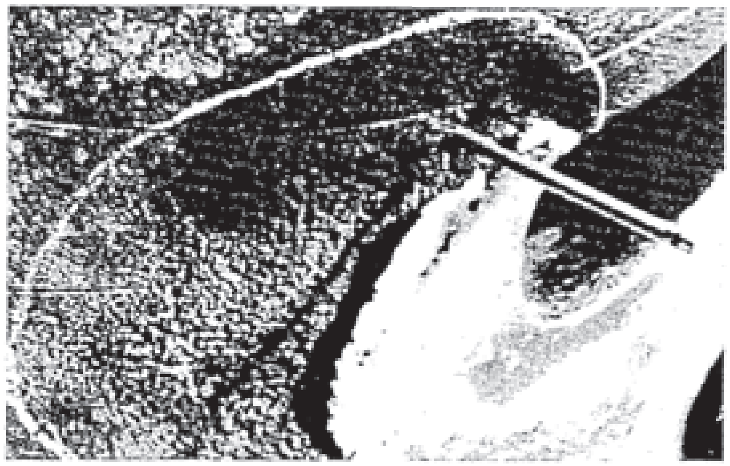


Good—*Cast aluminum bronze has excellent resistance to cavitation erosion.*



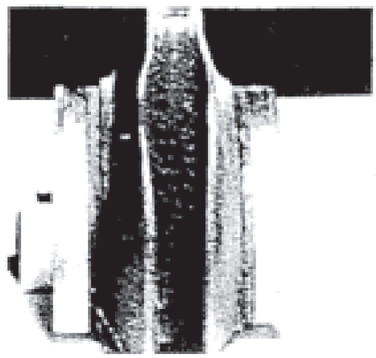
Poor—*By comparison, cast steel deteriorates badly under cavitation conditions.*

Here is what cavitation did to . . .



Cast steel

Cast iron



Bronze

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NPSH

A pump's suction capability is equally as important from a hydraulic standpoint as the specified rating (Gallons per Minute and Total Head). The term commonly used to indicate the suction condition is NPSH (Net Positive Suction Head).

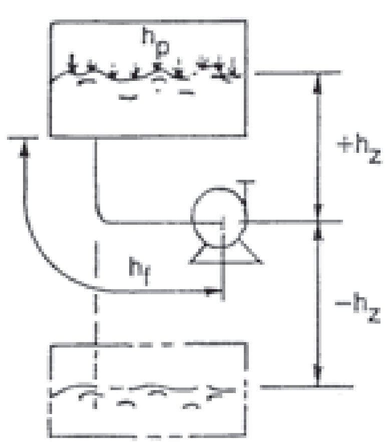
NPSH can be defined as "the absolute pressure at a datum line (normally, the centerline of the impeller eye at the suction nozzle) minus the vapor pressure of the liquid (at pumping temperature) being pumped."

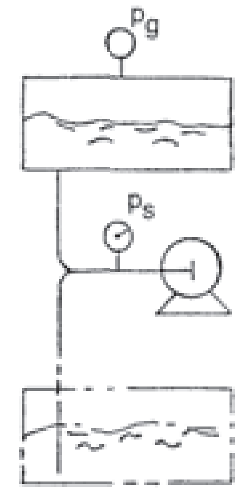
Two NPSH values must be established in determining the adequacy of a pump for the intended application: (1) the NPSH made available by the system and (2) the NPSH required by the pump for the specified rating.

1. NPSH Available. The available NPSH is usually calculated empirically in proposed systems and developed from test results in existing systems as indicated here.

   A. Proposed Systems (units in feet of water)
   NPSH Available = $h_p \pm (^*)\, h_z - h_f - h_{vp}$
   where:
   - h_p - absolute pressure acting on suction liquid surface
   - h_z - liquid suction height above or below the pump impeller centerline
   - h_f - total head losses in the suction including exit, entrance, fitting and pipe losses at the intended flow rate
   - h_{vp} - vapor pressure of the pumped liquid at the pumping temperature

   (*) for liquid level above centerline + applies; for liquid level below centerline − applies





   B. Existing Systems (units in feet of water)
   NPSH Available = $p_g \pm p_s + \frac{V_s^2}{2g} - h_{vp}$
   where:
   - p_g - gas pressure in closed tank or atmospheric pressure in open tank
   - p_s - gage pressure reading in the pump section pipe corrected to the pump suction centerline (steady flow conditions should exist at the gage tap; five to ten diameters of straight pipe of unvarying cross-section are necessary immediately ahead of the tap). The corrected gage reading is a minus term in the equation if it is below atmospheric pressure.
   - $\frac{V_s^2}{2g}$ - velocity head at point of measuring Ps (based on actual internal diameter of pipe) at point of pressure taps
   - h_{vp} - absolute vapor pressure of the pumped liquid at the pumping temperature

2. NPSH Required. All too often, the term "NPSH Available" is interpreted to be the same as submergence. In certain cases, this may be practically correct but an overall analysis is required as submergence is but one algebraic factor in the "NPSH Available" equation as given under "Proposed Systems." Absolute gas pressure acting on the suction liquid surface, friction losses through the suction system, and liquid vapor pressure at the pumping temperature are the other factors that individually or collectively could be predominant.

"NPSH Required" is the minimum absolute pressure needed at the pump suction nozzle to effect desired flow. This absolute pressure must be sufficient enough to overcome (1) the internal friction losses in the pump suction, plus (2) the head necessary to prevent vaporization in the pump, plus (3) the velocity necessary to maintain the desired flow through the pump. "NPSH Required" data should be supplied by the pump manufacturer.

Methods of determining the actual "NPSH Required" of a centrifugal pump are given in the Hydraulic Institute's test standards. A 3% drop in head is the standard used to determine the pump's NPSH criterion (C).

Any change in performance — either a drop in head or power, of efficiency at a given capacity, or a change in sound or vibration — may be an indication of cavitation but, because of the difficulty in determining, NPSH (C) is accepted as evidence that cavitation is present.

It should be noted that the average pump will give noncavitating performance at NPSH values only 1.3 times above the NPSH C value at capacities above 85% of the best efficiency point (BEP), and 1.7 times above the NPSH C value at capacities below 85% of the BEP.

The affinity relationships define the manner in which head, capacity, HP and NPSH vary in a centrifugal pump with respect to speed changes. If a pump operates at or near its cavitation limit, other factors also have an effect and the critical limiting NPSH value may vary other than as the square of the speed. Some of these factors are: thermodynamic effect in the vapor pressure of the fluid; change in surface tension; and test differences due to the relative air content in the liquid.

Cavitation Relationship

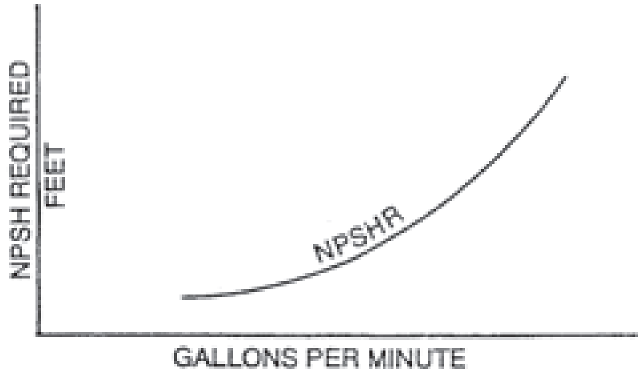
In order for a pump to operate free of cavitation, the system's "NPSH Available" must be equal to or greater than the pump's "NPSH Required." Care should be taken in this regard to make the most economical selection, particularly at very low NPSH conditions where NPSH requirements may even be given in inches instead of feet of water. It is not uncommon for the price of a pump to double when the "NPSH Available" is decreased as little as a foot in the low NPSHR region (3 to 2 feet, as an example).

Where at all possible, raise the suction liquid surface, increase the size of your suction line, reduce the number of fittings and make the most direct piping approach possible.

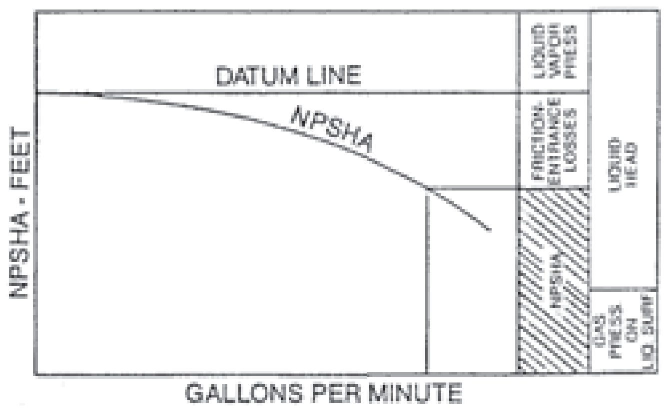
Additional gas pressure acting on the suction liquid surface in a closed system would also increase "NPSH Available" if the vapor pressure (temperature of the liquid pumped) could be held.

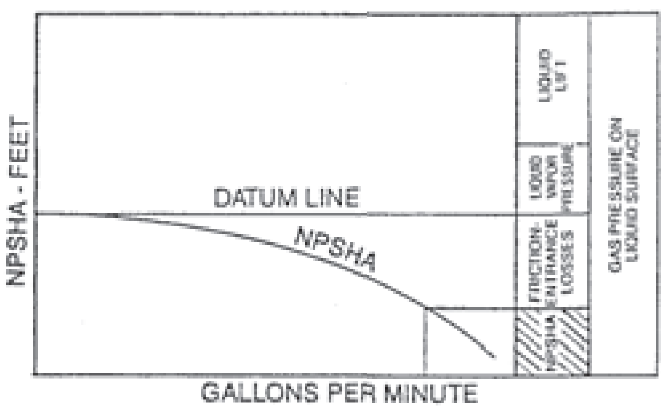
NPSHR increases as capacity increases in a varying exponential manner. Several factors employed in the pump's design have an effect on the pump's suction capability, particularly, impeller eye area, blade angle, blade leading edge, number of blades, inlet area at blade edges, peripheral velocity of the inlet blade edges (related to pump RPM) and overall smooth transitional passageway design.

A typical pump NPSHR vs Capacity Curve as provided by the pump manufacturer is shown below.

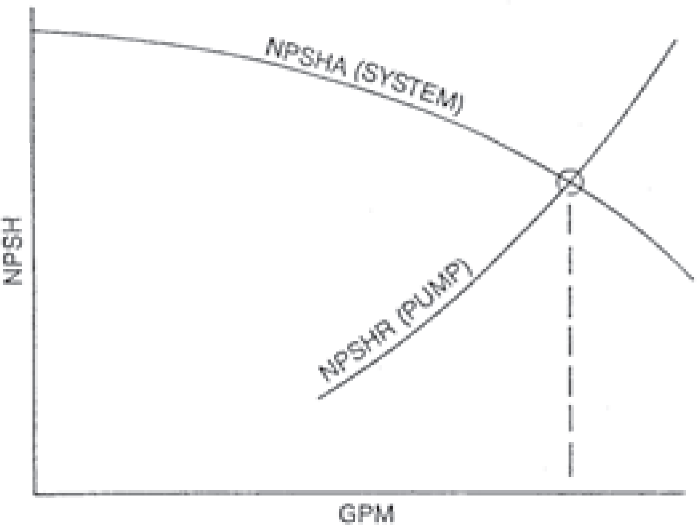


System curves covering different applications established from data are illustrated in this manner:





Once the developed NPSHA curve and the pump manufacturer's NPSHR curve are plotted on the same graph, we can determine the capacity where cavitation will begin to occur if the suction system is the predicating factor:



To satisfy the suction requirements, the pump must be operated at a capacity less than the intersection point of the NPSHA and NPSHR. At operation beyond this point, one can expect a cavitating state to exist.

Other Factors

Liquid Characteristics

A factor oftentimes overlooked in pump NPSH application engineering is the difference in liquid (and its vapor) characteristics relative to those of 68°F clear water, which is considered the base standard and is used almost universally in a manufacurer's standard test setups.

Experimental tests and actual applications handling various liquids over a broad temperature range have shown that the "NPSH Required" in a specific pump at an intended capacity can vary. Terms such as "NPSH Corrections" and "NPSH Adjustment" have been used to identify this differential. Even when just handling water you may note a change in "NPSH Required" as the liquid temperature rises. Minute though it may be below 200°F, particularly in water, it will become more apparent in closed systems where higher liquid temperatures can be reached.

Always keep in mind the total perspective of what is occuring relative to the system you have, the pump requirement based on cold, clear 68°F water and the liquid being pumped along with its characteristics in both the liquid and gaseous states relative to the pump's operating temperature.

Specifics other than vapor pressures of some liquids and a few published results of some specific NPSH individual tests are lacking. The intent here is to make everyone aware that such a condition exists and can have some effect on the overall performance, usually being favorable at increased temperatures if the same "NPSH Available" can be maintained.

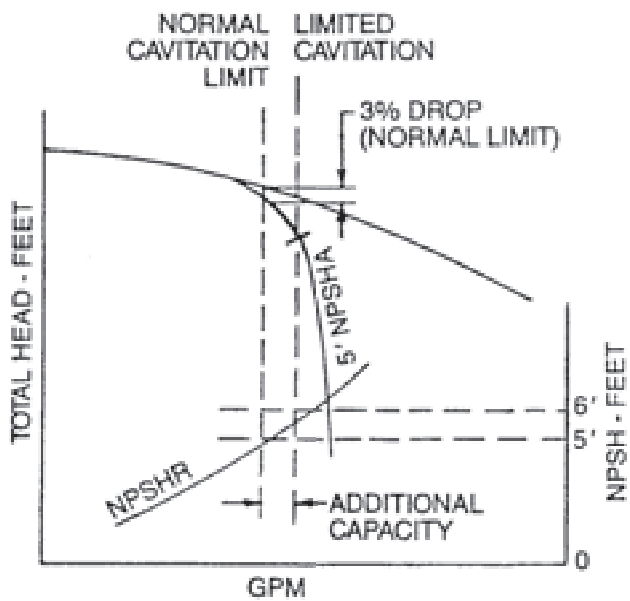
Minimum Flow

At times, pumps operating at extremely low NPSH conditions which, supposedly, would provide attainable flow rates, react erratically, developing fluctuating, unstabilized flow due to extensive internal recirculation patterns occurring within the pump impeller. The low flow requirement relative to the pump's optimum point of operation can cause this problem in certain pumps. The pump manufacturer should advise the capacity range for the unit when the "NPSH Available" drops below 5 feet of water so that the user can design a system which maintains flow within the pump's specified capacity range.

Special Applications

Knowledge of pump cavitation and its effects along with a thorough understanding of required and available NPSH allows some applications to operate in a "controlled" or "limited" cavitation range (initial phases). Where applicable, such as in distillation processes, an appreciable pump cost saving may be realized.

In this situation, the pump is operating in a range commonly referred to as the "break" or "knee" of the pump characteristic curve.



In the case shown here, one may be able to use a 5-foot "NPSH Required" unit rather than the 6-foot NPSH unit normally called for.

We do not recommend this as a common fix but make note of it as an example of how knowledge of NPSH and cavitation effects can be utilized, in controlled applications, to provide an adequate economical alternate.

The limit for the intended point of operation in a particular application will usually have to be determined empirically, dependent on noise, vibrating hydraulic stability and pump material allowance based on the projected point of operation from factory test results.

Summary

Ignoring pump suction requirements can be costly. Inadequate capability can result in a marked reduction in head and capacity or even a complete failure to pump. The result is pump cavitation — a phenomenum previously discussed in article 3. Briefly, the reduced pressure in the impeller inlet allows some liquid to change to vapor, thus choking part of the flow, causing erratic behaviour.

As the flow continues through the impeller passageways, a pressure build-up begins. A point is reached where the vapor bubbles collapse and revert back to the liquid state. This is referred to as an implosion and can cause serious damage to the impeller metal in the immediate area where it occurs.

Any change in performance, either a drop in head, power or efficiency at a given capacity, or a change in vibration or sound may be an indication but, because of the difficulty in determining just when the change starts, a drop in head of 3% is usually accepted as evidence that cavitation is present. At this time, there is no known method of indicating the point of incipient cavitation.

In recent years, it has been noted that liquids other than water may react differently, cavitation-wise, under the same dynamic conditions. Standard testing of pumps is normally based on results obtained with clear, cold, fresh water at a temperature not over 85 °F in conformance with the Hydraulic Institute Test Code. Liquid properties such as pressure, temperature, latent heat of vaporization and specific heat help determine the cavitation characteristic. Temperature alone will drastically change the cavitation characteristic of water.

In short, many applications may have to be resolved after having made some compensation for differing properties and conditions. In some instances, only actual tests will provide the necessary answers.

Pump cavitation is not only destructive to the impeller but causes radial and thrust loads against the impeller profile, resulting in excessive vibration due to the fluctuating load. Noise is produced by the collapse of the vapor bubbles at the point where they enter the region of higher pressure and implode.

Insufficient NPSH at the desired flow is the number one problem in pump application. It can be resolved by increasing the system's available NPSH or by selecting a pump with a lower NPSH requirement.

Conclusion

Familiarization with NPSH requirements, along with a basic understanding of cavitation and ways of avoiding it, can go far in reducing the number of misapplications where centrifugal pumps are employed.

Also, know your pump vendor's capabilities. Does he or can he supply the pump NPSHR data you require to make a sound engineering decision? Can he conduct special NPSH tests under varying conditions which compare closely to your specific requirements?

Positive steps taken to solve difficult suction conditions during the design stage will reduce the risk of cavitation, thereby eliminating costly modification of the suction system and/or pump along with the usual upsetting delay in time.

Pump Primer

Basic Information About Centrifugal Pumps

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Number 5 in a series.

by John H. Horwath
Chief Engineer, Ampco Pumps

System Curves - Introduction

A system curve is a graphical representation of a flow range through a process arrangement against the pressure (expressed as head) required to complete the work.

The hydraulic energy may consist of (1) potential head (based on elevation of the liquid relative to a selected datum line), (2) kinetic head (energy in a unit weight due to its velocity) and (3) pressure head (energy contained in the liquid because of its pressure).

A process system physically consists of several components which may include: piping, valves, nozzles, vessels, flow meters, process equipment and other liquid handling conduits through which flow is required.

Curve Development

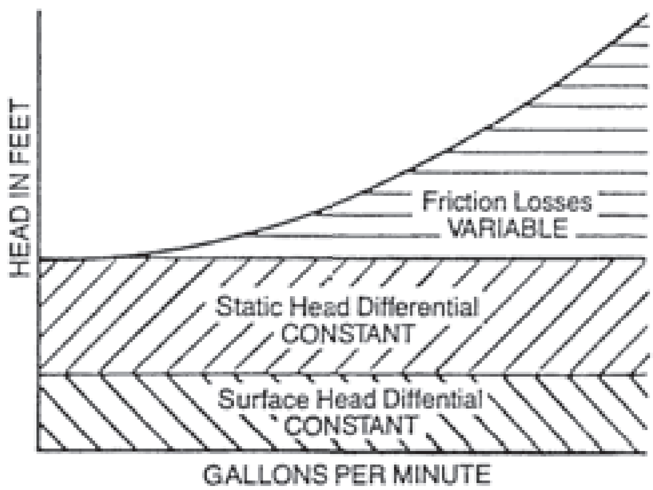
Basicially, two types of conditions must be overcome to induce flow at a desired rate. They are: dynamic head (friction losses occuring in the system which vary according to the rate of flow by approximately a square ratio) and static head (constant differentials which exist at suction and discharge points of the system, i.e., the gas pressure acting on the liquid surface at each end of the system, and the difference in elevation of the liquid surface at each end).

It is common to initially determine each head requirement (suction and discharge) independently since the first condition to be determined is the NPSH available (see article 4 in this series). Next, we can determine the total head differential (suction plus discharge algebraically) which the pump must develop to attain the desired flow. Note that a centrifugal pump's capability to hydraulically satisfy a rating specification goes beyond meeting a developed differential pressure at the specified flow in that the NPSH available at this same flow must be equal to or greater than that required by the selected pump to operate properly.

When analyzing a system for the purpose of selecting a pump, the resistance to liquid flow through the specific components may be developed from pipe friction tables, actual test data or data supplied by the manufacturer of the particular component or system. Excellent sources for pipe friction losses include Cameron's "Hydraulic Tables" and Hydraulic Institute's "Engineering Data Book." In some instances, it may be necessary to conduct your own tests at known flow rates to obtain the friction loss (pressure drop) across the component in question.

Total Head Determination

Determination of the total head requirement of a system at various flow rates can best be seen from Figure 1 below.



In terms of a formula, the system head can be stated as:

Total Head (TH) =
ΔH surface + ΔH static elev. + H friction
(algebraically added)

All units shall be expressed in terms of feet of water. When the total has been established for one flow, it is a simple procedure to calculate a series of flow points for plotting a system curve over a broader flow range. The only variable is the friction loss relating to flow as a square function. H friction includes pipe, fitting, entrance, exit and all other processing equipment losses related to a specific flow.

$$H_2 \text{ (friction } Q_2) = H_1 \text{ (friction } Q_1) \times \left(\frac{Q_2}{Q_1}\right)^2$$

When calculating the total head, remember that the constant static and surface heads must be algebraically added to the H (friction Q_2) calculation.

Improved System Design

System design, in many instances, is a trial and error affair, further compounded by "safety factors." The system designer, who will be held accountable if the pump selected is under capacity, will typically play it safe by adding a 10-15% safety factor. The pump manufacturer's application engineer will add his own "safety factor" for the pump, thereby further increasing the overdesign characteristics by another 5-10%.

Obviously, a considerable degree of experience must be attained before one can reliably project requirements for specific applications; however, it would appear reasonable for all parties to provide their methods of determination, enabling "safety factors" to be scaled down from the cumulative effect that will otherwise result.

Commonly, throttling or orificing is utilized to compensate for overdesign. The more effective method would be to reduce impeller diameter by machining the impeller at the job site. This would result in decreased flow (and/or developed head), thereby reducing wear and energy costs. If conditions change over time and adequate HP is available, simply replacing with a larger diameter impeller may suffice. Changing speeds can also be effective under these circumstances; however, at this time the impeller route would appear to be the more economical method for the majority of applications.

Figure 2 indicates the wasted energy lost in throttling when an oversized impeller is employed, and the energy savings due to a reduced horsepower requirement. Remember that, by using a larger diameter impeller, you are paying for wasted energy which, in turn, is causing needless wear and tear to your system and pump in terms of erosion and heat buildup.

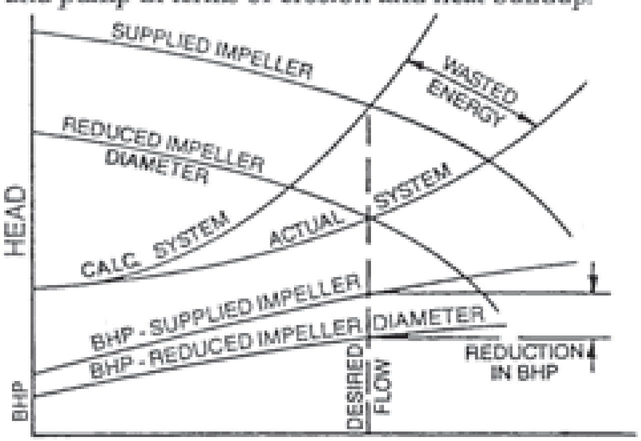


Figure 2 GALLONS PER MINUTE

Advantages

The use of graphs in pump applications provides not only a view of the "as is" condition, but illustrates such "what if" situations as:

1. Increase in pipe friction (aged system) - Figure 3

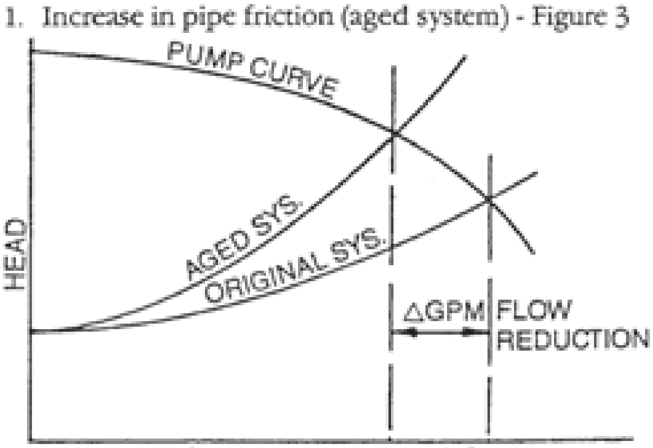


Figure 3 GALLONS PER MINUTE

2. Reduced pump volumetric efficiency ("worn" running clearance) - Figure 4
3. Effect of changing suction conditions - Figure 5
4. Effect of added safety factors - Figure 6

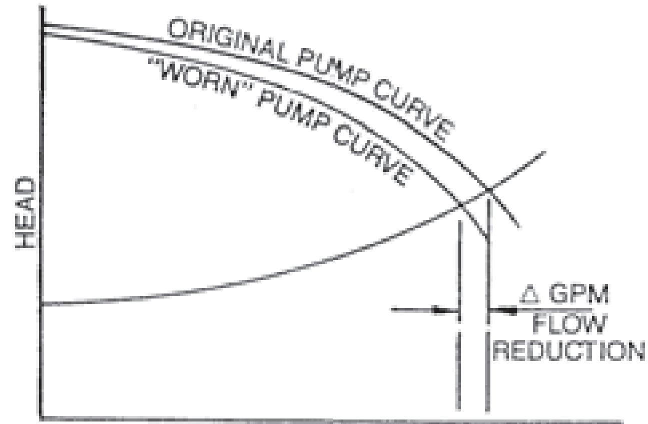


Figure 4 GALLONS PER MINUTE

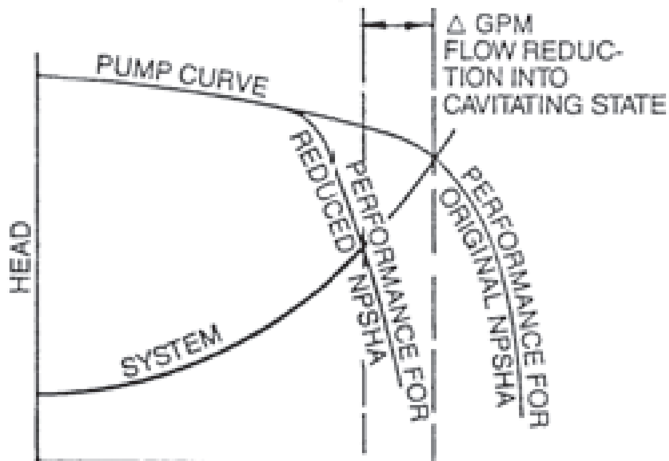


Figure 5 GALLONS PER MINUTE

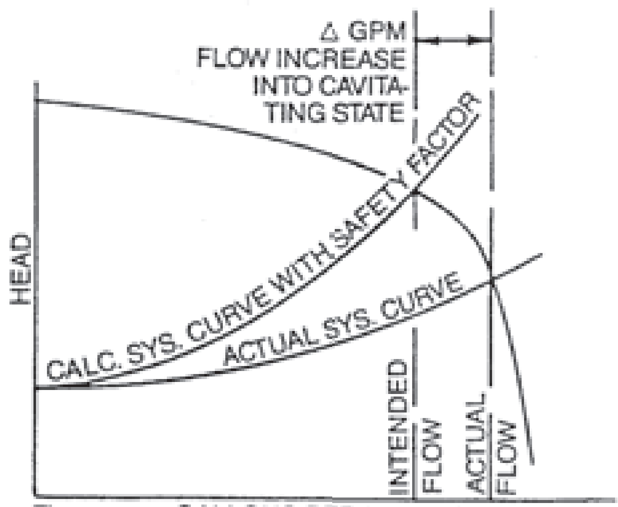


Figure 6 GALLONS PER MINUTE

Parallel and Series Operation

A stable pump characteristic (ever rising) is recommended when two or more pumps are combined in either parallel or series operation. Two pumps in parallel produce the combined capacity of both at the same head (Figure 7). Two pumps in series produce the combined head of both pumps and the capacity of one

(Figure 8). This relationship is true for the pumping units alone, but is not necessarily true when the pumps are part of a system.

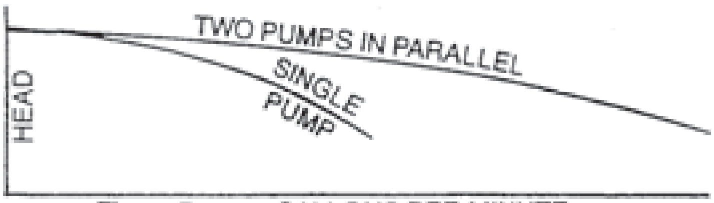


Figure 7 GALLONS PER MINUTE

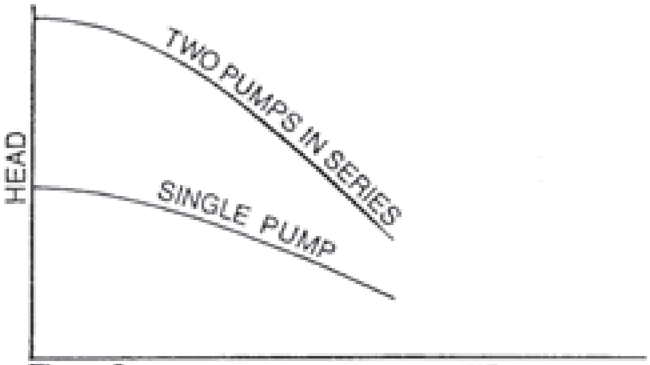


Figure 8 GALLONS PER MINUTE

Parallel

This arrangement, producing greater flow than provided by an individual unit, is more common. The option of using one large pump or two small units depends on the requirements of the intended operation.

A rising head-capacity curve is especially essential for reliable service in parallel application. A drooping characteristic curve can potentially intersect a given head at two different points as illustrated in Figure 9.

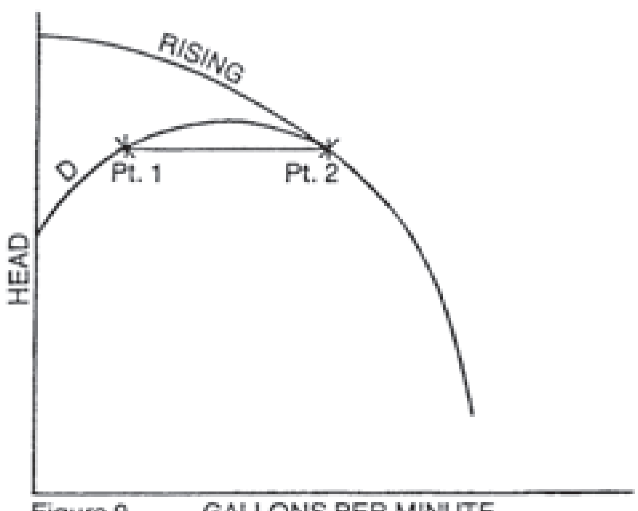


Figure 9 GALLONS PER MINUTE

The effect of this can be a "hunting action" between the two points, causing variations in capacity and pressure. The beginning of the pressure variation lies in the fact that, at some time, the pressure in the discharge line is higher than that developed in the pump and a movement to reverse flow occurs.

Series

Series operation is employed where insufficient pressure is developed by a single unit or where limited suction pressures are available. Characteristically, a low-speed pump has a lower NPSH requirement than a higher speed pump. When used as a second-stage unit, the higher speed pump can develop the higher pressure requirement needed at the discharge point. Pumps with rising characteristic curves are preferred because they develop the head within a narrow capacity range.

Recommended Practices for Series and Parallel Operation

In series operation, both units should be provided with by-pass systems to allow either unit to be operated independently. This permits operation without the idle unit being on-stream. A by-pass system also allows operation at a reduced flow rate during maintenance, inspection or repair of either pump.

In parallel operation, it is essential to have identical suction arrangements to prevent one unit from being "starved." A similar discharge arrangement to a common header is also strongly recommended.

Summary

The interaction between a pump and its system has been discussed in some detail in the preceding pages. The more thorough the understanding of this subject, the smaller the chance for application error. The system, while being static in a physical sense, is as responsible for the hydraulic operation as the rotating pump itself. Once this fact is recognized, the operational aspects of a pump/system arrangement become significantly less complicated.

Pump Primer

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Practical Limits for Centrifugal Pumps

This reference, based on a combination of theoretical study and practical experience, covers the "normal limits" of the typical general-purpose centrifugal pump. The intent is to broadly establish parameters defining probable pump capabiliities under various conditions.

Capacity

Initial efforts should be made to apply a pump to a system operating near the pump's highest efficiency point. When operation is continuous, it is doubly important that the point of operation be confined within 50% of the pump's best efficiency point (bep) of its impeller diameter. Operation in this region normally eliminates the need to specifically check limitations based on hydraulic, mechanical and thermodynamic conditions.

One exception might involve a system with a low available NPSH condition which could dictate an oversize pump selection. Operation at significantly reduced capacity will result in a high differential pressure in the casing volute acting against the side profile of the rotating impeller. The resulting force becomes a radial load which, acting on the impeller shaft, can cause shaft and bearing failure, premature impeller and wear ring wear, and mechanical sealing problems where inadequate mountings are employed.

A second problem resulting from reduced capacity — noted particularly in high head (over 140 feet) pumps operating at less than 20% design — is thermodynamic buildup. At partial capacities involving low efficiencies, much of the wasted energy is converted to heat and transferred to the liquid being pumped. As this process continues, the pump casing absorbs some heat, with a portion being dissipated through convection and radiation. Maximum temperature rise for a specific application must be determined based on hydraulic and mechanical conditions.

Operation at partial capacity for substantial periods may also result in erosion of internal areas subjected to aggravating recirculating flow patterns which deviate from the intended smooth transition flow lines of design capacity. Damage need not be limited to erosive action alone. Cavitation damage can also occur in recirculating areas where developed velocities can provide high vacuum pockets, resulting in cavitation in a supposedly high pressure area.

Pressure

Pressure limitations are primarily determined by the pump casing suction and discharge connections. Until recently, in working pressures below 300 psi, casing and cover wall thicknesses were considerably heavier than necessary to satisfy foundry sand casting practices. Changing manufacturing processes, employing sheet forming, investment casting and evaporative foam casting, substantially reduce wall thickness and, in many instances, now are a predicating factor. Other factors include the number and material of casing bolts as well as shaft sealing, where both the seal and gland arrangement need to be analyzed relative to allowable physical limits.

In handling corrosive media, the potential for substantial corrosion attack dictates that design calculations be based on wall thickness less a 1/8-inch corrosion allowance.

Other modification which are available to provide increased working pressures include: heavier flanges, thicker casing walls, the addition of ribs and higher strength case material. Substantial additional fixed costs and extended deliveries may result when a pump manufacturer must modify an existing design to meet additional specifications.

Temperature

Liquid temperature can have a significant effect on the media being pumped, on the physical properties and corrosion resistance of the housing material, on the type of shaft sealing employed and on the make-up of the complete rotating assembly.

While normal procedure would be to pump at ambient temperature unless the process dictates otherwise, there are circumstances where operation at high temperatures would be beneficial to the pump's performance. One of these areas is viscosity. Typically, an increase in liquid temperature reduces the liquid's viscosity. This, in turn, expands the pump's hydraulic range and increases efficiency. Higher temperatures can also be beneficial in keeping solutions in a dissolved state.

Generally, a material's corrosion resistance will decrease as pumping temperature increases, thereby accelerating the corrosion rate of attack. Mechanical properties may also be adversely affected by temperature rise.

Mechanical shaft sealing is prevalent in today's pump. Seal materials are dependent on temperature as well as other factors. Of prime concern is the necessity of providing adequate lubrication to the sealing surfaces. The media being handled must remain in a liquid state either through a reduction in temperature or an increase in absolute pressure in the affected area to maintain a sufficient level of lubrication.

Temperature, along with required RPM and potential loading conditions, dictates the type of assembly required. Liquids outside the 40° to 250°F range will require closer scrutiny relative to the intended bearing arrangement and specific approval by the pump supplier. The pump manufacturer will normally advise the pumping frame temperature limits for grease and oil lubrication based on the type of mounting employed. Actually, the bearing frame temperature may run considerably lower than the pumping temperature, due to its design, tending to result in substantial radiation loss.

Viscosity

The performance of a centrifugal pump is affected when handling viscous liquids. A marked increase in brake horsepower, a reduction in head and some reduction in capacity occur when viscosities exceed 100 SSU. Typically, pump performance data is based on handling cold water (32 SSU).

The maximum viscosity that a conventional centrifugal pump can handle is dependent on pump size and design. Viscosities as high as 4000 SSU (similar to latex paint) have been handled satisfactorily with well-designed conventional centrifugals. As viscosity increases, it becomes increasingly important, in selecting a pump, to provide a near maximum impeller diameter, thereby reducing chances of the cut-water being ineffective, resulting in an internal rotating shearing action similar to a car wheel spinning in mud.

Specific Gravity

A pump, in establishing liquid velocity, develops kinetic energy by moving a measured volume of liquid per time increment at a particular velocity head. The standard liquid used in determining pump performance is cold, clear water with a 1.0 specific gravity. Pumping liquids with different specific gravities will have a direct proportional effect on the brake horsepower requirement per the formula:

$$BHP = \frac{\text{Head (in feet)} \times \text{Capacity (GPM)} \times \text{Sp. Gr.}}{3960 \times \text{Pump Efficiency}}$$

It is important that this calculation be made based on the maximum BHP requirement of the impeller diameter in its overall operating range. The motor horsepower selection should also be based on this maximum requirement.

Liquid - Solid Mixtures

These can be divided into two categories, namely "slurries" and "solids in suspension." A slurry indicates a mixture of a liquid and a finely divided solid. When solid sizes exceed 1/16", the mixture is referred to as "solids in suspension." The minimum velocity applicable to the system is established by the velocity required to keep the solid content in suspension. Since a pump is basically a velocity machine, minimum velocity is not a problem, but the higher velocities incurred in generating a head can cause an extremely high rate of wear in the hydraulic passageways.

Pumping a liquid-solid mixture should result in the same head as pumping clear water, reduced by the additional hydraulic losses caused by the presence of solids in the pump passageways. It is important to realize that solids suspended in a liquid cannot absorb, store or transmit pressure energy as pure fluids can.

No absolute corrections can be recommended for this condition. The nature and percentage of the solids will affect the pump's performance to varying degrees. About the only definitive factor that can be established is the size of solid capable of being passed by the pump. (The pump manufactuer should supply this data expressed as a maximum sphere diameter). While some parameters may be developed from text books, technical papers, pump manufacturer's data, etc., only actual experience coupled with good engineering judgment will provide a realistic approximation.

Because of the inherent properties of solid-liquid mixtures plus particle size and variation, determination of the pump's capability to handle solid content varies widely. We arbitrarily have established a 25% solid content by volume to be used in initial considerations. Regardless of hydraulic capability, the prime considertion may well be the excessive wear caused by the solid content and its effect on pump life and performance.

Speed

Operation of a centrifugal pump should be limited to the maximum speed indicated on the pump's nameplate. Any intended service beyond the listed speed must be thoroughly reviewed with the pump manufacturer's application engineers.

The centrifugal pump, being a velocity machine, will develop significant additional head ($speed^2$) and a substantial increase in brake horsepower ($speed^3$) as well as a changing NPSH requirement (usually higher) as the speed is increased. These higher pressures and loads result in additional physical strain on the pump.

Most centrifugal pumps designed in the U.S.A. have, up to now, been developed to operate on 60 Hz motor design speeds of 1200, 1800 and 3600 RPM. While variable and higher speed capabilities are available, none have yet made a real impact. The unit most likely to achieve this within the next decade is the variable frequency controller used with AC motors. As the frequency controller gains acceptance, production costs will drop, making it an even more attractive cost-saving element.

In most instances, only a qualified pump manufacturer can determine whether his pump is suitable for operaton beyond its original design speed range. Working with the motor manufacturer, the suitability of a specific motor driver is established and a motor AC drive package is then determined.

Entrained Air

Entrained air has an adverse effect on pump performance. As little as 1% entrained air by volume can reduce pump head and capacity substantially. Under no circumstances should a standard centrifugal pump be expected to handle more than 3% air by volume, as measured under pump suction conditions.

Substantial effort should be made to keep air out of the liquid entering the pump. This commonly occurs if a vortex develops at the suction pipe inlet. Adequate submergence or baffling can help prevent this condition. A leak in the suction line or pump stuffing box operated under a vacuum may also introduce air into the pump's suction. A well-designed entry coupled with good maintenance practices will alleviate most entrained air problems.

Dissolved gas or gas evolving from a chemical reaction can also prove troublesome. Common practice calls for the use of a larger diameter impeller (not necessarily a larger inlet) to meet the hydraulic performance based on handling cold clear water alone. Where the percent of air (or gas) exceeds the recommended limit for standard pumps, which may vary for different designs, it may be appropriate to consider less efficient pumps designed specifically for handling two-phase flow.

Comprehensive information covering some of the specific areas introduced in this issue will appear in future articles in the "pump primer" series.

Pump Primer

Basic Information About Centrifugal Pumps

Ampco Pumps Co., Inc.
(414) 643-1852 Telephone
(414) 643-4452 Facsimile

Number 7 in a series.

by John H. Horwath
Chief Engineer, Ampco Pumps

Energy Savings - Introduction

One of the biggest expenses today in liquid processing operations is the cost of energy. It is not uncommon for the cost of wasted energy in a continuous pump service over a period of two years or less to surpass the initial cost of the pump.

As a result, more attention in our current "competitive" era is being directed toward reducing operating costs. Higher pump efficiencies along with higher motor efficiencies are immediate solutions. Other equally important factors include system design and pump application. In this issue of the "pump primer," we will approach the subject of energy savings from each of these aspects.

Pump Efficiencies

The most obvious direct savings in the pump unit results when a smaller size drive can be used because of a pump's higher efficiency and non-overloading performance characteristics. (A non-overloading pump provides a series of impeller diameters having tangential relationships with motor standard horsepower ratings to insure adequate power requirements regardless of flow - see figure 1).

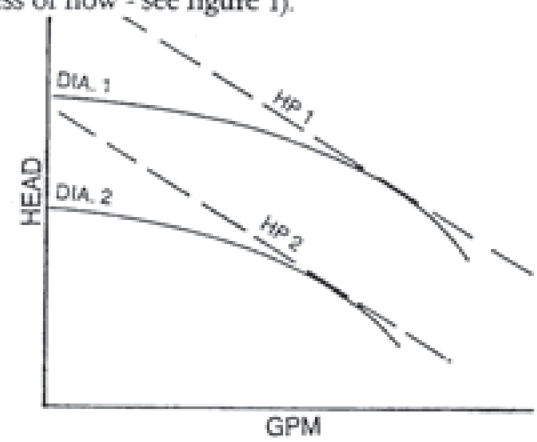

Figure 1

All drawn horsepower not utilized in developing the required water horsepower:

$$WHP = \frac{GPM \times TH \times Sp.\ Gr.}{3960}$$

is wasted horsepower. Since the water horsepower requirement remains the same at the specific rating for any pump, the less efficient units at the point of operation will demand a greater brake horsepower requirement:

$$BHP = \frac{GPM \times TH \times Sp.\ Gr.}{3960 \times \text{pump efficiency}}$$

The wasted HP is the difference between the brake horsepower and the water horsepower as indicated in figure 2.

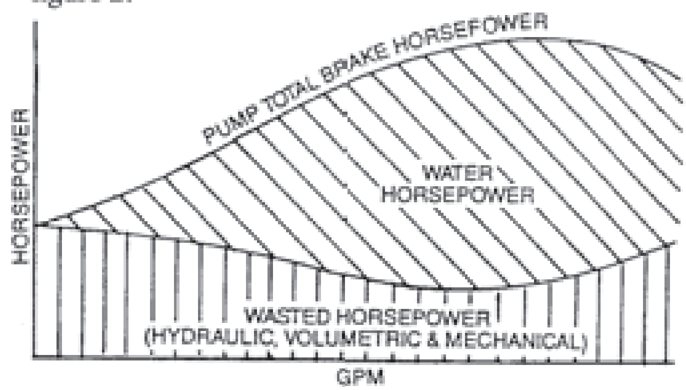

Figure 2

Of the three primary pump losses shown under wasted horsepower (hydraulic, volumetric and mechanical), the hydraulic loss will usually be the most significant. Hydraulic losses include surface friction, turbulence, eddy currents and separation occurring in the complex flow path of the rotating impeller and stationary volute. The loss effect will vary with the rate of flow.

Wasted horsepower is dissipated through local heat buildup and in noise or physical damage (usually resulting from cavitation and erosion caused by poor flow patterns).

Proper hydraulic design is necessary if higher efficiency is to be attained. Requirements include: (1) smooth surfaces, (2) streamlined flow paths, (3) gradual conversion of velocity energy within cross-sectional areas to pressure energy and (4) easement in directional change.

The pump designer employs nature's basic principle of liquid flow (path of least resistance) in providing passageways as best can be compromised when proceeding alternately through stationary and rotating passageways of varying velocities along with the final transitional phase of energy conversion.

Many manufacturers have sacrificed proper design to remain cost competitive. Ampco, while continuing to rely on its extensive casting, fabricating and machining expertise, has adopted the investment and lost foam casting methods to improve pump efficiency while reducing manufacturing cost.

When evaluating pumps, request efficiency data if it doesn't appear on the published curves. You might also inquire about the method of testing employed. Are calibration tests in conformance with Hydraulic Institute or any other engineering standard? If horsepower curves and standard head-capacity curves are provided, efficiency curves can be developed by using the following formula:

$$\text{Efficiency \%} = \frac{\text{GPM} \times \text{TH} \times \text{Sp. Gr.}}{\text{BHP} \times 3960} \times 100$$

System Design

In general, there is more area for improvement of energy conservation in the system than in the pump. It has been estimated that, within the last decade, over 20% of required energy is wasted because of system deficiencies. Primary causes are "orificing" and "throttling" to reduce overflows. As within a pump, system throttling, with its destructive nature, can play havoc with components, reducing their life.

Short-term orificing and/or throttling can be used in balancing a system. However, it would be more expedient, where possible, to either reduce the impeller diameter or decrease the pump's speed. Obviously, a variable-speed unit, with its instant response, would be most desirable. Unfortunately, current variable speed pricing will, more than likely, offset projected cost savings. A more economical approach would be to reduce the impeller diameter to the point where minimal, if any, throttling would be required.

Excessive pipe friction losses can be reduced by using larger pipe diameters, long sweeping elbows and valves with minimal resistance losses. While this will involve higher initial costs, they will be more than offset by the energy savings resulting from reduced head requirements.

Continual equipment checks and periodic maintenance will also assure a more reliable and efficient processing system.

Pump Application

Practical approaches to pump energy conservation from the user's perspective include higher driver efficiency, good maintenance practices and use of broader high-efficiency ranges.

In evaluating pump performance, the user should determine the efficiency rating at the intended point of service. Unfortuately, peak efficiency cannot be maintained over an extended capacity range. Therefore, instances will occur where a pump which is less efficient overall will have a higher efficiency at a specific point of operation. In these cases, final decisions should be delayed until the entire operating sequence is analytically reviewed. If the drop in efficiency is minimal and the unit meets all other requirements, it may be advantageous to stay with a wide capacity — higher efficiency range.

The National Electrical Manufacturers' Association (NEMA) has recently published a new standard defining energy-efficient motors. The definition includes a table of nominal and minimum efficiency levels for electric motors which categorizes motors as "energy efficient" or "high efficiency."

NEMA's publication MG-1 is the standard reference for motors and generators in the United States. The standard definition is provided by MG-1-1.41.2 and MG-1-12.55. The full load nominal and minimum efficiencies for open and enclosed motors are found in table 12-6B at the end of this article.

Some bid evaluations request the "wire-to-wire efficiency," termed "W/W," where:

$$\text{W/W} = \frac{\text{GPM required}}{\text{GPM actual}} \times \frac{\text{H required}}{\text{H actual}} \times \text{E motor} \times \text{E pump}$$

Summary

User interest in pump requirements rises significantly during periods of "energy crisis" and then declines as the crisis subsides. Be aware that wasted horsepower relentlessly carries on its destructive attack regardless of energy costs. Why pay double for trouble?

TABLE 12-6B

FULL LOAD EFFICIENCIES / OPEN MOTORS

	2 POLE		4 POLE		6 POLE		8 POLE	
HP	Nominal Efficiency	Minimum Efficiency	Nominal Efficiency	Minimum Efficiency	Nominal Efficiency	Minimum Efficiency	Nominal Efficiency	Minimum Efficiency
1.0	——	——	82.5	80.0	77.0	74.0	72.0	68.0
1.5	80.0	77.0	82.5	80.0	82.5	80.0	75.5	72.0
2.0	82.5	80.0	82.5	80.0	84.0	81.5	85.5	82.5
3.0	82.5	80.0	86.5	84.0	85.5	82.5	86.5	84.0
5.0	85.5	82.5	86.5	84.0	86.5	84.0	87.5	85.5
7.5	85.5	82.5	88.5	86.5	88.5	86.5	88.5	86.5
10.0	87.5	85.5	88.5	86.5	90.2	88.5	89.5	87.5
15.0	89.5	87.5	90.2	88.5	89.5	87.5	89.5	87.5
20.0	90.2	88.5	91.0	89.5	90.2	88.5	90.2	88.5
25.0	91.0	89.5	91.7	90.2	91.0	89.5	90.2	88.5
30.0	91.0	89.5	91.7	90.2	91.7	90.2	91.0	89.5
40.0	91.7	90.2	92.4	91.0	91.7	90.2	90.2	88.5
50.0	91.7	90.2	92.4	91.0	91.7	90.2	91.7	90.2
60.0	93.0	91.7	93.0	91.7	92.4	91.0	92.4	91.0
75.0	93.0	91.7	93.6	92.4	93.0	91.7	93.6	92.4
100.0	93.0	91.7	93.6	92.4	93.6	92.4	93.6	92.4
125.0	93.0	91.7	93.6	92.4	93.6	92.4	93.6	92.4
150.0	93.6	92.4	94.1	93.0	93.6	92.4	93.6	92.4
200.0	93.6	92.4	94.1	93.0	94.1	93.0	93.6	92.4

TABLE 12-6B

ENCLOSED MOTOR

HP	2 POLE Nominal Efficiency	2 POLE Minimum Efficiency	4 POLE Nominal Efficiency	4 POLE Minimum Efficiency	6 POLE Nominal Efficiency	6 POLE Minimum Efficiency	8 POLE Nominal Efficiency	8 POLE Minimum Efficiency
1.0	—	—	80.0	77.0	75.5	72.0	72.0	68.0
1.5	78.5	75.5	81.5	78.5	82.5	80.0	75.5	72.0
2.0	81.5	78.5	82.5	80.0	82.5	80.0	82.5	80.0
3.0	82.5	80.0	84.0	81.5	84.0	81.5	81.5	78.5
5.0	85.5	82.5	85.5	82.5	85.5	82.5	84.0	81.5
7.5	85.5	82.5	87.5	85.5	87.5	85.5	85.5	82.5
10.0	87.5	85.5	87.5	85.5	87.5	85.5	87.5	85.5
15.0	87.5	85.5	88.5	86.5	89.5	87.5	88.5	86.5
20.0	88.5	86.5	90.2	88.5	89.5	87.5	89.5	87.5
25.0	89.5	87.5	91.0	89.5	90.2	88.5	89.5	87.5
30.0	89.5	87.5	91.0	89.5	91.0	89.5	90.2	88.5
40.0	90.2	88.5	91.7	90.2	91.7	90.2	90.2	88.5
50.0	90.2	88.5	92.4	91.0	91.7	90.2	91.0	89.5
60.0	91.7	90.2	93.0	91.7	91.7	90.2	91.7	90.2
75.0	92.4	91.0	93.0	91.7	93.0	91.7	93.0	91.7
100.0	93.0	91.7	93.6	92.4	93.0	91.7	93.0	91.7
125.0	93.0	91.7	93.6	92.4	93.0	91.7	93.6	92.4
150.0	93.0	91.7	94.1	93.0	94.1	93.0	93.6	92.4
200.0	94.1	93.0	94.5	93.6	94.1	93.0	94.1	93.0

NEMA Standard

Pump Primer

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Dynamic Sealing - Introduction

Dynamic sealing, relative to centrifugal pumps, occurs at the point where the shaft used to rotate the impeller passes through the casing wall. Some sort of restrictive device must be incorporated into the pump housing to limit liquid leakage from the housing at the shaft entry and prevent air from entering the housing at the same location. Dynamic shaft sealing is the number one mechanical problem in maintaining satisfactory pump operation.

To assure proper sealing performance, the pump's rotating element should be checked periodically during seal maintenance procedures. The seal manufacturer's tolerances and instructions must be followed. Too often, only a brief glance at the external portion of the shaft sleeve is made. It is recommended that the shaft runout also be checked. Runouts exceeding a total indicator reading of .006 inches may need to be reduced by replacing either the shaft, bearings or, in the case of small close-coupled pumps, possibly the entire motor. Other readings should be taken to assure proper shaft alignment with the housing, i.e., the stuffing box face should be within .002 inches perpendicular to the shaft, and axial thrust movement should be no more than .005 inches.

The area surrounding the point at which the shaft enters the pump is commonly referred to as the stuffing box area. It may, depending upon pump design, be either a separate component or an integral part of the pump casing. Initially, the stuffing box took the form of a cylindrical recessed housing which accommodated a series of packing rings around the shaft or shaft sleeve. The packing, being resilient, can be compressed to the desired "diametric squeeze" fit by a gland capable of adjustment in an axial direction. This method of sealing led to the term "stuffing box."

During operation, the fluid being handled flows through the slight clearance between the moving parts (shaft sleeve) and the stationary packing, thereby acting as a lubricant for the packing. The fluid also removes some of the heat resulting from the friction occurring between the compressed packing and the rotating shaft sleeve.

When the lubricity of the liquid being pumped is unsatisfactory, external lubrication may be introduced into the stuffing box depending upon the specific installation. This can range from grease to an induced circulation system. The external lubrication must be introduced at pressures of 5 to 10 psi greater than the prevailing operating pressure in the stuffing box area.

Shaft Protection

It was customary to equip pumps with inexpensive, replaceable shaft sleeves to protect the shafts from normal friction wear. This sealing arrangement, illustrated in figure 1, has been in existence since the development of the modern centrifugal pump in the late 19th century.

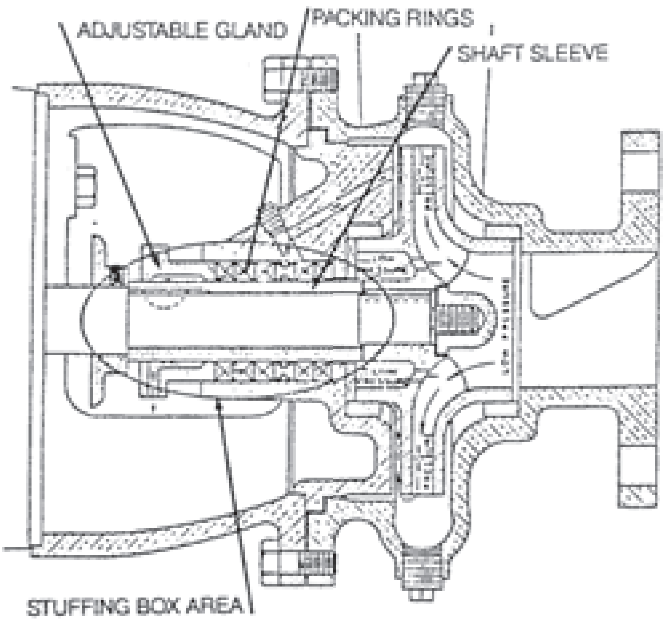

Figure 1

Packing is still employed for a substantial number of applications. Current packing materials include animal, vegetable, mineral and synthetic fibers along with metals containing dry lubricants and binders in combinations sufficient to handle most fluids in varying pumping conditions.

In recent years, the removal of asbestos as a packing material left somewhat of a void in this type of sealing arrangement. Teflon braided filament, while chemically inert and with a high lubricity factor, is very sensitive to compression, resulting in profuse leaking, burning up or scoring of the shaft sleeve within a minimum adjustment range. Graphite filament yarn appears more pliable and has outstanding self-lubricating and heat-dissipating characteristics. Consult packing manufacturer's recommendation charts for specifics. Today's charts not only include the recommendation for a particular medium but list the PV (pressure velocity) limit, temperature limit and pH range as well.

Mechanical Sealing

Mechanical sealing has come into its own within the last 35 years. Originally developed for applications such as auto cooling system pumps when ethylene glycol was introduced as a coolant, the mechanical sealing arrangement practically eliminated shaft leakage of the relatively expensive liquid — a significant improvement over previously used stuffing boxes.

The mechanical seal is comprised of stationary and rotating members (figure 2), each with its own sealing responsibilities. The stationary portion of the seal handles the stationary area between the pump's gland and the seal's seat (usually, an elastomeric cup, O-ring or gasket). The rotating portion is responsible for the rotating area between the shaft sleeve and the seal's rotating assembly (usually an O-ring, V-ring, bellows, wedge, etc.).

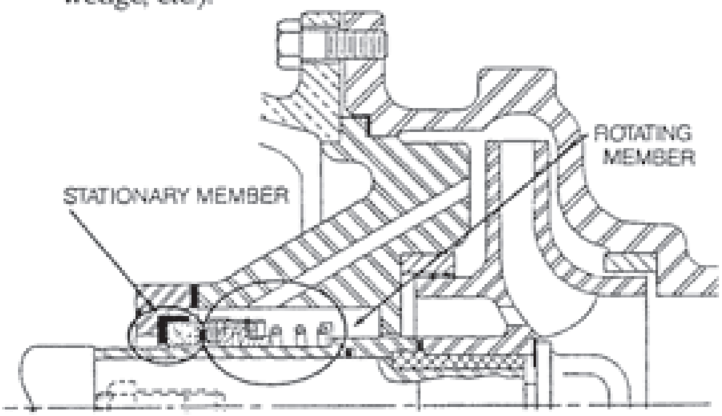

Figure 2

In a relative sense, each of these seal members are stationary within themselves. Primary sealing occurs on a plane perpendicular to the shaft centerline at the juncture of the seal's stationary and rotating members. The actual sealing surface in this plane consists of flat, lapped, highly polished faces on each component which are initially forced together by springs or similar devices. Material selection of the sealing surfaces is not only dependent on lubricity factors and resistance to corrosion but on the ability to withstand pressure-velocity situations, temperature, abrasive solutions, etc.

Sealess Sealing

A third method of sealing is found in the "sealesss" pump which as the name implies, has no sealing chamber. The unit consists of a pump/motor in which all moving parts (in both pump and motor) are in a sealed "can." A second can separates the stationary motor stator from the rotor.

Another sealess design employs magnets — one connected to the pump shaft and a second driven by the motor with a stationary non-magnetic containment barrier separating the magnets and isolating the fluid being pumped from the environment.

Seal Comparisons

Now that various seal arrangements have been identified, the capabilities, limitations and effects each of these has on pump performance are discussed briefly in the following paragraphs.

Packed Boxes

Packed boxes permit the highest leakage rate (45 to 60 drops per minute). The actual rate depends on pump size, RPM and liquid properties. Packed boxes are reliable to the degree that temporary adjustments and repairs can be readily made without delaying pumping operations. Ease of inspection enables early detection of deteriorating conditions.

Sealesss Sealing

The "sealess" arrangement is ideal when **no** leakage can be tolerated. While reliability of this sealing method has improved, its delicate design requires considerable preventative maintenance and disciplined care. Area of application is limited by the properties of the liquid being pumped. Bearing life can be a major concern because slippage is greater than in a standard sealed unit and volumetric efficiency is reduced since 3 to 5 percent of pumpage is directed to bearing lubrication and flushing.

Mechanical Seals

This type of sealing method is the current leader in the pumping industry, virtually monopolizing the field.

Mechanical seals, while not leak-proof (up to 5 drops per minute are acceptable), can be employed in a variety of arrangements and materials, limited mainly by the pump's design. Proper mechanical seal selection should be dictated by the following criteria: (1) pump size, type, speed and shaft diameter, (2) fluid being sealed, (3) maximum stuffing box pressure, (4) fluid temperature, (5) specific gravity, (6) fluid abrasiveness and (7) lubricating qualities of the fluid.

Most mechanical seal manufacturers publish recommendation charts based on years of field experience. This data should be thoroughly reviewed and a seal representative consulted before making the initial selection. The pump manufacturer can also assist in the appropriate selection, relying on his own empirical experience coupled with data relative to the pump's design.

Often, the pump user is most knowledgeable through first-hand experience with the product being handled. For this reason, maintenance people are an invaluable source for detemining both pump materials of construction and sealing practices.

Advancing Technology

The continuing development of new sealing face materials along with modern machining techniques which provide flatness in light bands and surface finishes in micro-inches have contributed greatly to improved sealing devices. Augmented by P-V (pressure velocity) charts and recommendation guides, pump users can face seal selection with far greater confidence than in the past.

Pump Modifications

Particular attention should always be given to further requirements or limitations which may be essential for the pump to function properly. In some cases, a pump other than the originally selected unit may be required to meet these modifications.

Such modifications may include: flush, vent, flush and vent, quench, flush and quench, injection, double seal, outside seals, tandem seals, balanced seals, etc. The seal manufacturer's guide will define each modification.

Auxiliary equipment which may also be required includes: heat exchangers, pressure lubricators, adhesives, separators, cut-off or alarm switches, pressure reservoirs, etc.

With this discussion, it becomes obvious why the selection of an appropriate mechanical sealing arrangement has literally become a science.

However, while a wide range of options/modifications exists, the majority of services can be capably handled with a standard seal arrangement. Still, accurate reporting of previous successes or failures by the pump user will provide valuable clues to the seal engineer during the selection process.

From a practical point of view, recommendations for new applications should begin at the most basic and economical arrangement, matching acceptable performance with adequate room for modifications or other seal types. Once the sealing arrangement is satisfactory, it may still be possible to go back to a basic stuffing box. Nevertheless, in a "first-time" situation, options should be available, if so needed, in the design and experimental testing stages.

Maintenance

Obviously, seal life is dependent on application, materials of construction and maintenance. To operate satisfactorily, the mating faces (both stationary and rotating) must be parallel to each other at all times and must always be separated by a film of sealing liquid. **Never run a seal dry.** It can be ruined in a matter of seconds. Some type of lubrication should also be present at the sealing surface.

When premature failure does occur, both the pump manufacturer and seal manufacturer should be contacted for evaluation. Often, the failed seal must be returned for analysis. In this case, all pertinent data regarding the operation will be studied, including hours of service and, at times, even samples of the liquid being pumped.

As new materials are developed and coupled with good seal design, some current modifications or auxiliaries may be eliminated. Obviously, this is not a "mature" product. Demands for greater speed, higher pressures and temperatures and new processing techniques continue to grow. Radical design changes are not expected until a more fundamental understanding develops of what really occurs between the stationary and rotating faces of the seal.

Conclusion

Just as, hydraulically, the system's characteristics coupled with the pump's characteristics determine flow, so will the seal selection, installation, operation and maintenance be directly related to the pump's sealing performance.

Pump Primer

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Maintenance - Introduction

Since long-term breakdown cannot be tolerated in most services, maintenance procedures and a contingency plan must be established in advance to reduce any economic loss caused by down-time.

During building and start-up, it is common to use outside personnel. It is during and/or immediately after this period that operating personnel should acquaint themselves with the pumping unit — particularly its running performance. This will be of considerable aid in establishing a standard for future reference. Instruction sheets, maintenance booklets and parts lists provided with the unit should be assembled in a file referenced to a specific pump.

All possible performance data should be recorded once the system has stabilized and is functioning properly. Suction and discharge pressure readings, indicated flow — if possible, seal leakage loss rate, bearing temperature, noise and vibration levels all provide input to a pump's performance in the system. Granted, it isn't likely that all this data can be measured. However, through familiarity with the unit, developing changes can be detected merely by relying on sight, hearing and touch.

From the start, operating personnel must be made aware that any change in the system, including the fluid being pumped, may have an effect on the pump's performance. It is, therefore, advisable to also record fluid temperature, specific gravity, viscosity, liquid concentration, percent of solid concentration, other additives and properties.

A proper maintenance procedure should begin with a file for each pump. All pertinent data discussed earlier in this issue relative to the pump, fluid handled and system should be included. Complete records of maintenance and repair costs along with a log of the unit's operating hours should be kept. In addition, complete pump identification — size, type, operating speed, manufacturer, serial number and material of construction should be noted. Pertinent comments and/or photos of specific problems can also be of interest.

Maintenance Procedures

Daily Check — possibly the most important inspection will be the daily or, in critical areas, even hourly observation made by your personnel of the following:

1. pressure reading and flow indicator
2. seal leakage rate
3. change in operating sound
4. change in bearing temperature

Minimum Inspection — typically a minimum inspection is made at 6-month intervals with results noted in the pump's maintenance file. The inspection includes:

1. check of mechanical seal or packing assembly
2. check of bearing lubrication

Annual Inspection — includes the two minimum inspection areas plus:

3. check of shaft sleeve wear
4. removal of packing/seal for inspection
5. bearing check
6. flushing and cleaning of internals plus drains, and sealing of water piping and other piping.
7. check of wear ring (or housing bore) and diametral clearance of impeller hub

Overhaul — should be performed periodically. Proper frequency depends on a number of factors: (1) type of service for which the pump is intended, (2) basic pump design, (3) fluid being pumped, (4) materials of construction, (5) number of start-stops and (6) typical period of operation. Some severe services may require monthly overhauls; other less-demanding services might need this type of maintenance every 4 or 5 years.

Contingency Plan

For inspection findings and breakdowns, a contingency plan should be developed. To begin with, an adequate supply of probable replacement parts should be kept on hand — depending on the extent to which you can afford to risk a shut-down.

A minimum requirement for pump maintenance should include the following items:

1. wear ring set (if applicable)
2. shaft sleeve
3. mechanical seal kit (complete with gasket set) or packing set (complete with gasket set)

For the average requirement, the following additional components should be added:

4. impeller
5. impeller screw
6. shaft (pedestal unit only)*
7. bearing set (pedestal unit only)*

* For close-coupled units integrally mounted on special motors, it is normal to contact the nearest Motor Service Center for replacement of shaft and bearings.

Where service cannot be interruped, a complete standby pump unit fully assembled (possibly even in a bypass line) is recommended since it is not reasonable to expect overnight service on all major parts from the factory, particularly where specials may be involved. The degree of risk in downtime must be resolved by the user. If a number of identical units are in service, the inventory of replacement parts and/or units may be reduced to a 1 to 3 or 1 to 4 ratio or even greater, depending on the service.

Beyond the economics of stocking critical replacement parts, it is recommended that an adequate maintenance training program be instituted. Someone — usually in maintenance or engineering — should become knowledgeable about pumps. Through training, daily exposure and inherent ability, the resident specialist can avoid repeating certain trial-and-error scenarios to solve pump problems. He can also more intelligently discuss problems with the pump manufacturer, thereby reducing the "language" barrier which might otherwise create delays.

Production personnel may be used for this function, although most of these people are so involved with other on-going duties that they are hard-pressed to thoroughly familiarize themselves with the science of pumps. However, they can be responsible for conducting "daily checks" and reporting changes or suspected problems to the proper source.

After resolving a pump problem, a complete description of the problem and remedy should be entered in the permanent file for that pump. Should a pattern of problems begin to appear, the pump manufacturer should be contacted. Options or remedies unknown to the user may be available. Also, photographs can be effectively used to provide a more accurate and graphic record of part failure than a written description.

Normal inspection practices may not always detect damage caused by corrosion or erosion. Frequent internal checks are recommended on initial services where this problem is suspect.

Ordering Replacement Parts

A manufacturer can best serve your replacement requirements if you identify both the part to be replaced and the pump for which it is intended. Include the serial number as well as other data appearing on the pump nameplate. Be specific — identify the part in question from a cross-section drawing usually included in the operating manual and use the indicated part number and nomenclature in your description.

Assembly and Disassembly

The following procedure, used in conjunction with the manufacturer's instructions, is basic to all centrifugal pump repair work. It should always be performed in a clean work area:

1. Disassemble each part carefully, taking extreme care to prevent further damage.
2. Inspect each part as it is taken off the unit and place each item in order of removal to help ensure proper reassembly.
3. Protect all finish machined surfaces.
4. Make necessary repairs or replacements and carefully rebuild. Each part must be clean — particularly the gasket and sealing surfaces. Again, follow the manufacturer's instructions precisely.

Commom Troubles and Their Causes

Pump operating troubles may be either hydraulic or mechanical. Hydraulically, a pump may fail to deliver sufficient pressure or may lose its prime after starting. Mechanically, a pump may consume excessive power, vibrate more than normal, generate noise, develop stuffing box difficulties or experience part breakage.

A definite relationship will usually exist between certain difficulties in both categories. For example, an increase in the wearing ring clearance (mechanical) will lead to increased leakage resulting in lower, less efficient hydraulic performance.

While the pump is the prime mover of a system, it is not always at fault when expected hydraulic performance isn't met. An analysis of the listed causes for trouble reveals that the pump is directly responsible less than half the time.

The following list is intended to assist users in determining the cause of any pumping trouble. Faulty installations can then be corrected and a clear description given the manufacturer if assistance is required.

1. No liquid delivered
   a. pump and suction line not completely primed
   b. speed too low
   c. required discharge head too high
   d. suction lift too high (should not exceed 15 feet dynamically of cold water unless otherwise guaranteed)
   e. impeller, piping or fittings completely plugged
   f. wrong direction of rotation

2. Insufficient capacity
   a. air leaks in suction pipe or stuffing box
   b. speed too low
   c. required discharge head too high
   d. suction lift too high (should not exceed 15 feet dynamically of cold water unless otherwise guaranteed)
   e. impeller, piping or fittings partially plugged
   f. insufficient positive suction head for hot water or other volatile liquids
   g. liquid viscosity too high
   h. mechanical defects — wearing rings worn, impeller damaged, packing defective
   i. wrong direction of rotation
   j. suction pipe entrance too close to surface of liquid

3. Insufficient pressure
   a. speed too low
   b. mechancial defects - impeller damaged, packing defective

c. small impeller diameter

4. Pump operates for a while, then quits
   a. leaky suction hose
   b. air leaking in through stuffing box - water seal plugged
   c. suction lift too high (should not exceed 15 feet dynamically of cold water unless otherwise guaranteed)
   d. air or other gases in liquid being pumped
   e. suction pipe and fittings not completely freed of air during priming

5. Pump takes too much power
   a. speed too high
   b. excessive liquid being pumped because required head is lower than anticipated.
   c. viscosity and/or specific gravity higher than specified
   d. mechanical defects - binding at wearing rings from distortion due to piping strains, shaft bent, impeller rubbing casing, stuffing box too tight
   e. wrong direction of rotation

Long-Term Storage

Ideally, a pump should be stored in a controlled atmosphere at an even temperature 10°F or more above the dew point with a relative humidity of 50% or less and with little dust and no harmful fumes present.

From a practical sense, a clean, dry location would satisfy most requirements.

Units stored in environments containing substantial dirt, dust, moisture or other harmful materials must be completely covered with heavy, transparent plastic. Several bags of silica gel should be placed inside the plastic wrap to reduce moisture.

Regardless of the storage environment, it is recommended that the rotating element be turned over by hand for several revolutions at least monthly.

Conclusion

In summation, use a systematic approach in establishing both a pump maintenance program and a contingency plan to contend with breakdowns. Also, don't overextend maintenance. . . specifically, don't over-grease ball bearings and don't overtighten packed stuffing box glands — just follow the manufacturer's instructions.

Best Advice — Be Prepared!

Pump Primer

Basic Information About Centrifugal Pumps

Ampco Pumps Co., Inc.
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Number 10 in a series.

by John H. Horwath
Chief Engineer, Ampco Pumps

Corrosion - Introduction

Tens of billions of dollars are lost annually in the US because of corrosion damage to metals. It is one of the most troublesome and costly, yet least understood, aspects of engineering design. More damage is caused by corrosion than by any other single factor.

Corrosion may be defined as the deterioration of materials through chemical, electro-chemical or mechanical-chemical attack. Although an extensive variety of chemical attacks can occur, this article will limit itself to the most common occurences involving general-purpose centrifugal pumping of corrosive media.

General Corrosion
General overall surface penetration by chemical reaction causes a uniform reduction of the pump casting's wall thickness. Selection of a more resistant material will normally reduce the net loss rate.

Crevice Corrosion
Intense localized corrosion which occurs within crevices and other shielded areas is categorized as crevice corrosion. A good pump designer will eliminate this situation as much as possible in the design stage.

Pitting Corrosion
Extremely localized attack which, many times, develops during stagnation periods (downtime). This type of corrosion can be devastating at the immersion line where deep cavities can rapidly develop in an irregular pattern.

Galvanic Corrosion
Occurs when two different metals in contact (or connected by an electrical conductor) are exposed to a conductive solution. To avoid this action, select combinations of metals in close proximity in the galvanic series. Also, avoid combinations in which the area of the less noble material is relatively small. Under some conditions, adding a suitable chemical inhibitor to the solution may reduce the reaction.

Galvanic Series

Corroded End (anodic end - or least noble)
- Magnesium
- Magnesium Alloys
- Zinc
- Aluminum
- Cadmium
- Steel or Iron
- 18-8 Stainless Steel (active)
- Lead
- Tin
- Nickel (active)
- Brasses
- Copper
- Bronzes
- Copper-Nickel Alloys
- Nickel-Copper Alloys
- Nickel (passive)
- 18-8 Stainless Steel (passive)
- Silver
- Graphite
- Gold
- Platinum

Protected End (cathodic - or most noble)

Erosion Corrosion

This type of corrosion is attributed to the removal of protective surface films (oxide layer) by local, high-velocity turbulent conditions, exposing the unprotected metal surface to a direct corrosion attack. This continuing action, coupled with an accelerated erosion rate brought on by a deteriorating surface condition, leads to premature failure of the pump.

Cavitation corrosion, described in an earlier article as the rapid formation and collapse of vapor bubbles resulting in implosions, is, in essence, a form of erosion corrosion.

Many factors influence the resistance of materials to corrosive solutions. These include: temperature, concentration, degree of aeration, catalytic contaminants, solids in suspension, velocity, stagnation period, influence of re-circulation and equipment design.

Obviously, good corrosion resistance is a top prerequisite for handling corrosive media. Since it is normally impractical to totally eliminate corrosion, the "smart" engineering approach is to utilize a "fully" resistant material of construction.

Material Selection

Be aware that the proper material selection is ultimately the user's responsibility. Only the user can control the liquid being pumped relative to concentration, temperature, etc. The responsibility of the pump manufacturer is to provide the selected material in a physically sound condition and to the correct chemical compositon while meeting specified hydraulic requirements. In practice, a creditable pump supplier will provide guidance based on his experience with previous applications involving similar services.

It is recommended that the material selection be carefully considered, lacking any previously acceptable history. Information sources which can be referenced include: (1) first-hand, in-plant experience, (2) similar applications at other locations, (3) Hydraulic Institute Corrosion Chart, (4) Mechanical Seal Recommendation Chart, (5) pump manufacturer's experience, (6) appropriate literature, texts and similar references.

All too often, none of the available data will specifically meet the stated conditions, particularly where solutions containing several chemicals of varying concentrations are being pumped. This, when added to a temperature increase of as little as 10°C, can double the corrosion rate.

The presence of dissolved air or other oxidizing agents in the liquid can alone be responsible for causing an accelerated corrosion rate in an otherwise assumed-to-be-acceptable situation. Minute traces of an element involving only several parts per million can also, under certain conditions, be sufficient to cause a severe corrosion attack where no serious reaction would normally be expected based on the stated basic composition of the solution.

By the same token, pitting corrosion occurring during an extended down-cycle period can, many times, be reduced immeasurably by draining and flushing the pump on entering the "down" cycle.

Corrosion Testing Vital

After initial selection of acceptable materials of construction, it is advisable to run corrosion tests in an environment similar to actual pumping conditions. Test racks can be assembled for this purpose. The services of a consultant might also be retained to perform laboratory measurements of such electro-chemical corrosion phenomena as potentiodynamic polarization scans and Tafel Plots. These and other techniques like pitting scans and corrosion behavior diagrams using micro processor-controlled instrumentation will provide results in hours instead of months required by typical test rack methods.

While none of these methods will match actual pump conditions, they will usually eliminate the least desirable materials of construction.

Pilot tests can also be performed in the lab, using a model pump to handle the liquid solution. Even this, though, is no substitute for on-site, full-scale tests under normal operating conditions.

Corrosion Rate vs Cost

In selecting a suitable corrosion-resistant material, the most important comparison will be the projected corrosion rate vs the cost of the material. No single material can be recommended for the entire spectrum of corrosion service. Begin the selection process with a broad overview, taking into consideration corrosion resistance, pump efficiency and cost factors based on the pump's anticipated life expectancy. To pay 15 times as much for a unit with only twice the life expectancy based on cost alone is not a sound investment, yet this scenario does exist.

The following table may be helpful in the selection process:

Corrosion Rate

Rate	Comment
0 to 0.006 inches/year	Considered fully resistant.
0.006 to 0.020 inches/year	Extensive preliminary testing required before acceptance.
Over 0.020 inches/year	Not recommended — consider alternate materials.

Beyond corrosion rates, workability of the material is also important. Some materials may be acceptable from a corrosion standpoint but entirely unacceptable from a casting or machining aspect because of the limitations they impose on a centrifugal pump's proper hydraulic design, i.e., being unable to provide efficient flow passages. In this case, higher energy costs would result and pump life would be reduced due to the erosive action of the fluid on the inefficiently designed impeller and/or housing.

Be sure to keep informed as new materials are developed to handle higher processing temperatures, pressures and overall requirements.

Non-Metallic Materials

Other methods of resisting corrosion employ non-metallic coatings, plastics and elastomers. Each has limited areas of application and perform effectively only if precautions are taken. Coatings cannot include pin holes which expose the base metal and plastics lose strength as temperatures increase. Also, phenolic and epoxy parts have a tendency to "creep" and lose dimensional integrity.

Reactions that may occur are quite different from the attacks on metallic materials. They include: (1) decomposition, (2) oxidation and disintegration, (3) surface hardening, (4) permeation, (5) surface attack, (6) severe swelling, (7) sub-surface degradation and (8) loss of strength. It becomes obvious that these reactions may be difficult to evaluate, particularly in plastics where sub-surface degradation can occur.

Conclusion

To summarize, be as factual as possible in describing the media to be pumped. Don't overlook previous experience or similar applications to reach appropriate solutions. Discuss your situation with both the pump and mechancial seal manufacturers. Look for a consensus of opinion before selecting a specific material, even for test purposes. Conduct the corrosion test under conditions closely approximating actual pumping, particularly in respect to velocity.

Final material selection must be based, not only on corrosion resistance, but on workability of the material and overall cost (this includes initial cost, replacement parts, downtime, life expectancy, maintenance time, etc.) as well as the reliability factor.

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Sound - Introduction

A general definition of what constitutes sound is in order before relating this subject specifically to pump operation.

Sound is defined as mechanical energy transmitted by pressure waves in an elastic medium such as air or water. Sound is constantly around us and is transmitted to our sense of hearing. Sound allows us to react accordingly, depending on our physical ability to hear the sound and our interest in the sound being developed.

Most of the sounds we hear are caused by objects vibrating in contact with air. Sound propagation in air can be compared to waves in water. The waves spread uniformly in all directions, decreasing in amplitude as they move from the source. Obstacles placed in their path will disrupt the flow pattern in that particular area.

Disrupting sounds, many of which have become prevalent in today's society, can be classified as noise. Basically, two types of noise exist: (1) airborne noise which is carried through the air in the form of pressure fluctuations and (2) structure-borne noise which is transmitted through a solid body in the form of vibrations.

Measurement of airborne noise is conducted through the use of microphones and sound level meters at specific frequencies.

Measurement of structure-borne noise requires the selection and mounting of transducers at the point(s) of highest broad-band volume and direction.

Sound is universally measured in decibels (dB). While four decibel scales of measurement exist, only the "A" network is commonly used today. Decibel scales are related to an absolute pressure scale. In terms of absolute measurement, the normal ear can detect a sound pressure of 20 millionth of a pascal - 0 dB(A) - which is a factor of 5,000,000,000 less than the standard atmospheric pressure of 14.7 pounds per square inch. At 100,000,000 millionth of a pascal - 134 dB(A) - the ear would be at the threshold of pain.

Using the "A" network, examples of a broad range of sound sources are offered below:

	Decibels dB(A)
Jet Plane	130
Boiler Factory	115
Heavy Traffic	100
Normal Office	80
Quiet Office	60
Whisper	40
Threshold of Hearing	0

Working Standards

The Federal Government (Walsh-Healy Act of 1972) has standards for occupational noise exposure (OSHA) that provide worker protection when sound levels exceed established limits. Feasible administrative or engineering control must be employed under these conditions. If such controls fail to reduce sound levels to those specified by OSHA, personal protective equipment shall be provided and used. OSHA allows no more than 90 dB(A) total noise at locations where employees are stationed for periods of 8 hours.

Pump Operation

Noise, while uncomfortable to the ear, can be a signal of potential trouble in the operation of a pump and/or system. Operating, maintenance and engineering personnel prefer a centrifugal pump system to be relatively quiet. However, because the pump is the initiating source of fluid movement, the universal tendency is to brand it as the basic cause when high noise levels are reached.

In reality, more noises will normally develop in the system than in the typical pump-motor combination, particularly if the unit requires less than a 25 HP driver.

Actually, noise generation in a pump-system arrangement inherently begins during the development of the pump's design as the hydraulic and physical characteristics are established and ends with the design and installation of the system along with its environment.

Hydraulically, centrifugal pump selection should be based on operaton close to the pump's BEP (best efficiency point). A pump's efficiency at the intended point of opertion has a direct effect on the noise developed by the pump. Other pump noise contributions may include: bearings, cut water clearance, suction and discharge velocities and, of course, impeller imbalance.

Another noise attributed to the pump is cavitation. This is not necessarily a pump problem if the pump meets the stated NPSH requirement and the actual requirement is less than that.

Reducing Noise in the Pump

Operation of the pump at its Best Efficiency Point obviously will aid in reducing the unit's noise level since the hydraulic efficiency is highest at the BEP capacity. In a good pump design, the resulting losses at the BEP are, primarily, friction losses. Away from the BEP, shock losses occuring at the point where liquid enters and leaves the impeller become an ever-increasing factor.

Some noise reduction can be expected if an impeller diameter, running in close proximity to the casing cut-water, is slightly reduced, thereby decreasing the high velocity and turbulence incurred at the cut-water separation.

The smoother the transition throughout the pump's flow path, the less shock and turbulence occur. Since the energy lost during this transition is responsible for a substantial amount of noise, less shock and turbulence translate to less noise and higher efficiency.

Mechanically, a balanced impeller will also provide smoother, quieter operation.

Formulas have been developed for estimating the PWL (Sound Power Level) of centrifugal pumps based on the hydraulic horsepower and efficiency of the pump at the point of operation. Once the PWL is established, further empirical steps can help estimate the SPL (Sound Pressure Level) in dB(A) units.

Reducing Noise in the Motor

Motor fan noise is usually the major contributor to the noise generated by a pump-motor unit. A thick plastic or metal fan guard will appreciably reduce noise from this source. Adding some type of mastic will also reduce the sound level when particularly thin metallic designs are involved.

Other methods of sound reduction include: special precision ball bearings, dynamically balanced rotors, air intake mufflers and acoustical enclosures. As the horsepower requirement increases in a pump-driven operation, it is normal for the driver noise to be substantially higher than that of the pump and, therefore, the dominant noise factor.

Reducing Noise in the System

Controlling the noise level in a circulating system requires careful attention to every element in the system, i.e., valves, pipe supports, size of piping and controls used in addition to the pump and its immediate auxiliaries.

Reducing the rate of flow will decrease fluid velocity and result in a lower noise level.

Noise caused by sudden changes in velocity and direction of the flow stream can be reduced by smoothing out these areas. Keep in mind that changes of this kind will not only favorably lower friction losses, they will also extend the life of the system as less wasted energy is expended.

After becoming familiar with the factors contributing to noise in a system, it helps to have a sense of "feel" to correctly detect and quickly eliminate or reduce the noise problem. This is usually an acquired skill.

Vibration

The most common source of vibration in rotating machinery is unbalance followed by misalignment between the pump and motor shafts on frame-style units. Often, ultra-quiet bearings may dampen the resonance and/or amplification to a tolerable limit.

Offensive vibrating sources in machinery must be isolated and reduced or eliminated. When vibrations are transmitted to sheet metal processing equipment, the sound can be greatly amplified.

Structure-borne noise can often be dampened by using resilient mountings on the pump-motor unit and flexible connections in close proximity to the pump at both the entry and discharge.

Summary

As mentioned earlier, sound waves spread evenly in all directions, decreasing in amplitude as they move from their source. When encountering an obstacle in its path, part of the sound will be reflected, part absorbed and the balance transmitted through the object. How much sound is reflected, absorbed or transmitted depends on the object, its size and the wavelength of the sound.

Airborne noise is usually controlled with acoustical absorbing (isolating) material positioned within the room or area of the originating noise source.

To this point, we have dealt with the noise emitted by a single pump and its system which may or may not be in the same location. Be aware that total noise at a single location may include other noise sources and their reflections from walls, ceilings and equipment. Noise at an employee's work station is an example of environmental noise and is a measurement of all sound to which that individual is normally subjected to.

Since sound is measured on the dB (logarithmic) scale, dB levels cannot be added or subtracted in the usual way. Information on this subject is readily available from manufacturers of sound measuring equipment, sound specialists, textbooks, technical articles, etc.

Pertinent Points

In a total approach to noise control, all possible vibration paths from the source to the ear must be explored. Thorough examination will assure selection of the most economical and desirable solution to a noise problem.

1. Noise in a pump operation is a system problem in which the pump is just one of many noise sources.
2. Select a pump to meet the hydraulic requirements at the pump's highest efficiency.
3. Isolate rotating equipment with dampening-type materials and techniques to reduce noise levels throughout the entire system.
4. When necessary, airborne noise can be reduced with sound reducing enclosures or proper room treatment, if feasible.
5. Noise levels drop geometrically with distance.
6. Reflecting surfaces should be treated to decrease sound pressures.

For an in-depth analysis of this topic, sound consultants are available as are short-term educational courses offered periodically by manufacturers of sound measuring equipment.

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Motors - Introduction

Basically, an electric motor converts electrical energy into mechanical energy. Since electric motors are the most popular prime rotating movers of centrifugal pumps, some fundamental information relative to their performance and requirements may be beneficial.

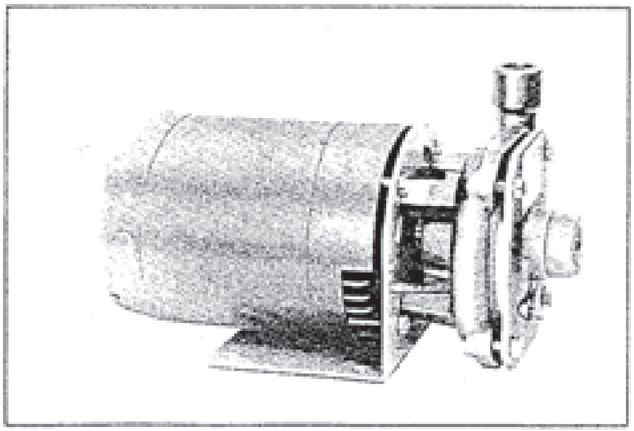
Because motors are usually quite dependable, little thought is given to their selection. Many motors are purchased as integral components of process equipment.

However, environmental and energy requirements can play an important role in motor selection. It has been estimated that electric motors consume about 60 percent of the electricity generated in the United States. This article will limit itself to the general purpose A.C. motor which is the most common industrial unit.

A.C. Motors

These motors are available in two size categories — fractional and integral horsepower. The great majority are 3-phase motors (up to 100 HP) operating at 460 volts although some fractionals are single-phase and operate from common-line voltages of 115 or 230 volts.

Overall, the squirrel-cage, constant-speed, 3-phase induction motor is the most common electrical power source. In brief, this unit consists of two windings. One, referred to as the stator, is stationary and is embedded in slots in the motor frame. The second, or rotor winding, consists of a series of copper or aluminum bars embedded lengthwise in the iron core of the rotor which is the moving element of the motor.



Ampco TYpe KC2 close-coupled pump mounted to a fractional horsepower motor.

A.C. current is applied to the windings of the stator pole, producing an electro-magnetic field which varies in strength and polarity as the current flows in alternating directions. An electrical field is created within the rotor, producing a force upon the rotor's conductors which tends to make the rotor revolve in the same direction as the electrical field.

The squirrel-cage motor is characterized by inherently good speed regulation, low cost and uncontrolled acceleration. Five standard types are manufactured to NEMA Design Specifications. They are referred to as NEMA Design A, B, C, D and E.

NEMA Design B is considered the standard general-purpose motor. Applications include driving centrifugal pumps (both francis and radial impellers), fans, blowers and machine tools. Design B motors have normal starting current acceptable to most power systems. This design has relatively high breakdown torque and low slip. Low-slip motors are intended for relatively constant loads and long running times. Typically, the speed variation from no-load to full-load is less than 5 percent. The popular speeds for centrifugal pump-motor units operating at 60 hertz are 3600, 1800 and 1200 RPM. At 50 hertz, 3000 and 1500 RPM are common speeds.

Environment

Next to speed and horsepower, the environment under which a motor will operate is a most important consideration in the motor selection process. Environment determines the type of motor insulation and motor enclosure that should be used. Both decisions must be made after careful analysis to ensure longest possible motor life at the least possible cost.

Ambient temperature, immediate air contaminants, altitude and other environmental factors must also be satisfied through modification of the electrical and/or mechanical motor components.

Various types of motor enclosures used for protection are included in the NEMA Standards for Motor Types.* Enclosures are rated by the degree of protection provided.

While a number of enclosures and their variations exist, this discussion will be limited to the most common types. More detailed references can be found in

* NEMA STANDARDS — Developed by the National Manufacturer's Association, a non-profit trade association of manufacturers of electrical apparatus and supplies engaged in standardization to facilitate understanding between the manufacturers and users of electrical products.

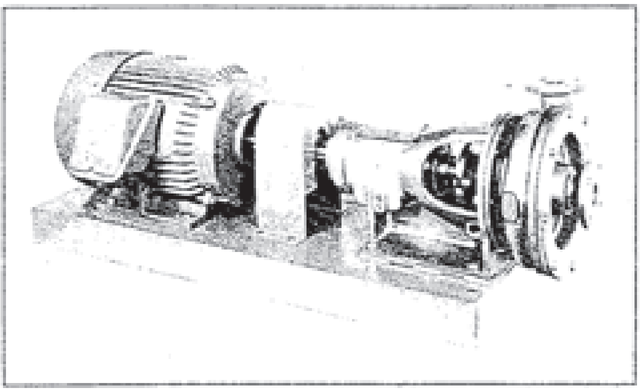
the NEMA Standards Publication No. MG1, Section 1.25 (Open Machine), Section 1.26 (Totally Enclosed Machine) and Section 1.27 (Machine with Encapsulated or Sealed Windings).

Open Enclosure

The most economical motor, at first cost, is the standard open type. Open units are constructed to allow unhindered operation when liquid or other particles strike or enter the motor enclosure at any angle from 0 to 15° downward from the vertical.

TEFC Enclosure

Totally enclosed motors are usually more expensive than open types. TEFC enclosures are designed to prohibit the free exchange of air between the inside and outside of the motor case. This enclosure is required when the environment is hostile to internal motor parts. It is the most common enclosure in the process industries.



Ampco Type HBH frame pump direct-coupled to a standard integral horsepower, totally-enclosed motor.

Encapsulated or Sealed Windings

These units have random windings filled with an insulating resin which also forms a protective coating. Application of this style is intended for exposure to environmental conditions too severe for typical varnished windings.

Many motors are overspecified relative to enclosures and protection. Careful analysis of requirements should precede motor selection. Be aware that, under the general classifications described above, many subclassifications exist which may be offered by individual motor manufacturers.

Insulation and Temperature Rise

The basic purpose of the insulation system is to keep the electrical conductors from coming in contact with each other and the motor frame. Any type of contact would create a short circuit and motor failure. NEMA lists four insulation classes designated as A, B, F and H. Temperature ratings of each classification are: Class A (105 °C), Class B (130 °C), Class F (155 °F) and Class H (180 °C). Effects of temperature on winding insulation life are decreased perceptibly when the insulation temperature is reduced or when a higher class insulation material is employed. Total temperature is the sum of the ambient temperature, rise in temperature through resistance, and the hot-spot allowance. Therefore, a Class A insulated (105 °C) motor operating in 50 °C ambient temperature with a 10 °C hot-spot allowance would be limited to a 45 °C temperature rise.

Today's "T" frame motors have higher surface temperatures than motors of older design. Typical values of full-load surface temperatures may range from 80 to 100 °C. This in itself, though too hot to touch, does not necessarily indicate an overload.

In addition to insulation failure caused by thermal degradation, failure can also be caused by such effects as chemical attack and moisture.

Cooling

Open drip-proof motors draw in outside air and pass it through internal components to remove heat generated during operation.

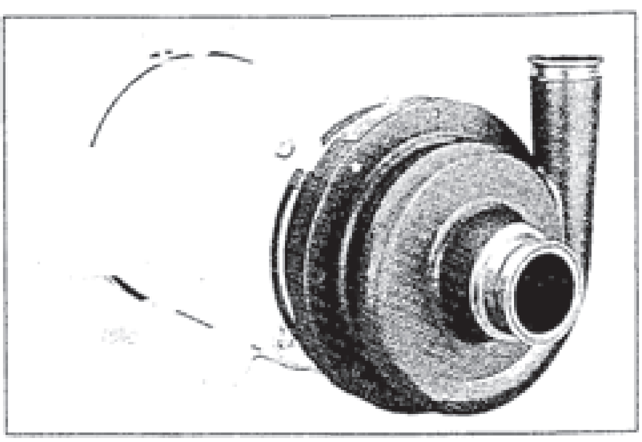
Cooling of TEFC motors is accomplished with a continuous flow of air over the outside surface of the motor. The air flow is generated by a fan mounted on the rear of the motor shaft. Usually, the motor enclosure also provides breather or condensate drain holes to remove accumulated moisture.

Motor Frames

NEMA has established dimensional standards for both open and enclosed motor frames through 150 HP and for close-coupled motor units through 60 HP.

When specifying motors, close attention must be given to the entire frame size designation, particularly when close-coupled motors are involved. Standard dimensions for Hydraulic Institute /NEMA Type C, Face-Mounted Motors shall be in accordance with NEMA Standard MG 1-18.615. A special classification offered by several motor manufacturers is usually identified as the "West Coast" pump motor. Within the above NEMA designation, two types of shaft extensions (JM and JP) are available.

Close-coupled pump motors are built to facilitate mounting of the pump impeller on the integral motor shaft extension and the pump's housing adaptor directly on the motor's C-flange.



Ampco Type DC2 close-coupled pump mounted to an integral horsepower motor.

While overall standards for frames, C-flanges and shaft extensions are identical for all motor manufacturers, the bearings used may vary somewhat by size. This will have a direct effect on the radial and axial load limits of the motor bearings as well as on their life factor. In addition, there are other limiting factors which have to be considered relative to speed, size, lubrication, internal shaft design, etc.

Service Factor

The service factor is a multiplier applied to the rated horsepower which determines the maximum horsepower a motor can safely deliver. When a motor operates in the allowable overload range, the unit will probably have a greater temperature rise, a lower shaft speed and a change in efficiency.

In sizing a pump motor, the pump's performance date (curve) should be studied to assure adequate available horsepower for the specified application. Good pump design provides characteristics which will result in a tangential relationship between the pump impeller's head-capacity curve and the brake horsepower requirements as they apply to the pump's performance over a series of impeller curves.

Efficiency

Basically, motor efficiency is related to motor horsepower with higher horsepower units achieving in excess of 90% efficiency. In most cases, the efficiency of a motor is approximately constant when operated at one-half or more of its full load torque rating.

The energy not converted to mechanical power turns into heat within the motor frame. The heat must be removed from the unit by the cooling medium which, in a standard motor, is the air surrounding the motor. The motor's ability to dispose of the waste heat, or losses, is directly related to the temperature of the surrounding air which is referred to as "ambient" temperature.

Direction Rotation

General-purpose motors are usually made with symmetrical parts to allow rotation in either direction. Three-phase motors can be reversed by interchanging any two of the three incoming power leads to the stator winding. There is no standard direction of rotation since the phase sequence of the incoming lines cannot be predetermined. Motor-pump units should be carefully tested for correct rotation relative to the pump's requirements before being placed in service.

Motor Problems

The biggest single cause of motor problems is due to bearing failure — usually resulting from misalignment or faulty lubrication. Motors should be lubricated according to instructions supplied by the motor manufacturers. Be aware that too much lubrication can be as detrimental as too little. In practice, over lubrication results in more failures than under lubrication.

Close-coupled pumps can experience seal failure if not properly maintained. In certain case, it may be less expensive to replace smaller motors than to repair them.

Where severe moisture exists, space heaters can be employed on integral horsepower motors to keep the internal air of the motor, when in the idle mode, above ambient temperature. This prevents moist air from being drawn into the motor as the unit cools down in response to ambient temperature changes.

Motors should be inspected at regular intervals. Time and extent will depend of the degree of operation, type of service and the environment. Above all, keep the motor clean and ventilating openings clear.

Summary

In any new pumping application, the key to electric motor reliability is to carefully define the application requirements and then, based on those findings, select the appropriate motor type and size, enclosure, insulation system and bearings.

Extreme care must also be exercised in ordering replacement motors to assure that complete interchangeability exists. This includes meeting all of the specifications of the original unit including special materials of construction as well as other electrical and physical modifications.

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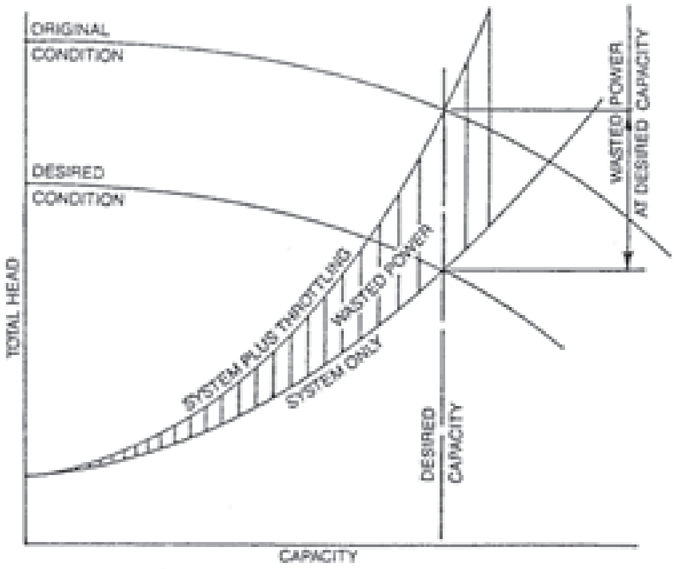
Variable Speed Drives - Introduction

Liquid processing industries are becoming increasingly aware of the need to consider and implement all energy saving efforts, especially during periods of rising energy costs.

In pumping systems, energy losses can result from valve throttling and orificing to balance systems having oversized pumps, and in attempting to utilize a single unit to meet the demands of several system requirements. It has been conservatively estimated that over 20% of the energy consumed by pump-motor units is dissipated in controlling flow rate alone.

The accompanying chart shows the power wasted in a typical throttling operation. Remember that the wasted energy, apart from consuming utility dollars, is causing needless wear and tear on the system and pump components in terms of erosion and heat buildup.



Reduction of this wasted energy can be realized through examination of existing systems and more precise determination of head requirements for proposed applications. Two options are available to modify the original condition hydraulic requirements: (1) change impeller diameter and (2) change pump speed. Either of these methods can be effective since a centrifugal pump, being a velocity machine, is primarily dependent on the tip velocity of the rotating impeller which will usually be the main vector of the resultant velocity.

Adjustable Speed Drives

Where versatility is required to meet several hydraulic conditions, it may be well to consider some type of variable speed arrangement. Process control loops are increasingly being governed by adjustable speed drives. Initially available only as a manual adjustment, the process has developed units that can react automatically to a variety of sensing devices capable of changing speed in response to signals received from such variables as flow temperature, pressure and liquid level.

Today, various types of adjustable-speed drives are employed in centrifugal pump process applications. The most common are: (1) mechanical, (2) eletromechanical, (3) solid-state d.c. and (4) solid-state a.c. Each of these is described in some detail below.

Mechanical

Mechanical variable-speed drives are the least expensive and oldest devices for varying the speed between a driving shaft and a driven shaft. Some of these drives will increase the speed of the output shaft at the expense of reduced torque.

Belt drives were, at one time, quite popular with efficiency reaching as high as 95%. Typically, adjustments were made manually. Belts also provided good overload protection since slippage would usually occur in these situations.

Among the negative aspects, heat buildup has always been a problem. Even when operation is within rated speed and horsepower conditions, heat is a major factor contributing to belt deterioration. Heat and belt wear tend to produce variations of 5-10% in drive ratios, therefore speed control accuracy is quite low for this method.

Other types of mechanical variable-speed drives include: geared transmissions (five-speed ratios), variable-pitch chain drives (which provide infinite variable speeds), traction drives and fluid drives. One overall complaint is that response is relatively slow.

Electro-Mechanical

Another method of varying motor speed is the generation of torque by means of eddy-current devices. This type of drive has been available for more than 60 years and is still quite popular today. Basically, this is an electric clutch (slip device) used to vary the degree of slip between a drive and driven element. The cost of these units is relatively high compared to mechanical drives.

The eddy-current drive converts a.c. line power directly to rotational power through the use of standard squirrel-cage induction motors. The full-load speeds of these motors typically run 2-4% less than the motors' synchronous speed.

To develop a variable-speed unit from a motor with a practically constant speed, an eddy-current clutch is added to the transmission. A tachometer generator, mounted integrally with the clutch output shaft, provides a signal proportional to the output speed. By comparing this signal to a setable reference signal, the level of excitation to the clutch coil can be adjusted. This allows the actual speed to be realigned to the set speed when the load is varied.

The efficiency of the eddy-current clutch is linearly proportional to slip. Speed ratio is usually limited to less than 2 to 1. The one distinct advantage of this style of drive is that the location of the clutch control is more accessible than that found on the more cumbersome mechanical drives.

Solid-State — D.C.

Speed regulation of d.c. motors goes back to the turn of the century. However, in the last two decades, solid state controls have become quite popular. The fact that speed can be controlled precisely over the motor's entire range makes this method attractive.

Electronic controls for d.c. motors are quite simple. The initial investment for this package is usually the most expensive. In operation, it becomes more cost effective - consuming only enough energy to satisfy immediate requirements. By comparison, in the case of slip-coupling drives, the excessive energy generated by the prime mover at its maximum speed load is wasted energy.

Negative aspects of these drives include: the limited modifiable features available with d.c. motor lines, the extremely expensive cost of enclosures other than open, and the problems inherent in the brush/commutator area relative to wear and corrosion.

Solid State — A.C.

This method consists of an a.c. motor and a variable frequency control capable of varying the rotational speed of the motor shaft by adjusting line power. The solid state - a.c. system has always been considered the most attractive variable speed drive except for the control itself. Within the last decade, several developments in the electronics field have reduced control complexities while, at the same time, significantly reducing price. As a result, this system is now less expensive than solid state - d.c. motor alternatives. Expectations are that further technological electronic advances within this decade will continue to drop prices. As this occurs, increased production will result in still further price reductions.

A.C. Motor Preference

The preference toward the a.c. motor rather than its d.c. counterpart is due to its basic simplicity. Furthermore, it is lighter, smaller, more durable and relatively inexpensive, making it most interesting as research continues to search for the ideal low-cost "black box." Components of the "black box" generally include a power convertor, power invertor, control regulator and a potentiometer. The power convertor changes a.c. to d.c. The power invertor changes d.c. to variable-voltage/variable-frequency a.c. power. The regulator controls the actions and response of the convertor and invertor. The potentiometer and on/off switch tell the regulator what speed is required.

Selection Procedure

The procedure for selecting the correct drive for a specific application can become quite complicated. Intangibles such as ruggedness, control flexibility, expected life and adequate environmental protection must be considered along with defined engineering parameters such as required maximum horsepower, torque and hydraulic requirements and efficiency.

As a first step, review your hydraulic requirements with the pump manufacturer so that a specific pump selection can be made. Areas to be considered at this time should include: maximum pump BHP-speed, maximum torque-speed and desired speed ratio. A determination of the maximum radial and thrust loads which would be developed must be compared to that allowable with the pump mounting or (in a close-coupled pump) the motor shaft and bearing arrangement.

Once all the pump parameters have been established, preliminary inquiries can be made of the potential drive and control equipment suppliers to determine what, if any, limitations may exist to meeting intended requirements.

Remember, too, that the hydraulic characteristics at various speeds may deviate from that calculated using the affinity laws. This is particularly true when a change in speed range greater than 10% is being considered. Another notable example is the pump's NPSH requirements.

While the drive speed can be increased beyond the pump's stated range, do not proceed in that direction unless the pump manufacturer advises that the pump can also exceed the stated limits.

With both base-mounted and close-coupled units, be aware that a 10% increase in speed above that designed for the pump can result in increases in head, torque, radial and axial loads in excess of 20%.

The speed of an a.c. motor is proportional to the signal frequency supplied by the drive control. Centrifugal pumps fall into the variable torque category - torque increases as the square of the speed.

Summary

Not only have variable speed pumps become genuinely automated control elements in process regulating systems, they have also eliminated other less effective elements.

Controlled pump systems (C.P.S.) reduce wear on pump components as well as within the system itself. In addition, the infinitely variable frequency regulation provided by C.P.S. offers significant savings in electrical energy consumption. The current consumption is effectively matched to shaft output of the pump in relation to flow rate and head requirement.

It appears that the versatile a.c. motor equipped with a reliable, less complex control will likely become the most popular method of speed variations in small process centrifugal pumps (under 100 HP).

As usage increases in all application areas utilizing this technology, the cost of these type units will continue to drop, making them all the more attractive. Ever-widening speed limitations, including potential operation beyond the standard 3500-rpm electric motor, could, in some instances, create further demand. This applies, in particular, to the prospect of smaller pumps operating beyond the so-called standard 3500-rpm line which has been generally adhered to over the last 50-plus years.

Improvements in bearings, shaft sealing materials and manufacturing processes will provide the mechanical necessities for utilizing higher-speed drives. This will result in a smaller, more economical pump providing the same basic hydraulic characteristics of the larger 3500-rpm units in common use today. Obviously, other factors, primarily the NPSH requirement, will limit the degree of reduction that can be made.

Pump systems which are frequency-controlled may be shown to be both energy saving and complimentary to desired product quality.

Advantages gained by utilizing a full variable-speed range include: (1) broader hydraulic range, (2) less wasted energy, (3) greater application versatility, (4) extended pump life and (5) greater use of existing pumps.

Do NOT extend a pump's speed range beyond the maximum limit indicated on the pump's nameplate without obtaining specific approval from the pump manufacturer's application engineers. A number of factors must be determined: (1) the loads and working pressures that would be incurred at the higher speeds, (2) the capability of the pump and motor components to withstand these new conditions, and (3) the torque and horsepower requirements at the higher speeds. In the case of close-coupled pump motors, the motor manufacturer should advise whether the motor bearings and shaft are appropriate for the anticipated radial and thrust bearing loads, shaft deflection at the impeller vertical centerline and the unit's critical speed.

Pump Primer

Basic Information About Centrifugal Pumps

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Number 14 in a series.

by John H. Horwath,
Chief Engineer, Ampco Pumps

Viscosity - Introduction

The viscosity of a liquid is that characteristic which tends to resist an internal shearing force. As motion (flow) of a fluid is produced by shearing forces, viscosity is associated with fluid motion.

There are innumerable ways to measure viscosity. This article will confine itself to the three most commonly used measurements which are (1) dynamic viscosity, (2) kinematic viscosity and (3) Seconds Saybolt Universal (SSU).

The unit of dynamic viscosity in metric measurement is the dyne-second per square centimeter, referred to as the poise. Numerical values are commonly expressed in centipoises (100 centipoises equals 1 poise).

Kinematic viscosity may be obtained by dividing the dynamic viscosity by the mass density (specific weight divided by the acceleration of gravity). The unit of kinematic viscosity in metric measure is the square centimeter per second, referred to as the stoke. Here, too, the more common terminology is centistokes (100 centistokes equals 1 stoke).

Seconds Saybolt Universal relates to the viscosity measurement in terms of time (in seconds) required for 60 cubic centimeters of the liquid to flow vertically through a special capillary tube.

The viscosities of most liquids vary appreciably with changes in temperature. The influence of change in pressure usually is negligible. The viscosities of fluids unaffected by the magnitude and kind of motion to which they may be subjected, as long as the temperature remains constant, are said to be Newtonian.

Effect on Pump Performance

Centrifugal pumps are quite sensitive to, and will usually cause, reductions in capacity and head as the viscosity of the liquid increases. Centrifugal pump performance is almost invariably specified by the manufacturer on the basis of handling clear, cold water with a viscosity in the area of 32 SSU. In most instances, viscosities under 100 SSU will not have a characteristic noticeably different from that of cold water which, by the way, is almost universally used as the fluid medium for factory testing.

Effect on System

System friction losses for viscous liquids increase drastically as the viscosity of the fluid media increases. Two sources for viscous friction loss data are: Hydraulic Institute Engineering Data Book and Cameron Hydraulic Data.

If the liquid being handled has an unknown viscosity, it may be best to refer to one of several text or reference book sources such as: Chemical Engineering Handbook-Perry or Handbook of Chemistry-Lange's.

You will find that viscosity measurements are taken by various means and that conversion to the more common methods described here is often necessary. The Hydraulic Institute Engineering Data Book (First Edition) lists some 20 types of measurement conversions to the basic SSU units.

Since a liquid's viscosity is related to temperature, it is necessary to include this factor as well in determining the real viscosity of the liquid at pumping temperature when incomplete data is given. In addition to the reference sources mentioned earlier, an article in Chemical Engineering (July 16, 1979) titled "Viscosity-Temperature Correlation for Liquids" is an excellent source of information since it covers 326 substances in the liquid state.

In developing expected system and pump performance characteristics, keep in mind that a rise in temperature will decrease the system requirement and increase the pump's performance characteristic—a beneficial improvement both ways.

Determination of Pump's Viscosity Characteristics

The capability of a centrifugal pump to handle viscous fluids may, depending on capacity and head requirement, be as high as 15,000 SSU (3300 centistokes). However, for general-purpose centrifugal pumps, which are in the majority, 4,000 SSU (880 centistokes) viscosity is a practical limit.

Be aware that the factors being presented here for performance correction are based on empirical data that may well deviate from the actual performance of a specific unit. Until substantial similarity has been experienced between the calculated empirical and onsite test results on unit combinations of similar design and size, a conservative approach should be taken in specing out the pump requirement.

The empirical-based data in charts 1 and 2 is applicable for centrifugal pumps of conventional hydraulic design in their normal operating range, with either open or closed impellers. Do not use this data for mixed-flow or axial-flow pumps or for pumps of special hydraulic design for either viscous or non-uniform liquids. Use only where adequate NPSH is available in order to avoid cavitation and only when handling Newtonian (uniform) liquids. Gels, slurries, paper stock and other non-uniform liquids may produce widely varying results, depending on the particular characteristics of the liquid being pumped.

Given the complete performance characteristics of a pump handling water, the expected performance when handling any liquid with a specific viscosity can be determined.

For example, in Chart 1, based on a BEP (Best Efficiency Point under 100 gpm) of 52 gpm and head of 50 feet, follow the lower vertical line established at 52 gpm upward to the 50 Head in Feet diagonal line. At the intersecting point, move horizontally to the indicated viscosity (200 SSU) and then vertically to the top of the chart, where intersections with C_H, C_Q and C_η will indicate the correction factors that are applicable to the head, capacity and efficiency in establishing the unit's BEP at 200 SSU viscosity.

The following equations are used for the viscous performance determination at the best efficiency point:

Capacity.................................... $Q_{vis} = C_Q \times C_W$

Head.. $H_{vis} = C_H \times H_W$

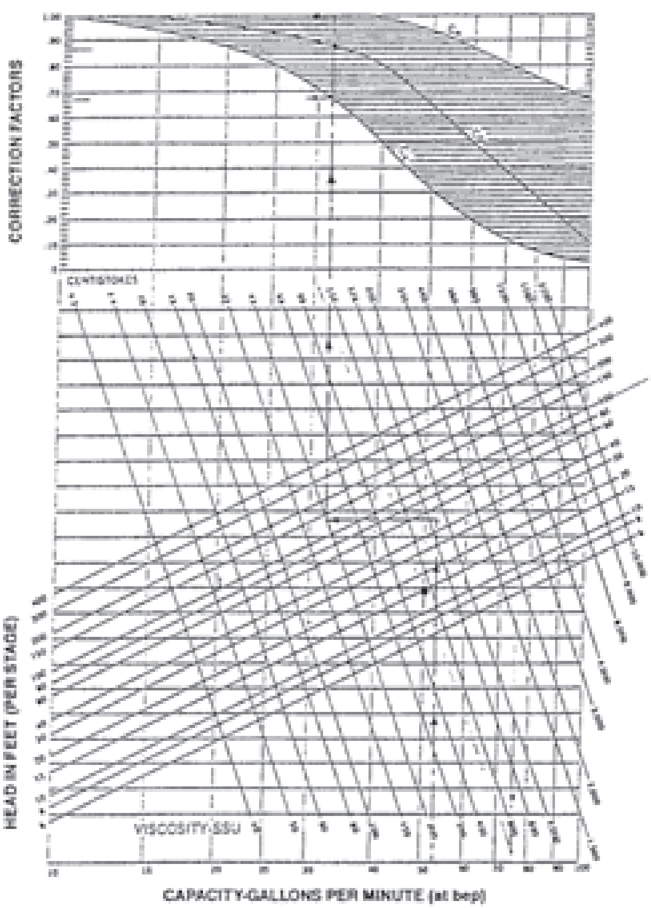
Efficiency.................................... $\eta_{vis} = C_\eta \times \eta_W$

$$bhp_{vis} = \frac{Q_{vis} \times H_{vis} \times \text{specific gravity}}{3960 \times \eta_{vis}}$$

To develop a portion of the expected characteristic, draw a curve similar in shape to the curve for water performance and with the same head at shut-off. The corrected efficiency point represents the peak of the viscosity efficiency curve which is similar in shape to that for water. Corrected brake horsepower curves are normally quite parallel to that of water.

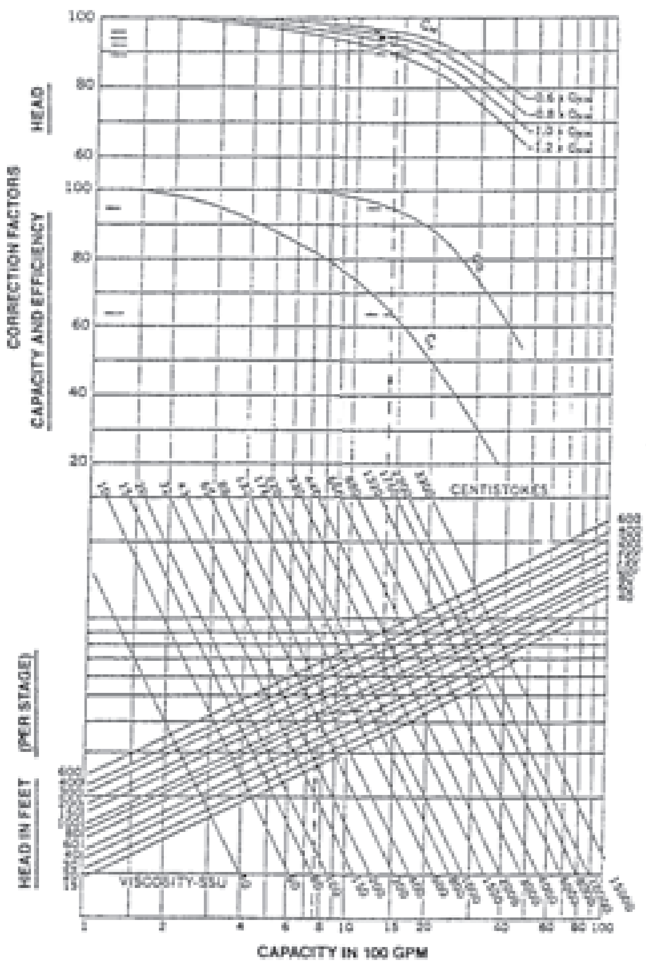
For units with a BEP capacity greater than 100 gpm, use Chart 2. The method of establishing correction factors is essentially the same except that, relative to head correction, there are four capacity points (60%, 80%, 100% and 120%) related to the BEP of the water characteristics.

Chart 1



ONE POINT DETERMINATION
PERFORMANCE CORRECTION CHART FOR VISCOUS LIQUIDS

Chart 2



FOUR POINT DETERMINATION
PERFORMANCE CORRECTION CHART FOR VISCOUS LIQUIDS

In this example, the pump water BEP is 750 gpm at 100 feet. Starting at a capacity of 750 gpm at the bottom of the chart, proceed vertically to the 100 foot head rating. At this point, move horizontally to the 1,000 SSU viscosity line. Again move vertically to the top of the chart. Here, intersections with C_H, C_Q and C_η will indicate the correction factors applicable to the head capacity, and efficiency in establishing the unit's 1,000 SSU viscosity performance at four points. Each point will normally have a different head (C_H) correction factor while the same factor will be used to correct capacity and efficiency at each point.

The following equations are used for the viscous performance determination:

Capacity.........

$$Q_{vis_{60\%}} = C_Q \times Q_{w_{60\%}}$$
$$Q_{vis_{60\%}} = C_Q \times Q_{w_{80\%}}$$
$$Q_{vis_{100\%}} = C_Q \times Q_{w_{100\%}}$$
$$Q_{vis_{120\%}} = C_Q \times Q_{w_{120\%}}$$

Head..............

$$H_{vis_{60\%}} = C_{H60\%} \times H_{w_{60\%}}$$
$$H_{vis_{80\%}} = C_{H80\%} \times H_{w_{80\%}}$$
$$H_{vis_{100\%}} = C_{H100\%} \times H_{w_{100\%}}$$
$$H_{vis_{120\%}} = C_{H120\%} \times H_{w_{120\%}}$$

Efficiency.......

$$\eta_{vis_{60\%}} = C_\eta \times \eta_{w_{60\%}}$$
$$\eta_{vis_{80\%}} = C_\eta \times \eta_{w_{80\%}}$$
$$\eta_{vis_{100\%}} = C_\eta \times \eta_{w_{100\%}}$$
$$\eta_{vis_{120\%}} = C_\eta \times \eta_{w_{120\%}}$$

$$bhp_{vis} = \frac{Q_{vis} \times H_{vis} \times \text{Specific Gravity}}{3960 \times \eta_{vis}}$$

Using the above brake horsepower formula, the viscosity BHP at each of the four performance points can be determined. Draw a smooth curve through these points. It should be similar to, but steeper than, the head-capacity performance curve and somewhat more parallel to the BIP-capacity curve. Its relation to the water BIP is also affected by its specific gravity.

Summary

Be aware that this subject is far from an exact science. The final measurement will always evolve from the pump's hydraulic flow pattern and the inherent characteristics of the liquid being pumped. There have been cases where even the general shape and direction of the developed curve have changed perceptibly from the anticipated characteristics.

It might also be helpful to review the material in article 6 of the Pump Primer series relative to practical viscosity limits of general-purpose centrifugal pumps.

Often, your own experience in handling a viscous material with a certain hydraulic design could provide your primary reference.

A most important point to remember is that a rise in liquid temperature will normally reduce its viscosity. If higher processing temperatures are permitted and an economical heat source is readily available, you may increase the pump's hydraulic performance while utilizing less horsepower. In addition, the system requirements will also be reduced accordingly, thereby providing an even more favorable situation.

Pump Primer

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System Piping Design - Introduction

There are two sets of inter-related conditions which need to be satisfied in establishing piping design in a system: (1) those imposed by the hydraulic operating conditions to determine the basic pipe size, material and wall thickness, and (2) those developed in laying out the physical piping system where thermal expansion, method of support and all other encountered loads must be accounted for in the structural design.

The first step is to establish applicable design parameters and code requirements such as:

- ► Liquid Characteristics
- ► Pressure
- ► Temperature
- ► Pipe Material Specifications
- ► Corrosion and Erosion Allowances
- ► External Loading
- ► Code Requirements
- ► Purpose of Flow
- ► Other

In the absence of codes, it is recommended that ANSI B 31.1.0 "Power Piping" be used as a reference. All assumptions made in establishing design criteria should also be so noted.

Hydraulic Pipe Sizing

Economical pipe sizes (primarily based on acceptable pressure drop) frequently cannot be used because of various constraints which must be placed on line sizes. Constraints commonly encountered include:

Erosion Limitation - High velocities can shorten the life of metallic pipe. The presence of abrasive solids can also be detrimental. In addition, large numbers of pipe fittings, with resultant high levels of turbulence, can produce erosive conditions. Obviously, lower velocities would be beneficial in this situation.

Process Control - To achieve good flow control in a pipeline, the control valve should absorb at least 30% of the total frictional pressure drop in the system.

Two-Phase Flow - Essentially, every chemical process involves some two-phase flow problems which, conceivably, could be of some significance, particularly in the suction side of a pump operation. The complexities in this area can be immense. References of patterns predicted by any general flow-pattern map may never be wholly successful. Forces that control one pattern (flow of large bubbles is dependent on gravitational forces and independent of surface forces) may be unimportant in other instances (flow of small vapor bubbles which depend basically on surface forces).

Flow Distribution - With the present day emphasis on large, low-pressure-drop unit processes and the ever increasing use of parallel equipment, it is essential that the design engineer become familiar with fluid distribution technology. He should be sure that his piping design will not cause an undesirable preferential flow to a particular piece of equipment.

Application - The application itself will often determine the necessary liquid velocity through the system with limitations relative to maximum and/or minimum velocities. Velocity restraints may be employed to maintain particle suspension, provide effective pipe cleaning action, move viscous materials, etc.

Line Velocities - After establishing basic parameters on all of the above, the desired velocity conditions can be formulated as a prelude to actual pipe sizing. In order to minimize the loss of head due to friction, the pipe size of suction and discharge lines should be at least a size larger than the pump's connections. Also, pipes should be arranged with a minimum number of bends with the longest possible radii.

Suction line liquid velocities in the area of 2 to 8 feet per second are common for low NPSH applications. A good general-purpose centrifugal pump operating at its best efficiency point will usually produce velocities of 9 to 10 feet per second, with some units running as high as 18 feet per second. Total suction line losses should not exceed 1 to 3 feet of liquid head.

Discharge line velocities will usually run somewhat higher, typically in the range of 10 to 25 feet per second.

Sizing

Good sizing methods are of economic importance since the cost of process piping typically ranges from 10 to 20% of the total plant cost. It is necessary to reach an economic balance between line sizes, pipe configurations, power requirements and capital costs.

While modern technology has replaced many sizing rules-of-thumb in recent decades, today's designer must still employ several guesstimated sizing techniques. Friction factors for aging piping, erosive wear patterns and corrosion effects are included in this category. However, even here closer approximations are being developed as on-going studies of extensive empirical data along with theoretical advances provide refined techniques.

In addition, losses through processing equipment must be provided by the manufacturer. Otherwise, it may be necessary to run tests to establish the pressure drop through these units, preferably at their design flow rate. Obviously, considerable time can be wasted developing a fairly accurate piping loss only to have to make a wild guess as to the drop through the process equipment itself. Since this drop can be substantial, accurate data is necessary to determine an adequate pump selection.

Pressure Drop (Friction Losses)

Two excellent sources for pipe friction loss data are *"Cameron Hydraulic Tables"* and the Hydraulic Institute's *"Engineering Data Book."* These sources cover metallic pipe and tube. Other means of conveying liquids such as hose, plastic piping, lined piping, etc. will require loss data from the specific manufacturers. Appropriate reference material should be developed and maintained as an aid in estimating system requirements.

Use caution in intermixing the two data sources described above as one is based on "old" piping, the other on "new". As piping usually deteriorates with age, the "old" piping data has a built-in safety factor. The "new" piping data does not and leaves the safety factor to individual discretion. Our suggestion is to add a 10 to 15% safety factor unless prior experience dictates otherwise.

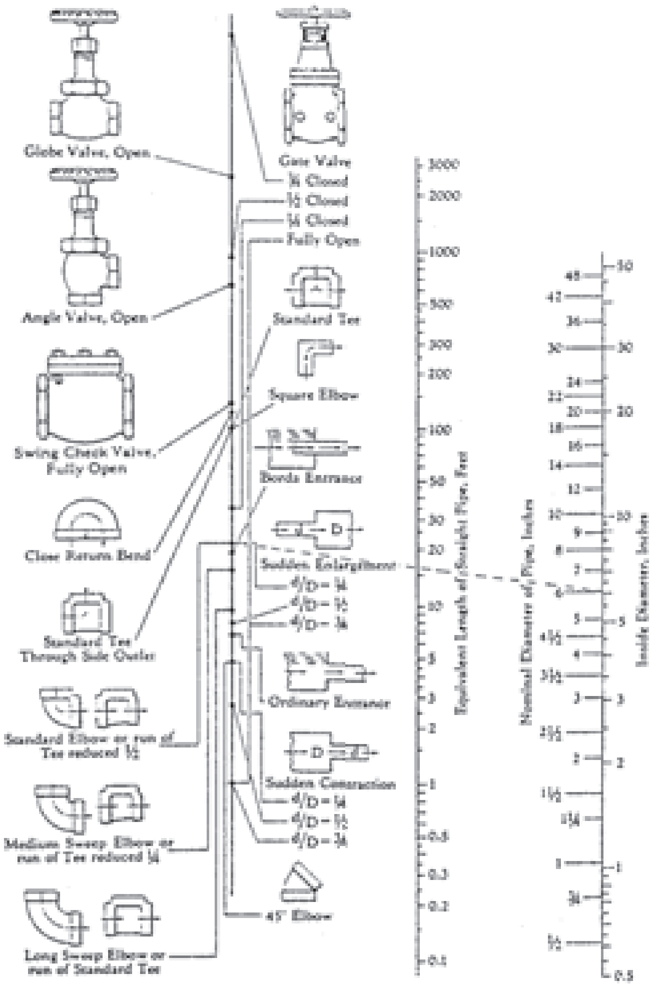
Several software programs have been developed for personal computers to determine pressure drop. It is essential that all constraints applicable to the service in question are positively addressed in the program.

Become familiar with the mechanics of your pipe sizing material. Be aware of the friction losses incurrred in steel pipe, cast iron pipe, copper and brass tubing, stainless steel tubing, various types of hoses and boots, and the plastics. Since each material employs its own nominal vs. actual inside diameter schedule, the friction factor can differ significantly.

Friction losses through pipe fittings are also available in the source material mentioned earlier. A simple method of accounting for the resistance of valves and fittings is to add to the pipe line an additional length equal to the pressure drop resulting from the valves and fittings in the line. Figure 1 provides an easy method for estimating equivalent line losses. Despite the limitations of this method, its simplicity often makes its use desirable. Since piping system design involves many variables which cannot be accurately evaluated, it is meaningless to strive for a high degree of accuracy in any one phase of the calculation.

Figure 1

Resistance of Valves and Fittings to Flow of Fluids



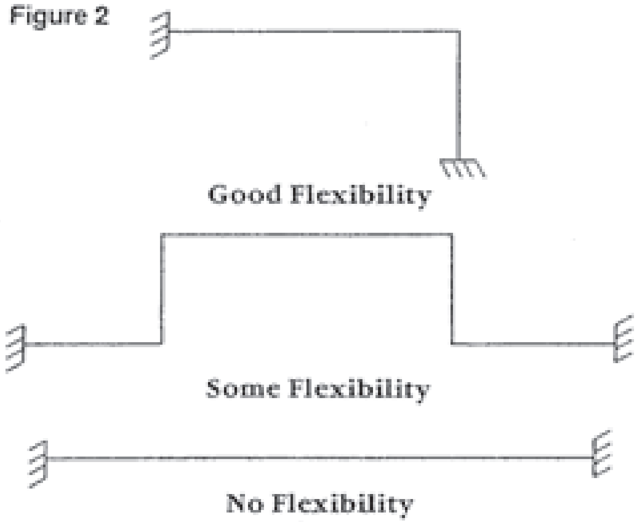
When developing new systems similar to existing ones where test data may be available, check calculated line losses against actual data. The same exercise can be followed after completion of a first-time system. After developing a history of actual data vs. calculated results, wayward assumptions which were made in the original calculations can be isolated and corrected. Calculations can then be made more accurately in the initial development stages of future piping systems.

Structural Design

Pipe Configuration - The basic rule is that any departure from a straight-line connection between two fixed terminals is an improvement. Flexibility in a pipe system is obtained by allowing a certain freedom of motion. It is important to arrange supports and other potential restraints so that this freedom is not inhibited.

Flexibility may be provided by changes of direction in the piping through the use of bends, loops or offsets, or by making provisions to absorb thermal movement by using expansion joints through those directions and magnitudes allowed by the joint design.

Figure 2



Bending stresses due to thermal expansion are not significantly affected by increasing pipe wall thickness. Rather, increasing wall thickness has an even more undesirable effect in that the reacting forces and movements are increased in a direct ratio.

Pipe configuration and size are also dependent on equipment requirements such as control valves and measuring devices. A thorough review should be made in the initial stages to assure that the developed piping adequately meets all process requirements.

A piping system may be cold springed or prestressed to reduce anchor forces and movements caused by thermal expansion. This is accomplished by shortening overall pipe length by any desired amount not in excess of the calculated expansion differential.

Analysis Requirement

Analysis will not be required if any one of the following conditions is satisfied: (1) duplication of a successful installation, (2) judged adquate by comparison with previously analyzed system, (3) piping system is uniform size with no more than two hangers, no intermediate restraints, non-cyclic service and satisfies:

$$\frac{DY}{(L-U)^2} \leqq 0.03$$

where

D = nominal pipe size (inches)

Y = sum of movements to be absorbed by pipe lines (inches)

L = developed length of line axis (feet)

U = anchor distance - length of straight line joining anchors (feet)

Analysis

Several piping analysis software programs are available for IBM and compatible PC's. However, in the absence of a suitable program, the time required to stress analyze piping configurations can still be substantially reduced through the use of isometrics. The pipe system shall be treated as a whole and the significance of all parts of the line, including restraints, shall be recognized.

In order to simplify the calculation of stresses and anchor forces for various configurations encountered in piping work, tables of common shapes can be found on pages 15 to 30 of *"Piping Design and Engineering,"* published by Grinnell Company, Inc.

A second series of tables on expansion bends and examples of their use are provided on pages 32 to 43 of the same publication.

When the configuration of a piping system is such that these tables will not apply, it may be necessary to use the basic equations of analytical methods found on pages 44 to 65 of Grinnell's publication.

For the analysis of special cases such as branch connections, variable or corrugated sections, hinged and moving anchors, etc., reference can be made to the bibliography of the publication for assistance.

The allowable stress range for expansion stresses shall not exceed the allowable stress range (SA) given in the following formula:

$$SA = f(1.25\,S_c + 0.25\,S_h)$$

See Section 102.3.2 (page 10) of *"Power Piping"* - ANSI 31.1.0 and Appendix A, Tables A-1 and A-2 for allowable stress values at the coldest (S_c) and hottest (S_h) temperatures to which the system will be exposed.

The stress due to thermal expansion which must not exceed the allowable expansion stress range is called "expansion stress" and is defined by the Piping Code as:

$$SE = \sqrt{(S_B)^2 + 4(S_T)^2}$$

where S_B is the bending stress due to thermal expansion and S_T is the torsional stress resulting from thermal expansion (this condition only occurs in multiple plane systems).

The Piping Code further states that the sum of the longitudinal stresses resulting from pressure, weight and other external loadings shall not exceed S_h.

Pipe Support

No firm rules or limits exist here. Support locations are dependent on pipe size, pipe configuration, the location of heavy valves and fittings, and the structure that is utilized for the support of the piping. Load effects due to service pressure, wind, seismic, etc. must also be taken into account.

The designer must exercise his own judgment in each case. The suggested maximum spans between hangers listed in the table below reflect the practical considerations involved in determining support spacing on straight runs of standard wall pipe.

Suggested Pipe Support Spacing

Nominal Pipe Size Inches	Suggested Maximum Span in Feet Water Service
1	7
2	10
3	12
4	14
6	17
8	19
12	23
16	27
20	30
24	32

Note 1. Suggested maximum spacing between pipe supports for horizontal straight runs of standard and heavier pipe at maximum operating temperature of 750° F.

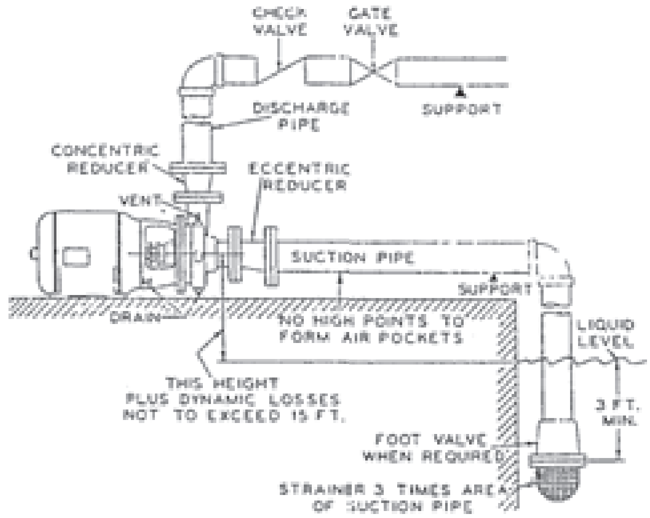
Note 2. Does not apply where span calculations are made or where there are concentrated loads between supports such as flanges, valves, specialties, etc.

Note 3. The spacing is based on a maximum combined bending and sheer stress of 1500 psi and insulated pipe filled with water and the pitch of the line is such that a sag of 0.1 in. between supports is permissible.

Typical Piping Arrangement

Shown below is a typical centrifugal pump arrangement. Note that the pipe to the pump suction and discharge nozzles should be supported near the pump so that, when the pipe is tightened, no strain will be imposed on the the pump casing. For the same reason, provisions should be made for pipe expansion and contraction on services handling hot or cold liquids. Proper support of the pump piping and allowance for expansion and contraction to eliminate casing stresses are of major importance since the centrifugal pumps designed today are small compared to the pipe sizes required to handle the pump's capacity.

Figure 3



PIPE AND FITTINGS ARRANGEMENT

Proper piping design incorporates many facets - all of which must be satisfied if proper performance is to be attained. Both hydraulic and structural requirements must be met if the system is to provide adequate service.

Remember that the friction loss in a pipe line is a square function of velocity. Doubling the flow rate quadruples the developed friction loss.

As in other topics covered in this series of Pump Primers, we have only provided a brief discussion of System Piping Design. You are encouraged to use the references mentioned to obtain a more thorough understanding of this subject.

Pump Primer

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Erosion - Introduction

While erosion has been acknowledged as a basic factor in pump design and application, little has been written relative to specific erosive effects which may occur when pumps are placed in actual service. Obviously, the infinite number of possibilities that coexist, coupled with the interactions of: flow (velocity, type, eddies, etc.), pressure, temperature, corrosiveness, wear resistance, properties specific to the media being pumped, and the actual point of operation all further complicate and, at the same time, narrow the application range of each service.

To begin with, erosion is defined as ''diminishing or destroying by degree.'' In the design and application of centrifugal pumps (kinetic machines converting mechanical energy into hydraulic energy through centrifugal action) there are several areas where erosion may occur. Pump surfaces of the rotating impeller and the stationary housing components are especially vulnerable to attack. Typical examples are shown in Figures 1 and 2.

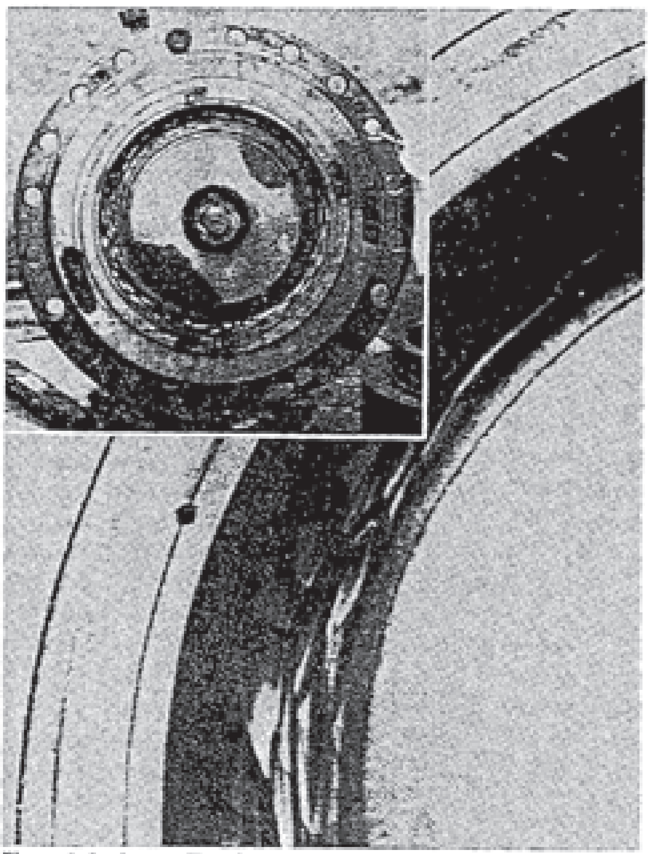

Figure 1: Stationary Housing

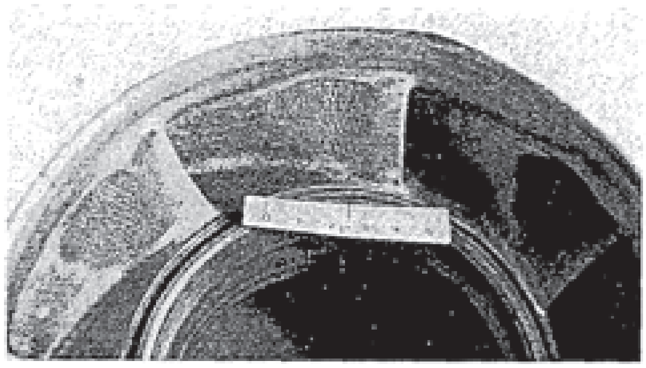

Figure 2: Rotating Impeller

Our discussion in this paper is centered on better understanding and effectively fighting the occurance of erosion relative to the pump and pumping system.

Basically, three forms of erosion occur in kinetic hydraulic equipment: (1) wear erosion, (2) corrosion erosion and (3) cavitation erosion.

Wear Erosion - This form of erosion refers to the loss of material from internal hydraulic surface areas of the pump. This action can usually be attributed to solutions which are poor lubricants or which contain abrasive matter - both contributing factors to adhesive and abrasive wear.

Abrasion involves flow particles which create wear debris, sometimes forming ahead of adhesive particles and referred to as ''cuttings''. Actually, in the secondary stage, much of the adhesive material will already have been plowed aside by the continuing flow augmented by wear debris, resulting in accelerated fatigue breakage of the embedded material.

A helpful first step in designing for wear resistance is to select high efficiency hydraulic equipment and utilize it in its highest range of efficiency. Once in operation, periodic observation of internal hydraulic surfaces of both rotating and stationary components, possible wear debris and general behavior of operating equipment should be made.

Next, the material requirement should be such that it will withstand wear as well as corrosion. All too often, corrosion will cause premature failure in what was originally a wear-resistant material.

Obviously, dramatic changes in the cross-sectional areas and direction (sharp turns and restrictions) of flow paths produce turbulence which, in turn, causes wear erosion. This action can occur in both the pump and the system. In essence, any time liquid flow is disrupted, an erosive condition is induced. A relationship also exists between velocity and erosion. Lowered velocities will reduce the erosion rate. A complex system with a large number of fittings may cause a high level of unwanted turbulence.

Eddies are disturbances of "*stream-lines*" in flow patterns. Since eddies incur additional losses, they should be suppressed or eliminated through suitable changes in the conduit.

Corrosion Erosion - This occurs because of the relative movement between a corrosive fluid and a metallic surface. As liquid velocity increases, the corrosion rate will generally increase as well. Mechanical effects add to the damage of the base metal. This type of erosion can completely remove protective films or layers developed in the initial chemical reaction. The result is a constant on-going surface attack that may also change the physical properties of the exposed surfaces. This explains why hardness itself is usually not a good criterion for predicting resistance to corrosion attack.

If the liquid contains insoluble solids, these particles themselves can physically remove the protective layer and initiate a constant corrosion attack on the exposed surface in the initial strike. Erosion here is actually a combination of abrasive and corrosive elements.

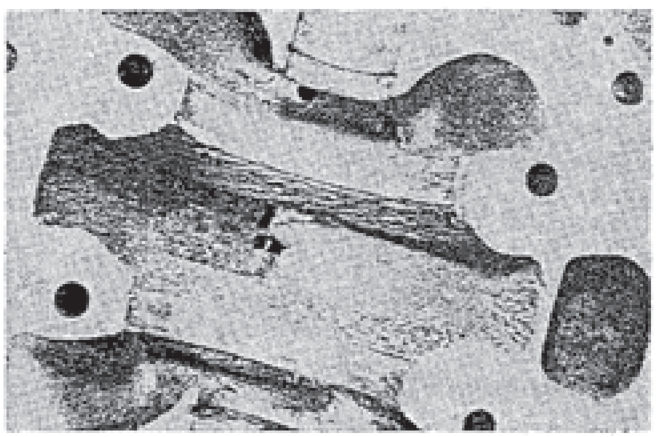

Figure 3: Corrosion Erosion

Cavitation Erosion - This was discussed briefly in article 3 of the Pump Primer series relative to inadequate NPSH being available in the eye of the rotating impeller. Recent investigations of further cavitation erosion have advanced a theory covering previously unexplained erosion cases in higher pressure sectors of the pump.

Referred to as recirculation, it is defined as a flow reversal at either the inlet or discharge tips of impeller vanes. Actually, there are two distinct internal flow patterns that can occur: suction recirculation, which is dependent on the pump inlet design, and discharge recirculation, which is dependent on the design of the impeller outlets.

Suction recirculation attacks, while similar relative to surface destruction, differ in the locations affected by insufficient NPSH at the pump inlet. In positive suction recirculation, the damage proceeds from the pressure side of the inlet edge of the vane through the metal to the low-pressure side. However, with inadequate NPSH, the attack begins on the low-pressure side of the blade and proceeds through the metal to the high-pressure side.

Discharge recirculation is due to the formation of vapor bubbles at the point where localized fluid pressures are reduced due to outward and inward flow in the same vane passage of the impeller. Specifically, the film shearing action at the interface between the inward and outward relative velocities develops a high velocity vortex in which cavitation occurs when the eye pressure is reduced to the vapor pressure of the fluid. This results in noise, vibration and cavitation damage to metal surfaces in what was previously thought to be strictly a high pressure area and thereby, immune to cavitation.

Figure 4: Cavitation Erosion

Now that we are somewhat familiar with the types of erosion that can occur within a pump, let's discuss what can be done to alleviate, or at least retard, this type of destruction. Because of the considerable degree of interaction between the three defined areas of erosion, many of the preventative measures described below apply overall.

Employ Hi-Efficiency Pumps

Inefficient units, with their poor flow patterns, are conducive to considerable wear problems regardless of the materials employed. Less costly open impellers are susceptible to cavitation erosion in the clearance between the rotating impeller blades and the stationary pump cover or casing. A typical example of this is shown in Figure 5 on the following page. This type of erosion is caused by a cavitating vortex (low pressure) developed in the clearance. Note that the closed impeller designed by Ampco stood up exceedingly well by comparison since its pressure gradient line is at the smaller diameter wear ring surface where the interface velocity is much lower. In this instance, the life of the closed impeller was 12 times that of the open design even though the open impeller was made of a superior wear-resistant material.

Use Sound Pump Application Principles

This includes running the unit near its BEP (best efficiency point). Limitations relative to NPSH availability must also be met. While parameters in these areas are quite definitive, those in the area of recirculation will remain vague until more empirical data is developed to provide sound practical limits. Enough work has already been completed in this field to enable pump designers to establish some approximate parameters for specific applications normally developed by actual service.

Always keep in mind that the user of a poorly designed or applied pump pays two-fold for his error - needlessly paying for massive amounts of wasted energy which, as it is dissipated into destructive erosion within the pump and system, effectively reduces the life expectancy of the unit.

Evaluate Materials of Construction

This must be accomplished in several aspects relative to the intended service. Corrosion, cavitation and wear must be included. Previous experience in similar applications can be the user's greatest aid in selecting an adequate material. While hardness is not the only criteria of resistance to abrasive wear, it does provide a guide in the selection of ductile materials ranging from cast iron through the bronzes to the 300 and 400 series stainless steels. No definitive method of wear rate related to abrasive particles has yet been established.

As it may not be practical to eliminate all cavitation conditions, it is reasonable to conclude that materials with a higher degree of resistiveness to cavitation be considered. While there isn't total unanimity as to which materials are superior, both nickel-aluminum bronze and the 300 stainless steels consistently rate higher marks.

Ceramic coatings applied to known areas of high erosion have been successful in some applications. And, on larger units, welded layers of stainless steel or aluminum bronze not only restore worn surfaces but often provide longer service life than the original base metal.

Success has been attained in some abrasive services with lined (rubber) pumps. The resilient properties of rubber and other elastomers have been found to offer a degree of protection. However, developed heads are limited by the relatively low velocities allowed. The narrow hydraulic range coupled with temperature restrictions and somewhat crude hydraulic design hamper overall acceptability of these units.

Employing Latest Manufacturing Methods

These include the investment and evaporative foam casting processes. Increases of up to 5% in pump efficiencies can be realized by substituting one of these processes for the traditional sand cast method. Benefits include smoother surface finishes with minimal flaws, resulting in fewer potential nucleation sites. This reduces the risk of bubbles which might otherwise form, grow and separate from the surface in low-pressure areas on the verge of cavitating.

Improved manufacturing methods have also, in many instances, reduced the pump's NPSH requirements because of improvements in the leading edges of the impeller blades and the smooth direction transition from horizontal to vertical flow in the inlet passage profile of the pump.

Preventative Maintenance Procedures

A periodic inspection system, much of which can be assigned to the operator, should be included in this program. Reference may be made to article 9 in the Pump Primer series covering maintenance.

Monitoring the operation of a system - frequently at first and periodically thereafter - is necessary to accurately gauge wear pattern development.

Reports summarizing operating hours supported by photographs are helpful in documenting problems as they develop. Awareness of personnel to changing noises, vibrations, shocks, etc. should always be specifically noted.

Summary

As in so many areas, final appraisal of a pump's performance is established by its "engineered application". Severe erosion can be reduced if a pump with a good hydraulic design is properly applied. However, at best this remains a "state of the art" determination. Where problems are incurred, options always exist to modify, correct or replace the current unit with another more applicable approach. If you can't keep the system simple, try to keep the liquid velocity as uniform as possible.

Erosion or wear can be reduced by selecting pumps for low fluid velocities (particularly in close-proximity regions such as running clearances, etc.) and by specifying castings of fine-grained materials.

Most important of all, in erosive service, use a capable, efficient pump as effectively as possible. Anything less will result in high costs for wasted energy and accelerated destruction of equipment through erosion (caused by recirculation, cavitation and eddies) and heat.

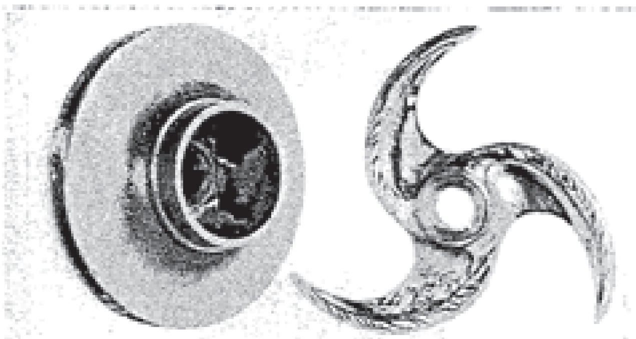

Figure 5
Compare the open impeller after 3 months, and the closed design after 3 full years, of pumping the same slurry at identical speed, capacity and head. Open impeller, badly worn on vane faces, had lost so much capacity it had to be replaced.

Pump Primer

Basic Information About Centrifugal Pumps

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Number 17 in a series

by John H. Horwath
Chief Engineer, Ampco Pumps

Pump and System Curves II

It may be advisable to review the material on basic system curves in Pump Primer No. 5 before taking up the discussion of more complicated systems in this article.

Introduction

When sizing a pump for a multiple-flow system, the flow requirements for the pump will be the sum of the simultaneous flows through each discharge outlet of the system. A piping system common to each of the lines will have its own frictional pressure drop that contributes to the pressure drop in each branch line.

In selecting a pump, the initial design approach is to establish the required head at a design flow consistent with the operating conditions of the "normal case". "Normal" is interpreted to be the point at which the process is expected to operate most of the time.

Where rated flows may be required to allow some degree of capacity variation under certain circumstances, a check of the developed system characteristics will indicate the head necessary to reach the required rated flow and the capability of the pump to provide the additional head required. The system must be reviewed to determine if there is adequate pressure drop available in the sensitive range of the control valve to effectively provide the additional frictional pressure drop created at the rated flow.

Reduced flow requirements should also be analyzed as to their effect on the valves and pump. The valves must be able to maintain effective control of the process without chattering (high pressure drop). Analysis of the pump should determine the extent of reduced operational effects such as radial loading, flow recirculation and heat buildup.

The following discussion presents some examples of more complex systems.

Example 1

Single pump operation of two lines in parallel with different elevation levels.

Begin by developing the individual head-capacity system curve for each line from point A to B and A to C. Then graph each segment relative to their static discharge head differential (ΔH). Next, determine

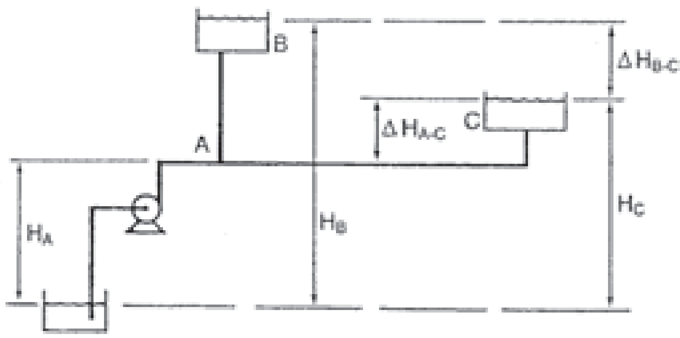

figure 1

curve R which is the resistance of the common line from entry into the suction line to discharge point A where the flow to tanks B and C separates. Add in parallel the lines to tanks B and C relative to their levels of operation. Next, add in series the common line (friction) losses that allow system curves (H_{YB} and H_{YC}) to be developed in parallel plus H_A in series. The intersection of this system curve (figure 2) with the superimposed pump characteristic will indicate the projected point of operation.

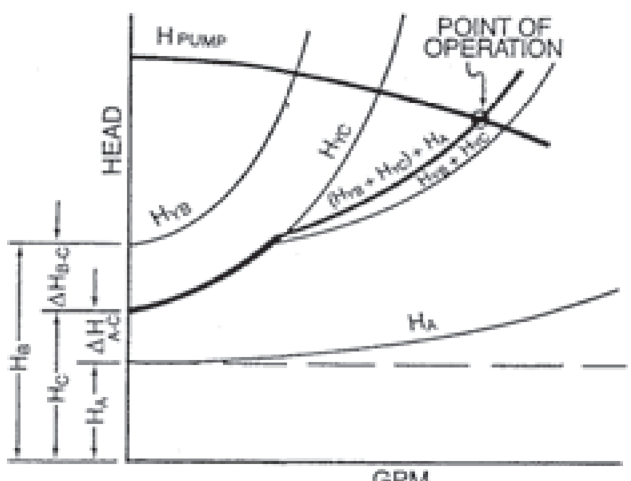

figure 2

This type of graph also indicates what can be expected to occur if specific conditions change.

Example 2

Two pipe lines in series with a side outlet (or part of flow diverted elsewhere).

In this example (figure 3) again begin by isolating the individual segments and then develop a graphical overview of the system.

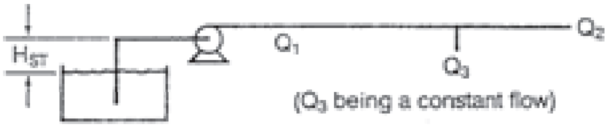

figure 3

Frictional losses designated by the total flow (Q_1), the constant flow (Q_3) and the secondary line (Q_2) must first be established and plotted individually, taking into account static lifts as well. Where series piping is utilized, the line losses in the system are additive (as shown in figure 4) once the total flow (Q_1) exceeds (Q_3).

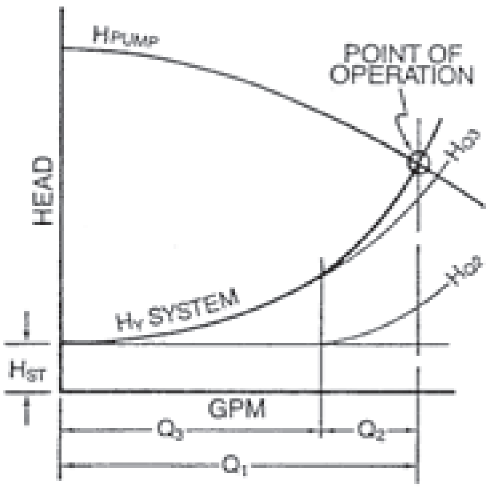

figure 4

Example 3

If two pumps are taking water from different levels, the two head-capacity curves may be developed (figure 5) from the two respective base lines in the usual manner of parallel pump flow where, at the same absolute head, the flow capacities of the two pumps are additive in developing a combined pump performance curve.

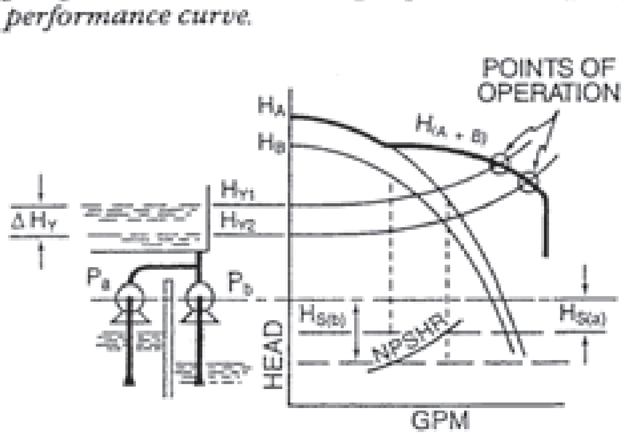

figure 5

This curve could be altered significantly if one or both pumps — due to a reduced system static-head requirement — resulting in greater flow, thereby increasing the NPSH required by the pumping units to the point where cavitation could occur in one or both of the pumps.

Example 4

Closed system-head curve for pump and branch lines (figure 6).

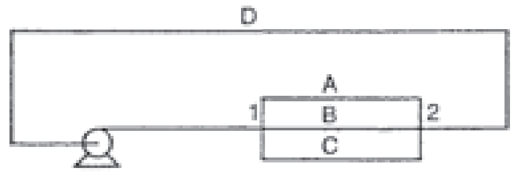

figure 6

In developing the following procedure (figure 7), based on each paralleled branch line having a different system characteristic due to the makeup of individual process equipment, fittings, piping and open valving in each segment, start by developing the three individual characteristics plus the common line (D) which encompasses the entire system, except for the

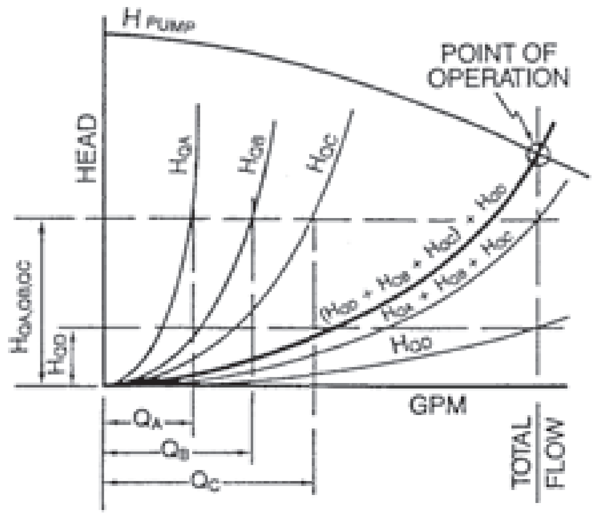

figure 7

three parallel segments (A), (B) and (C). Next, calculate individually the frictional losses of each of the three parallel segments (with fully open valving) and plot accordingly. Finally, establish a curve consisting of the total flow with each parallel segment being wide open.

Example 5

Centrifugal pumping system with a continuous recirculation line. The primary purpose of the recirculation process loop is to prevent overheating the pump by maintaining a minimum safe flow. A typical continuous recirculation line is shown in figure 8.

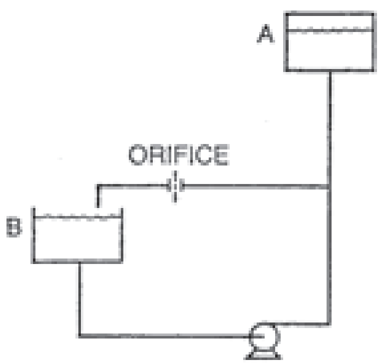

figure 8

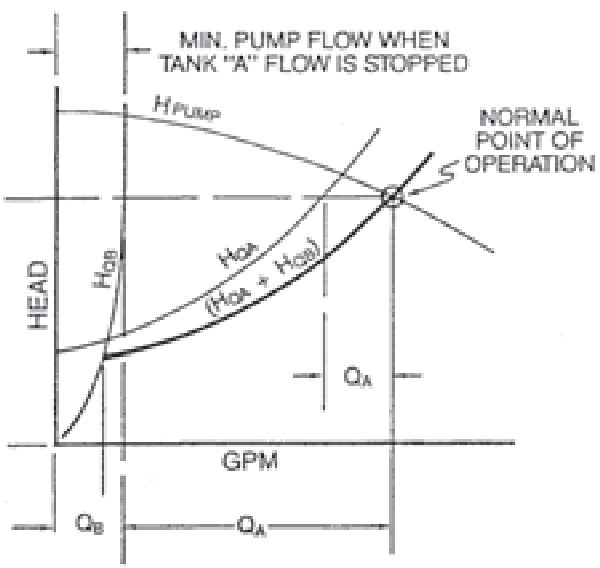

figure 9

A graphical analysis (figure 9) can be made to illustrate what develops when a by-pass orifice is introduced.

General

Just as a pump characteristic can provide much meaningful information relative to its capability, operation and design, so, to a good degree, will a partial system curve, developed from a minimal amount of actual data, provide insight relative to a system's performance.

The most common question asked of a pump application engineer is: "How much will the flow rate through my system increase if I go to a larger impeller diameter and/or pump?"

To check this in an existing system, taking actual data utilizing properly calibrated test equipment (pressure gauges and flow meter) positioned in accordance with Hydraulic Institute Standards, a fairly accurate forecast for a stable condition can be expected within approximately 15% of the measured flow range. The exception to this statement occurs when the system curve begins to sharply turn vertically upward, in which case, regardless of the increase in available pump head, hardly any additional flow will result.

Another factor is that the system's NPSH available at the desired flow must be met by or exceed that required by the pump. Upon meeting these requirements, the velocity head, entrance-exit and friction losses of a uniform solution can be assumed to be proportional to the flow $(GPM_1)^2$. Knowing this, the Total Dynamic Head for a greater or lesser flow (GPM_2) can be computed:

$$TDH_2 = \left[\frac{(GPM_2)^2 (TDH_1 - \text{Static Head})}{(GPM_1)^2}\right] + \text{Static Head}$$

Using this method to determine TDH_2, a partial system curve (figure 10) can be calculated that will allow us to superimpose other impeller diameters for the existing pump or even another pump size to determine an approximate point of operation. This would then allow us to select a size, speed and impeller diameter along with the motor horsepower capable of handling the new flow requirement.

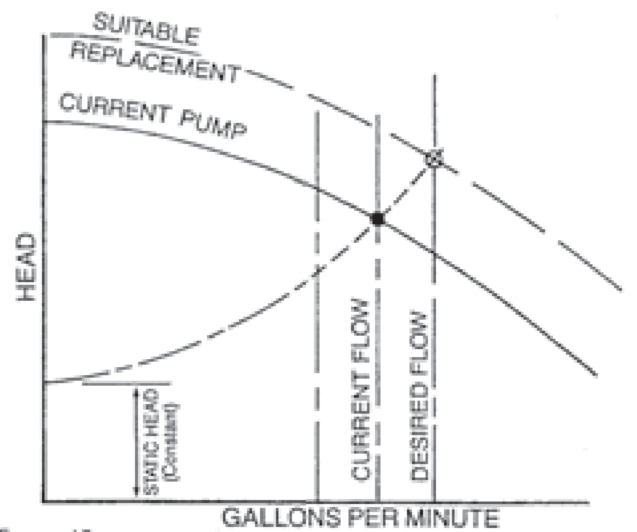

figure 10

In addition, remember that your system's NPSH available at the increased flow will decrease and the required pump NPSH will increase (refer to article 3 of this series relative to the NPSH factor).

Note also how your system-pump characteristics can control the degree of variation in your flow and pressure.

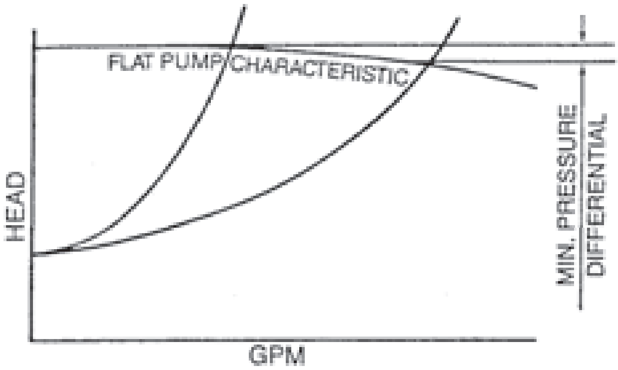

figure 11

The above curve shows how the use of a flat pump characteristic will maintain a relatively stable discharge pressure over an extended flow range.

By the same token, if you require a relatively stable flow with a flat system curve, select a pump with a sharp characteristic drop (figure 12).

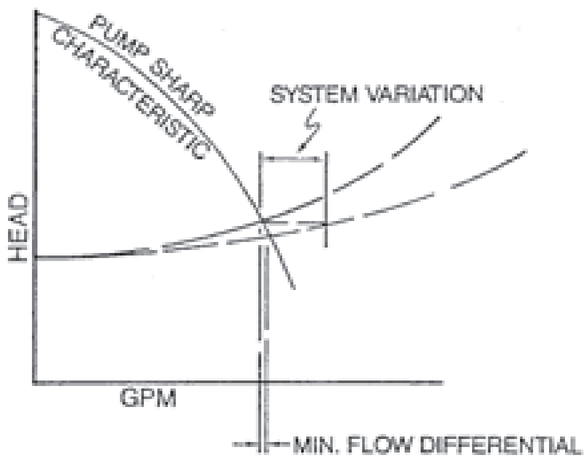

figure 12

In this instance, a change in the system characteristic would have minimal effect on flow.

Summary

A graphical analysis provides a much more composite picture of what is actually occuring hydraulically within a system and what ramifications may result as conditions change. Whether developed by a computer program or drawn out manually from calculator results, this analysis will aid immeasurably in solving specific problems. Note particularly that potential undesirable trends can be forecast and dealt with beforehand. Developed changes, dependent on their origin, follow progressively varying exponential factors over an extended range.

The graphical method will not only quickly provide adequate accuracy for most pump sizing determinations but will also provide an overview allowing for an analytical determination of the driver's requirements. The standard pump test tolerances (per Hydraulic Institute Standards) are:

at rated head,
+ 10% of rated capacity, or
at rated capacity
+5% of rated head under 500 feet
+3% of rated head 500 feet and greater

Coupled with the continuing changes occuring in an existing system and/or the varying data taken from reference sources along with the so-called safety factors employed by the project engineer, one can see why the graphical method is preferred. Taking general empirical data and calculating to three decimal places doesn't make the answer more accurate than the graph, it only provides a false sense of security. Developing a series of "what if" curves will allow one to head off potential problem areas as well as provide background for making intelligent decisions as changes occur in the system and pump performance.

Pump Primer

Number 18 in a series

Basic Information About Centrifugal Pumps

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Effects of Developed Hydraulic Loads in Centrifugal Pumps

Second only to shaft sealing in centrifugal pump problem areas are the pump's bearings and shafting. Poor maintenance procedures, inadequate protection of the bearings in hostile environments, misapplication and poor pump design are among the underlying causes of bearing and shaft problems.

This article focuses on the latter two causes. Maintenance procedures covering the other areas of concern were reviewed in article 9.

Prior to the 1950's, little was known by the typical user of volute-type centrifugal pumps relative to the shaft and bearing loads developed hydraulically within the pump housing. Knowledgeable pump designers generally were aware of the potential consequences and designed accordingly. For those who weren't, A.J. Stepanoff, America's foremost pump expert of that era, covered this subject in his book, "Centrifugal and Axial-Flow Pumps," first published in 1948 from a design and application viewpoint along with a number of technical papers released in this same period. In the mid-fifties, Donald S. Ullock of Carbide and Carbon Chemical Company presented a paper titled "Evaluating the Mechanical Design of End Suction Centrifugal Pumps" published in May, 1955 in Chemical Engineering Progress (vol. 51, no. 5, pages 207-222). Later, an appendix he wrote expanding on this subject was also published. Ullock's papers were written from the standpoint of those who specify, purchase, operate and maintain centrifugal pumps.

Prior to this time, there was a widespread tendency to design pump shafts primarily on the basis of horsepower transmitted although several pump designers did incorporate critical speed requirements as well. A substantial number of designers, however, overlooked or were unaware of the potential hydraulic resultant forces that could develop within the pump itself.

A rash of pump failures at that time helped arouse considerable skepticism as to the worthiness of 3500 RPM operation, which was beginning to gain in popularity after half a century of domination by the reliable 1750 RPM drives. The call for higher processing pressures and the advent of the economical 3500 RPM, squirrel-cage A.C. motor precipitated this change.

The increase in motor speed allowed pump heads to be developed that were four times greater than the same impeller diameter was capable of at 1750 RPM. Suffice it to say that this potential load factor increase was not always addressed directly by all pump manufacturers. Concerns over an increase in shaft and bearing failures led some large users to conduct their own detailed technical evaluations of a pump's capability to handle hydraulic loads which could develop within the unit (1) at the intended point of operation, (2) at shut-off and (3) at other capacities.

As a result of the developing problem areas, some pump manufacturers initially over-reacted by overdesigning, thereby having to utilize oversized bearings which, because of speed limitations, resulted in several premature failures. Mechanical seal life, based on a pressure-velocity factor, was also affected.

A well-designed pump is one which has the proper size of shaft for the radial and axial loads exerted by the impeller at its rotating speed. This can be calculated and checked by measuring the loads involved and then designing shaft and bearings accordingly for a specific flow range.

Today's textbooks, standards and specifications often make reference to shaft and bearing requirements, particularly related to shaft deflection and stress along with bearing life expectancy.

Radial Loads

While there are several methods of converting velocity energy to pressure energy, the most popular method, primarily because of its economical design, is the single volute casing. Here, a good engineered product will result in uniform pressure within the volute casing at the design (best efficiency) point capacity. For pumps in applications normally operating at or near the best efficiency point capacity, the radial thrust factor may approach zero as indicated in figure 1 for a typical volute-style centrifugal pump.

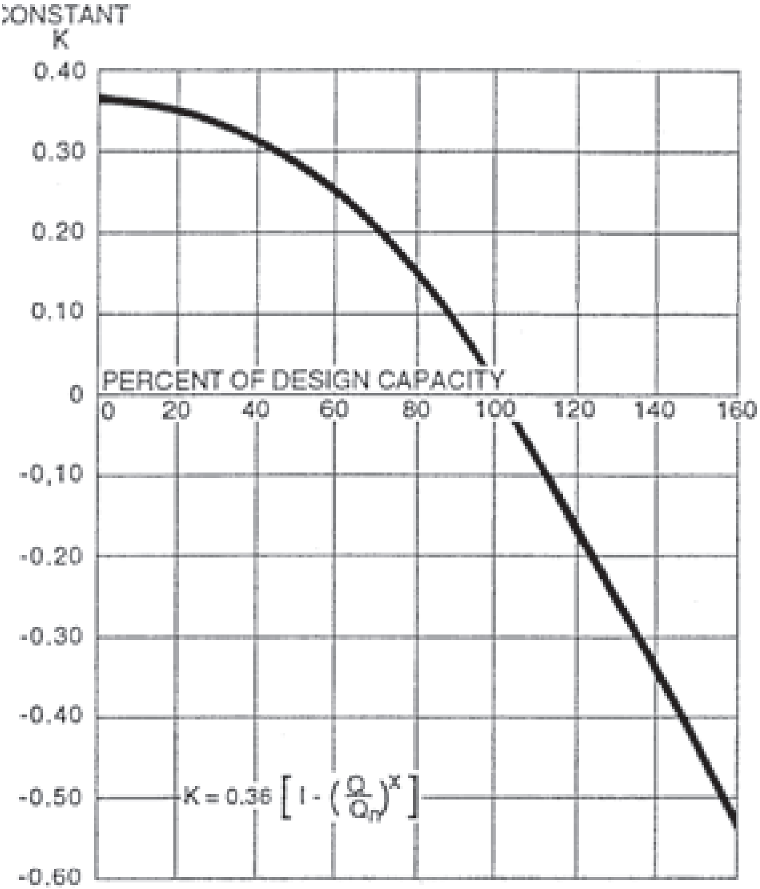

figure 1

The graph indicates that, on both sides of the best efficiency point this equilibrium of volute pressure - a highly significant pressure differential develops as the point of operation moves in either direction away from the pump's design point.

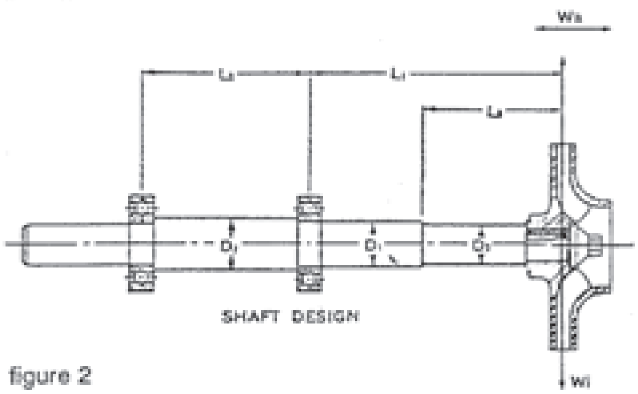

figure 2

A sketch of a typical overhung, single-stage impeller shaft and bearing assembly arrangement representing the rotating part of a centrifugal pump is shown in figure 2. The resultant radial force (W_r), when flow is less than design capacity, is along a radial line toward the center in a radial direction 270° from the cut-water. At design flow, the radial pressure is equal all around the impeller periphery. As flow increases beyond the BEP, the resultant of these forces is directed from the smaller volute sections toward the larger, lower-pressure sections. The weight of the impeller is indicated by W_i. The resultant axial thrust force (W_a) for the impeller and its direction is dependent on several specific hydraulic and design factors. Both forces, even in relatively small pump units (25 HP, 3500 RPM, 7-inch impeller), can develop substantial forces of several hundred pounds in each plane.

The immediate effect of a high radial resultant force, developed as a result of uneven pressure distribution in the volute casing acting against the impeller's radial profile, is a deflection of the overhung shaft at the impeller position. Since typical radial clearance between the impeller and wear ring of small pump units ranges from .005" to .015", galling or rapid wear of the components could easily occur as well as shaft breakage due to fatigue failure in the rotating shaft.

For development of the "K" factor relative to a specific pump, refer to the Hydraulic Institute Standards (fourteenth edition, pages 115 and 117) which also covers the method used to determine the radial thrust load based on the pump's design and performance characteristics.

The radial load in a volute pump is developed from the formula:

$$W_r = \frac{KH\,D_2\,B_2\,S}{2.31}$$

where the constant K depends on the pump's capacity, speed, casing, design, specific speed and other factors. The shut-off value of K, referred to as K_{so}, is given in the Hydraulic Institute Standards for typical pumps. The value of K at the intended point of operation can be developed using the following:

$$K_{op} = K_{so}\left[1 - \left(\frac{Q}{Q_n}\right)^x\right]$$

where Q = capacity @ operating conditions - GPM
Q_n = capacity @ best efficiency point - GPM
X = exponent, varying between 0.7 and 3.3, established by test. In the absence of test data, the component may generally be assumed to vary linearly between 0.7 @ specific speed 500 and 3.3 at specific speed 3500. (Specific speed determination of a centrifugal pump was provided in article no. 2).

Knowing K, we can proceed to determine the radial thrust in pounds at the point of operation by:

$$R_o = \frac{K_{op} \times H_{op} \times S \times D_2 \times B_2}{2.31}$$

where
H_{op} = head at the intended point of operation in feet
D_2 = impeller outside diameter in inches
B_2 = impeller width in its O.D. including shrouds
S = specific gravity of the liquid being pumped.

The data provided here relative to establishing the K factors was developed from empirical data of numerous tests on general-purpose centrifugal pumps. Individual pump manufacturers may have data on their specific designs which demonstrate other values.

Axial Loads

Axial loads are developed as the result of the varying pressures existing within the pump along with their effect on the suction and back-side areas of the pump impeller's exposed axial profile and shaft. The resultant axial hydraulic thrust is a summation of unbalanced forces on an impeller acting in the axial direction.

Several factors affect the magnitude of the axial thrust developed in a single-stage, overhung impeller pump. Proper sizing of the thrust bearing is dependent on factual knowledge of the thrust-bearing loads being developed. Relative to the impeller itself, against which the unequal internal pressures will act in developing a resultant axial thrust force, this includes: (1) pump design (open, semi-open or enclosed impeller), (2) the impeller's major diametral areas in the axial plane related to developed pressures, (3) back-side shroud ribs, (4) balance holes (size and number), (5) wear ring radial clearance and (6) shaft sleeve diameter. Based on the pump's working pressures from suction entry to volute discharge, the resultant magnitude and direction may be determined. Since a general-purpose pump may be exposed to a number of service conditions over a wide range of suction conditions, the thrust bearing of a single-stage unit must be designed to take thrust in either direction. For calculating these loads, refer to the detailed methods used by D.S. Ullock in his appendix "Evaluating the Mechanical Design of End Suction Centrifugal Pumps," to various pump textbooks or to simpler methods related to shut-off pressure, area of the impeller at the wear ring, overall area at the impeller O.D. and area at the shaft sleeve diameter.

Shaft and Bearing Loads

Once the axial and radial loads acting on the pump impeller have been calculated or established experimentally, the individual bearing loads and (in frame units) the bearing life and shaft deflection plus developed stresses in the shaft can be determined.

Typically, in a two-bearing arrangement, one support is "fixed" and subjected to the thrust load plus the statically determined part of radial load while the other is a "floating" bearing load consisting entirely of it's static radial load reaction occurring at the floating bearing location.

Close-coupled pump loads can be passed on to the motor manufacturer along with specific shaft location and direction of rotation. At one time, a major manufacturer of close-coupled pump motors provided a data sheet indicating allowable limits based on speed (RPM) related to radial and axial loads. Among the determinations that could be made was shaft deflection related to a reference point on the shaft end on which the impeller was mounted, based on a deflection factor provided as "inch per pound of radial load." The data sheet also provided a graphical parameter indicating allowable combined loads (thrust and a factored radial load) related to motor frame size and RPM. Axial thrust values were also provided for shafts in tension and for maximum permissable reverse thrusts where shafts would be in compression.

In proceeding through the exercise discussed here, it becomes apparent that a significant increase in radial loading develops as operation moves away from the BEP (best efficiency point). Most troublesome is the operation near shut-off where a steep performance curve results in a significant increase in Head developed which, in turn, is directly related to the radial thrust load (W_r).

Shaft Deflection

Shaft deflection is primarily (1) a direct function of the radial load and the cube of the overhung shaft length and (2) an inverse function of the moment of shaft cross-section inertia (at the overhung section) and the modulus of elasticity of the shaft material.

A typical overhung pump/motor shaft with a series of cross-sectional diameters of varying lengths and two bearing points requires a complex series of calculations by the pump designer. The intent here is to create awareness of the primary factors involved in this procedure.

In addition to remaining within the radial clearance of the wearing ring in order to maintain a high volumetric efficiency, shaft stiffness is also required in the mechanical sealing area (commonly .002" maximum deflection) to provide proper contact between the lapped faces of the rotating and stationary seals at all times.

Shaft Failure

Typically, shaft stress is greatest in the overhung cross-section immediately beyond the shaft's inboard bearing where the shaft diameter decreases. It is at this point where the high localized stress referred to as stress concentration commonly occurs. To minimize these stress raisers, it is necessary to provide smooth fillets of maximum allowable radii where abrupt changes in cross-sectional areas exist. Failure to reduce stress concentration in these areas may result in fatigue failure caused by stress raisers subjected to cyclic stress in the rotating shaft.

Bearing Life

Bearing life is defined as the number of hours at a given speed that 90% of a group of bearings will attain or exceed before the onset of fatigue. This basic definition is commonly referred to as bearing L_{10} life.

Bearing life is a function of two basic sets of conditions:

1. Application
   - a.) load
   - b.) speed
   - c.) temperature
   - d.) mounting
   - e.) lubrication

2. Bearing Characteristics
   a.) bearing design
   b.) material
   c.) manufacturing methods

Speed Limit

As bearing size increases, the approximate speed limit of the ball bearing decreases . Data is based on oil lubrication. With grease lubrication, the limit on speed is usually reduced by one-third.

In general, life expectancy has been raised as special processes, refined manufacturing developments and improved lubricants are employed in pump bearings.

For volute-type centrifugal pumps, where combined radial and axial load is greatest at or near closed discharge, the minimum bearing life should not be less than 20,000 hours (2.3 years of continuous operation). Commonly, today's users are specing in a minimum of 50,000 hours. Calling for an even higher life expectancy is not practical as environment and age then become the dominant life factors.

Summary

Obviously, the interest of the pump user is in trouble-free operation with a minimum of breakdowns. This has resulted in significant mechanical improvements over the years. The most important consideration in providing maximum trouble-free pumping service is to maintain continuing (normal flow) operation in the area of the pump's best efficiency point.

There are ways to significantly reduce both radial and axial load, primarily by redesigning. However, in most instances, these "fixes" are expensive and limited.

Basically , maintaining normal flow near the pump's BEP by PROPER APPLICATION provides the best solution.

Where options are limited, design modifications may be considered for either dealing with high loads by utilizing heavier shafting and larger bearings or by redesigning hydraulic components to decrease resultant forces by effectively reducing impeller profile and differential pressure sectors.

Pump Primer

Basic Information About Centrifugal Pumps

Ampco Pumps Co., Inc.
(414) 643-1852 Telephone
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Number 19 in a series

by John H. Horwath
Chief Engineer, Ampco Pumps

Standard Motor Controls

This article was prepared with the intention of providing engineers, maintenance personnel and others working with centrifugal pumps useful and informative reference material about AC squirrel-cage motor starters and controls. The pump motor and motor controller are interrelated and often are considered as a package when choosing specifics for a particular application. Simple motor controllers, called motor starters, start and stop the motor merely by applying and removing electrical power. More complex controllers may also condition the electrical power to provide required output characteristics. Motor protectors prevent motor operation under conditions which might cause overheating and motor winding damage. The functions are available separately or as various combinations in individual units.

Functions

The five major functions of a motor control system are intended to: (1) stop and start electric motors, (2) protect personnel, motors and control equipment, (3) govern motor speed, torque, horsepower and other characteristics, (4) maintain proper sequencing of motors, equipment, processes and operations, (5) sense and correct errors in operation of a motor, pump or process.

Code

Laws, codes, regulations and standards can be an important factor in the selection process of a controller for a specific application. OSHA has, per its section 1910.309, adopted as a national concensus standard, the National Electrical Code NFPA 70-1971. The purpose of the NEC is the practical safeguarding of any persons, buildings and contents from hazards arising from the use of electricity for light, heat, power and other purposes. This means that equipment must be labeled or otherwise determined to be safe by a nationally recognized testing laboratory. Care must also be taken to insure that all local codes are being met as well. NEC rules and provisions are both enforced by governmental bodies exercising legal jurisdiction over electrical installations and used by insurance inspectors.

Motor Protection

In addition to direct protection of the motor itself, the NEC calls for the protection of the branch circuit and feeder line. The intent of the code is to protect the motor, the motor control apparatus and the branch circuit conductors against excessive heating due to motor overloads or failure to start. Other protection provided by fuses and circuit breakers is expected to guard against fault conditions caused by short circuits or grounds and involving over-currents exceeding locked-rotor values of the motor.

Overload

The effect of an overload is a rise in temperature in the motor windings. The greater the overload, the more quickly the temperature will increase to the point of damaging motor insulation and lubrication. Relatively small overloads of short duration cause little damage but, if sustained, could be just as harmful as overloads of greater magnitude. The operation of the protective device should be such that the motor is allowed to carry harmless overloads but is quickly removed from the line when an overload of any length persists. Motor burnouts are said to be the highest single cause of electrical fires.

Overcurrent Protection

The function of the overcurrent protection device is to guard the motor branch circuit conductors, control apparatus and the motor itself from short circuits or grounds. The protective devices commonly used to sense and clear overcurrent are thermal magnetic circuit breakers and fuses.

Circuit Breakers

A circuit breaker is defined in NEMA Standards as a device designed to open and close a circuit by non-automatic means, and to open the circuit automatically on a predetermined overcurrent without damage to itself when properly applied within its rating.

A molded case circuit breaker is further defined by NEMA as one which is assembled as an integral unit in a supporting and enclosing housing of insulating materials.

Fuses

The basic function of a fuse is to protect against short circuits (overcurrents). Motors draw a high inrush current when starting. Single-element fuses have no way of distinguishing between this temporary inrush current and a damaging overload. A fuse, therefore, is not duly suited for motor overload protection.

Overload Relays
The overload relay provides ideal motor protection. It has inverse trip-time characteristics, permitting it to hold in during accelerating periods, yet providing protection against small overloads above full load current when the motor is running. The overload relay differs from a fuse in that it is renewable and can withstand repeated trip and reset cycles without having to be replaced. However, the overload relay does **not** provide short circuit protection which is provided by fuses and circuit breakers.

Overload relays are available as either magnetic or thermal. Magnetic relays respond only to current overload and are not affected by temperature alone. Thermal overload relays rely on the rising temperatures developed by the overload current to trip the overload mechanism.

Method of Selection
The first step in the selection of the starter process is to list the specific characteristics of the motor including: horsepower, phase, frequency and voltage. Once this is established, the required NEMA starter size can be determined from the accompanying tables. (The general power frequency in the U.S. is 60 Hertz. The use of these tables is restricted to full voltage starters using 60 Hertz alternating current up to 575 volts.) The maximum motor horsepower limitation for each NEMA size starter is shown, depending on specific voltage characteristics.

NEMA SIZE STARTERS
Maximum Allowable Horsepower
SINGLE PHASE

NEMA SIZE	115 VOLTS	230 VOLTS
00	1/3	1
0	1	2
1	2	3
1 1/2	3	5
2	-	7 1/2
3	-	15

table 1

NEMA SIZE STARTERS
Maximum Allowable Horsepower
THREE PHASE

NEMA SIZE	200 VOLTS	230 VOLTS	460/575 VOLTS
00	1 1/2	1 1/2	2
0	3	3	5
1	7 1/2	7 1/2	10
2	10	15	25
3	25	30	50
4	40	50	100
5	75	100	200
6	150	200	400

table 2

Since operation of centrifugal pumps is usually limited to rotation in one direction, the controls vendor should be advised that reversing service will not be required.

The next step is to select a starter type. The across-the-line type of starter is by far, the most common. This type places the motor directly across the full voltage of the supply lines, providing rapid acceleration, which is suitable for the typical small centrifugal pump.

At this point, one basic function of operation remains to be resolved - the option of using a manual or magnetic starter. A manual starter mechanically establishes the circuit to the motor while a magnetic starter employs electro-magnetic means to establish the circuit.

Manual Starter
Basically, a manual starter is an "on-off" switch with overload relays. It provides low-cost control and uncomplicated performance. It does not provide for remote operation or under-voltage protection. Manual starters are limited to single-phase motors up to 5 HP @ 230 volts and to three-phase motors up to 15 HP @ 600 volts. A manual starter is, simply, a hand-operated mechanism that makes and breaks the motor circuit. A thermal protective device included in the starter guards the motor against excessive currents.

Magnetic Starter
A magnetic starter is a device which operates electromagnetically to start, stop, control and protect the motor. Actuation of an electric circuit to an operating coil magnetically closed the power contact of the device. Magnetic starters permit remote operation and automatic control of pilot devices such as pressure switches, float switches, timers and similar devices.

Although they cost more than manual starters, magnetic units are more widely used. Magnetic starters can provide important operation features and are capable of withstanding frequent and hard use.

Like the manual starter, a magnetic starter contains a mechanism for opening and closing a set of contacts in the motor circuit and a thermal overload. Unlike a manual starter, the contacts in a magnetic starter are moved by an electro-magnet in the starter. When the magnet is energized, movable contacts close against stationary contacts, completing the electrical circuit. De-energizing the magnet opens the circuit.

Solid State
Solid-state motor starters use semi-conductor power and trigger circuits instead of contacts. The motor is accelerated by varying motor voltage from zero to full voltage on a stepless ramp. Acceleration time is adjustable up to about 10 seconds by means of a trimmer-potentiometer in the trigger circuit. The starter is activated by closing a remotely located switch wired to the starter terminals. In addition to thermal overload protection, solid-state starters provide protection against phase loss and phase reversal.

Recently, two U.S. manufacturers introduced motor starters with solid-state, current-sensing devices. It has been estimated that up to 25% of all motors requiring rewiring have been damaged because of single-phasing. Contactors controlled by micro-processors in some solid-state starters provide a communications "tie-in" to a control room, providing information relative to the status of the motor.

Combination Starter
This type combines a magnetic starter with a disconnecting device in one enclosure. The disconnect may be a motor-circuit switch or a circuit breaker.

Reduced-Voltage Starter ("Soft Start")
In some pump applications, particularly involving larger horsepower requirements, a "soft start" may be needed. In addition to the demands of some applications, power regulations may limit the current surge or voltage fluctuation that can be imposed on the power supply during motor starting.

Enclosures
The final step in the selection process is the determination of the proper enclosure. Controller enclosures provide protection for operating personnel by preventing accidental contact with "live" elements. In certain services, the controller itself must be protected from a variety of environmental conditions such as explosive gases or combustible dusts. Table 3 provides a listing of NEMA Standard enclosures.

NEMA Standard Enclosures

Type 1	General purpose
Type 2	Drip tight
Type 3	Dust tight, rain tight and sleet resistant
Type 3R	Rain proof and sleet resistant
Type 3S	Dust tight, rain tight and sleet proof
Type 4	Water tight and dust tight
Type 4X	Water tight, dust tight and corrosion resistant
Type 5	Dust tight
Type 6	Submersible
Type 7	(A, B, C or D) Hazardous locations Class 1, air break
Type 8	(A, B, C or D) Hazardous locations Class 1, oil immersed
Type 9	(E, F and G) Hazardous locations Class 2
Type 12	Industrial applications, dust tight and drip tight
Type 13	Oil tight and dust tight

Diagrams and Symbols
The basic language of controllers is the circuit diagram. Consisting of a series of symbols inter-connected by lines to indicate current flow to various components, the circuit diagram provides information relative to (1) the flow of current in the device being controlled and (2) the current flow to the device being controlled.

Wiring and Line Diagrams
Wiring (connection) diagrams include all the devices in a system and show their physical relation to each other. A typical wiring diagram of a three-phase magnetic starter with a start-stop push button station is illustrated in figure 1.

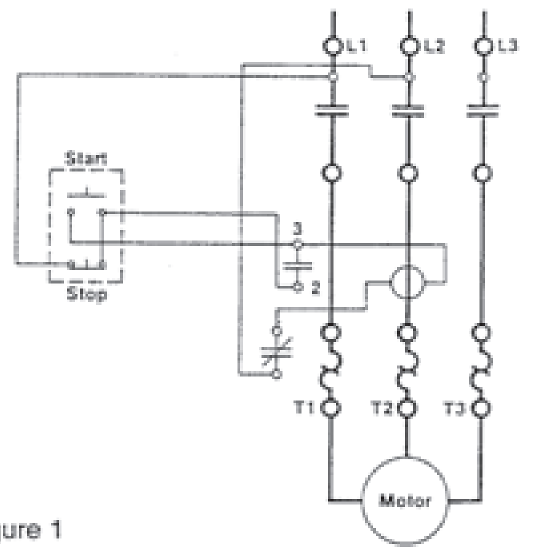

figure 1

All poles, terminals, coils, etc. are shown in their proper position in each device. These diagrams are useful in wiring systems although, in following the electrical sequence of a circuit, connections are not shown in an easy-to-follow manner. For this reason, some rearrangement of the circuit elements to form a line diagram is desirable.

The line (elementary) diagram is a representation of the system in its simplest form. Figure 2 is the line diagram for the wiring diagram in figure 1. No effort is made to place the various components in their actual position.

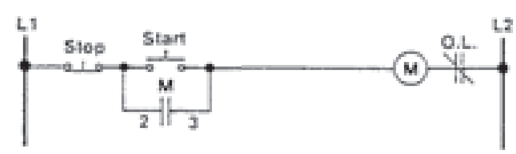

figure 2

Typically, all control devices are shown between vertical lines which represent the source of control power. Circuits are shown connected as directly as possible in an easily followed manner.

In short, a writing diagram provides the necessary information for physically wiring up a group of control devices or for tracing circuits in trouble-shooting situations. A line diagram provides information required to follow the operation of various devices in the circuit.

The most commonly used symbols are shown in figure 3. Familiarity with them will provide a better understanding of line and wiring diagrams.

Summary

Motor control manufacturers and their representatives are a most helpful source of information in developing standard motor control line and wiring diagrams for specific requirements. Their catalogs typically include numerous wiring arrangements, one of which will usually meet or at the very least will conform, with modifications, to most specific requirements. Motor control manufacturers are also a source of considerable literature which, with published handbooks and other reference material, provide an excellent source for acquiring basic information.

More complex requirements may demand the services of professionals in this field.

WIRING DIAGRAM SYMBOLS

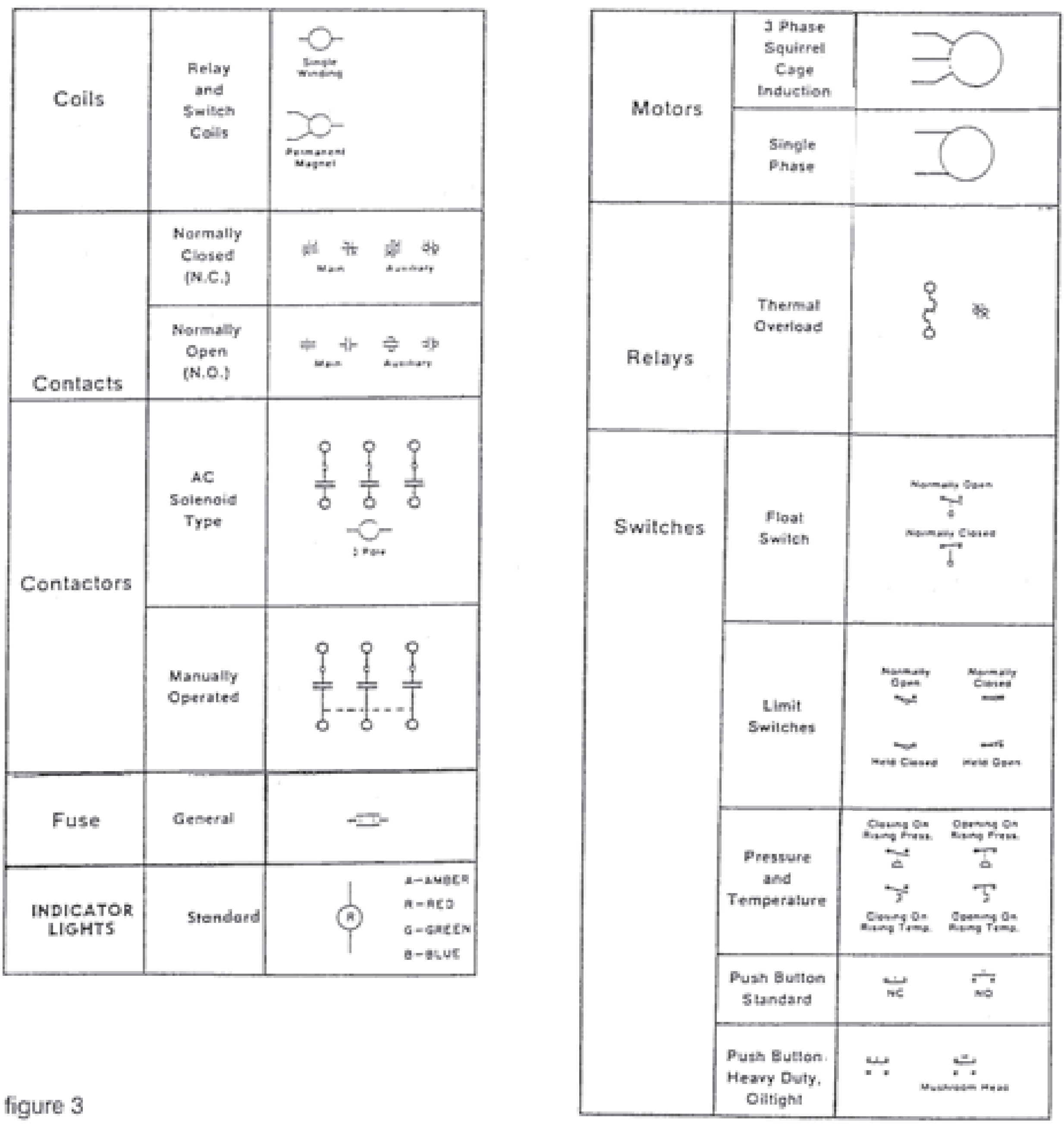

figure 3

Pump Primer

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Instrumentation – Introduction

Instrumentation is an outgrowth of man's desire to effect some means of accurate, repeatable measurement which can be readily applied to widely varied tasks.

In essence, instrumentation replaced man's personal inaccurate methods of estimating and made him aware of the need for concise standards.

The development of temperature measurement (Fahrenheit) in the 18th century, the invention of the pressure gauge (Bourdon) along with the discovery of electricity in the 19th century, provided the basis for the exponential advances which have occurred in this century. Progress in all areas, but particularly, in the electronic and pneumatic fields, provided the ingredients needed for developments in computers, data loggers, transmitters, etc. during the last several decades. Combining all these segments has resulted in the emergence of a new engineering field known as "Control Engineering."

While still in their infancy early in this century, gauges, meters, thermometers, etc. containing individual readouts were typically employed in the processing and power industries. Operators made periodic visual checks of each readout. Later, processors began to record this data, usually at specific time frames during operating cycles, using "clip-board" recording methods to either analyze on site or to take back to the office for calculation and analysis. Still later, in the 1930's and 40's, measured data was transmitted to a control room and monitored from a central area.

Since then, two types of transmitting systems have evolved - pneumatic and electronic. The pneumatic system usually employs air pressure at 3-15 psig; the electronic system requires 4-20 milliamps of direct current. Several factors - cost, size of plant, transmission distance, response time, reliability, etc. - dictate system choice.

Next, a strong force was required from the transmitter output signal to the controller in order to develop a mechanical action. The operating principles of a controller and transmitter are similar except that higher levels of energy are required to effect a physical change in the processing equipment. Actually, input/output devices are the interface between the controller and controlled system. Input devices convert physical qualities to electrical signals while output devices allow the controller to act on the system.

A transducer is a conversion device for translating the changing magnitude of one kind of quantity into corresponding changes of another kind of quantity. The latter quantity often differs from the former dimensionally and serves as the source of a useful signal. The quantities represent input and output. Transducers commonly measure pressure, temperature, light, magnetic fields, etc. Both mechanical and solid-state devices can be found in today's systems.

The five basic variable segments of measurement in a system by an element (transducer, gauge, thermometer, etc.) are:

1. temperature
2. pressure and strain
3. flow and level
4. pH and conductivity
5. data acquisition

Temperature

Temperature was probably the first measured variable and continues to be a critical system parameter. Measurement techniques from radiation pyrometry to basic filled systems are utilized.

The temperature sensing element should be located close to the point of desired reading. When media of different temperatures are mixed, the sensing element must be properly positioned to obtain a true mixed-temperature reading. If a temperature well is employed for the sensor, clean the well and use a heat transfer medium within the well and the measuring instrument. Temperature sensors, readouts and similar elements are commonly used with a device containing relay contacts to operate an audible alarm or annunciator.

Pressure-Strain

This is primarily a parameter indicator which provides immediate response to a point of operation within a process. Pressure instruments are usually categorized by their pressure-sensitive element and range. Most elements in use today were initiated by the development of the Bourdon tube which can be described as a flattened tube bent into a C-shape. As pressure increases, the tube flexes and tries to straighten out. A linkage mechanism employs a pointer to indicate pressure.

Improvements to the original Bourdon bent tube design, including the use of helixes, spirals, bellows and different materials are utilized today for the element in specific areas of application.

Flow

Measuring flow rate is important for individual pumps as well as for the entire process or plant. It is also important in some process plants where loops carry energy-laden media from which excess energy might be saved or reclaimed.

Many forms of flow measurement are available. Selection should be based on which is most applicable to a system's accuracy, type of flow (turbulence or laminar), viscosity, quality of the fluid medium (clean vs dirty), temperature, conductance, service and cost.

Forms of measurement include:

1. Differential Pressure
2. Magnetic
3. Mass
4. Oscillatory
5. Positive Displacement
6. Target
7. Turbine
8. Ultrasonic
9. Rotometer
10. Wiers and Flumes

Level

Liquid level measurement systems use a broad range of technologies to provide accurate and repeatable data. Common categories of liquid level sensing systems include: non-contacting, inserting probes, mechanical interface, hydrostatic head pressure, process pressure and interface. As with other instrumentation, several variations will usually do the job, so the choice becomes one of cost, accuracy and personal preference based on previous experience.

The current trend is away from mechanical devices and toward electronic and non-contact, non-intrusive level measurement. High growth rates are predicted over the next five years in the use of load cells, magneto structure, microwave/radar, nuclear and servo gauge instruments for level measurement.

Conductivity and pH

Conductivity/resistivity measurements are used in many applications: water conditioning, waste streams, reverse osmosis, acid, alkalis and salt concentrations, electroplating, etc.

The conductivity of any solution depends upon the presence within that solution of small, electrically charged particles known as ions. Conductivity measures the ability of a solution to conduct an electric current between two electrodes. As the ions present in the solution increase, the conductivity of that solution will become higher. If the number of ions is very small, the solution will be "restive" to current flow.

pH is a unit of measure which describes the degree of acidity or alkalinity of a solution. Measurement is made on a scale of 0 to 14. A pH of 7 represents a neutral solution. Lower values represent acidity and higher values alkalinity.

A rough indication of pH can be obtained using pH papers or indicators which change color as the pH level varies. More accurate measurements are obtained with pH meters. A pH meter is basically a high impedance amplifier that accurately measures voltage (minute electrode) and displays the results in pH units on either an analog or digital display.

Data Acquisition

Data acquisition systems measure the product and/or process used to collect information for documentation or analysis. Equipment ranges from plain recorders to complex computer systems. Data acquisition can serve as a control point in a system, bringing together a wide range of devices, such as sensors, to indicate temperature, flow and pressure.

An elementary data acquisition system has four components: (1) analog transducers, (2) one or more analog-to-digital converters to digitize transducer signals, (3) a controller to synchronize data sampling and storage, and (4) a processor which may share the same computer as the controller. Data can be stored by the controller itself, or passed to another computer.

Data acquisition systems are becoming very popular in manufacturing processes. The system or its controller may be a personal computer, a micro-processor, a single-board computer or a main-frame computer. It may be applied solely to acquire and analyze data, or it may also perform additional functions.

Display Choices

Today, instant displays are usually analog or digital. Each type has its advantages and limitations. Analog instruments are normally smaller, easier to read and less expensive while digital units are more accurate. Conversion of analog input to digital signals within instruments is achieved through special analog-to-digital circuitry.

The Institute of Electrical and Electronics Engineers (IEEE) classifies electrical transmission signals as: voltage, current, position, frequency and pulse with a further classification of (1) analog, where the signal transmitted is the electrical analog of the measurement or (2) digital, where the signal has been converted to a code representing the measurement. All five electrical transmission signals can be used in analog transmission but only the pulse signal can be used for digital transmission

General Service

In our service-oriented society, some overlapping of service in the instrumentation field has become evident. There is a growing demand for high-level control engineers representing, to some degree, a shifting of personnel from user processing companies to engineering service firms. It's uncertain whether this is a permanent trend. When service is beyond a maintenance staff's capability, a service contract alternative may be pursued to insure minimum down-time of the processing equipment.

In this environment, brand changing is also becoming rare. Processors are staying with one or two sources for the improved service that a technician familiar with the system can offer.

Preventative Maintenance

Programs involving routine inspections, cleaning, lubrication, replacement of spare parts and safety audits can be beneficial. A good preventative maintenance program should follow of schedule of specific times, dates or hours of operation predetermined by previous history or dictated by equipment manufacturers' specifications. Unplanned malfunctions or breakdowns must, unfortunately, also be anticipated in any good maintenance program.

Accuracy

Accurate measurements can only be assured if the monitoring is performed at a location in conformance with the instrument manufacturer's instructions. Dirt buildup or corrosion on either the element or equipment can result in erroneous readings.

Summary

The essentials of an instrumentation system include:

1. A sensing or measuring element
2. A means of comparing the measured value with an observed value
3. A final control element to provide the desired change in the measured variable
4. An actuator to change the position of the final control element as required
5. A relaying or force-building means of enabling a weak sensing signal to release sufficient force to power the actuator

To be most effective in making compatible instrument recommendations, control manufacturers must be provided with complete data in three areas: (1) application/environment, including the medium being processed, its temperature, properties, etc., (2) accuracy requirements, (3) instrument reading range.

Choosing the right transmitter for an application improves process efficiency. Choosing the right vendor assures that the entire process, from ordering the instrumentation to installation and maintenance will be accomplished smoothly and efficiently.

Instruments for the measurement of temperature, pressure and liquid level can usually be installed from standard drawings. Differential-pressure instruments, particularly those used for measurement of flow rate, generally require more detailed sketches of the specific operation.

To maintain a reliable processing operation, instrumentation should be recalibrated periodically in conformance with the manufacturer's recommendations and a regular cleaning program should be observed.

These procedures should be followed even though current instrumentation offers: increased reliability and durability, improved performance, less maintenance, greater consistency and lower overall cost (taking into consideration life cycle and maintenance).

Relative to pumping applications, instrumentation can provide:

1. An evaluation of performance along with constant monitoring of the pump and system.
2. Capability of allowing pump or system to automatically respond to change: **(a)** pump through speed control, **(b)** system through repositioning of control valves.
3. Ability to modify or maintain fluid temperature (could also control viscosity).

This article concludes the Pump Primer program which originated in 1988. If you're interested in receiving back issues, all twenty articles are still available and can be provided upon your request.

Address your request to:

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