CENTRIFUGAL PUMP APPLICATION—KEY HYDRAULIC AND PERFORMANCE CRITERIA

by

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INTRODUCTION

The essential elements to be reviewed when considering a centrifugal pump application are highlighted. The purpose is to ensure the procurement and/or application of a centrifugal pump which optimizes reliability, energy usage, maintenance costs, and achieves this in a safe manner. The need to understand the system hydraulics and complete process requirements is emphasized, as is the impact each system parameter has on pump design. The text is broken into three basic areas: pump boundary conditions, flow requirements, and pump specification—key elements.

Pump boundary conditions addresses NPSH considerations as they relate to flow requirements, potential for cavitation damage, suction specific speed, and mechanical seal temperature margin. This section also addresses system resistance variations, and addresses the effects of variations in dynamic and static system heads and the potential prime causes of variations in these heads.

Flow requirements discusses the determination of normal, maximum (or rated), and minimum flow requirements. A method of approximation of a pump's minimum acceptable flowrate is presented (based on the pump's hydraulic specification), which utilizes suction specific speed and moderating factors such as impeller head, NPSH margin, specific gravity, etc. Attention is paid to the effect of excessive specified flow on NPSHA, flow rangeability, and suction specific speed. Mechanical and hydraulic interrelationships are discussed to emphasize how flowrate variations affect radial and axial thrust, impeller suction and discharge recirculation, and potential for cavitation; also covered are the effect of variation in running clearances and changes in specific gravity. Critical aspects of both parallel and series pump application are addressed. The dangers of incorrect parallel pump application are highlighted and guidelines are offered for correct application of parallel pumps. Series pump operation is discussed primarily in terms of percentage split in pumps' combined total head, pump protection and system protection.

Pump Specification—Key Elements addresses communicating to prospective pump vendors the key parameters that may determine the metallurgy, mechanical seal design and mechanical seal peripheral's design, and pump hydraulic design best suited to the user's requirements. The liquid specification is highlighted, the important site and operating conditions are addressed, and the importance of clearly defining the key performance requirements in full is discussed.

The wrong pump, operated incorrectly, coupled with poor maintenance practices, results in unsatisfactory hydraulic performance, high energy costs, high maintenance costs, poor reliability, and increases the potential for an unsafe failure. Proper specification and selection principles, coupled with the implementation of correct operating and maintenance practices, will result in optimized centrifugal pump application. The net result will be a pumping application that meets all process demands, with a minimum of energy usage, and a low frequency of repair (high reliability), low overall maintenance costs, and low process debits.

Key Hydraulic and Performance Criteria are addressed along with their effect on performance, energy, reliability, maintenance costs, and safety. The interrelationships that exist between the hydraulic characteristics and the mechanical reliability is highlighted to ensure that neither are treated separately.

KEY HYDRAULIC TERMS

A review of the key hydraulic terms used when applying centrifugal pumps is called for to allow clear focus on their relative importance.

The term *head* is used instead of pressure or differential pressure, when referring to a centrifugal pump's performance, since a centrifugal pump generates head (sometimes referred to as total head or differential head), not pressure. The *head* generated is a measure of the increase in specific energy (energy per unit mass) of the fluid between the pump suction and discharge. Typically, it is foot-pounds per pound, which translates to simply feet (or meters), as the pound units cancel each other. Pump head should be considered as an energy term and not as a linear term when considering a centrifugal pump application.

Since this energy term is related to unit mass flow, and pump capacity is usually measured in terms of *volumetric flowrate* (e.g., U.S. gpm, or meters^3/hr), it is necessary to introduce a term to convert volumetric flowrate into mass flowrate; hence the use of fluid specific gravity (SG) in the calculation of *pump horsepower*. Equations (1) and (2) show the relationship between pump power and SG. For a given volumetric flowrate, *a centrifugal pump's* *horsepower will vary linearly with specific gravity*, while the pump's head will always be the same for any specified volumetric flowrate.

$$\text{Pump Hydraulic Power (horsepower)} = \frac{(Q.H.SG)}{3960} \quad (1)$$

$$\text{Pump Hydraulic Power (kw)} = \frac{(Q.H.SG)\ 9.81}{3600} \quad (2)$$

Where:

Q is flowrate in; U.S. gpm in Equation (1), and M^3/h in Equation (2).

H is head at flowrate Q in: feet in Equation 1, and meters in Equation 2.

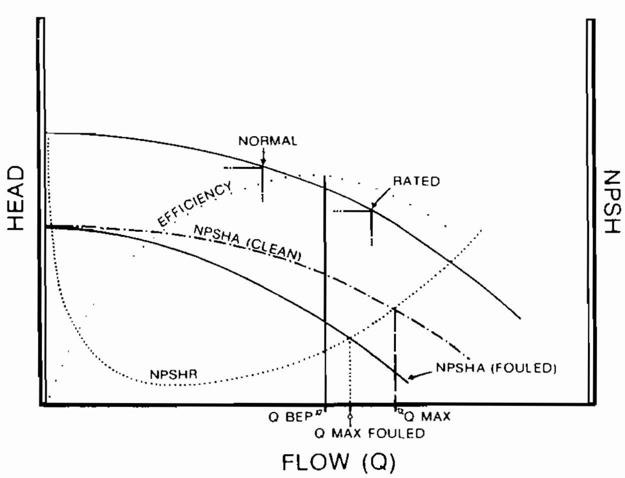
SG is specific gravity of the fluid at pumped temperature.

Net positive suction head (NPSH), while defined in linear terms, is really a measure of the specific energy in the fluid over and above that specific energy required to maintain the fluid in the liquid phase. It is defined as foot-pounds per pound (as is pump head), which, again, translates into simply feet (or meters). The fluid vapor pressure is subtracted from the calculated (or measured) pump suction pressure, resulting in a net pressure above the fluid vapor pressure. This net pressure (in psi) is converted to fluid head (ft) by multiplying by 2.31 and dividing by fluid specific gravity. Alternatively, this net pressure (in kPa) may be converted to meters by dividing by (9.81 × SG). The term net positive suction head must be defined to have real meaning.

The term net positive suction head available ($NPSH_A$) is a process system characteristic and reflects the fluid head losses in the suction piping system. $NPSH_A$ is the value of NPSH which, for a specific flowrate, will exist at the pump suction flange. It may be calculated for a planned pumping system, or it may be measured for an existing pumping system.

The term net positive suction head required ($NPSH_R$) is a pump characteristic, and defines the NPSH required by the pump, for a specific flowrate, to avoid loss of performance due to cavitation.

The $NPSH_A$ decreases as pump flowrate increases. The $NPSH_R$ increases as pump flowrate increases. Pump flowrate will be limited to the point at which these two values coincide. (The relationship between $NPSH_A$ and $NPSH_R$, and the effect of suction piping fouling on $NPSH_A$, is shown in Figure 1).



Figures 1. $NPSH_A$ Vs FLOW—The Effect of Suction System Fouling on Maximum Attainable Flowrate.

A point to note with respect to the true calculation of NPSH: The $NPSH_R$ for a centrifugal pump is based on zero fluid velocity at the pump suction flange. This means that the velocity head of the fluid should be added to the $NPSH_A$ based on pressure measurement alone, when accurate $NPSH_A$ is required. On average, the velocity head in a piped inlet (as opposed to the inlet being submerged in the liquid) will calculate at 1 to 1.5 ft (0.3 to 0.5 meters), and is often overlooked when calculating $NPSH_A$ from field pressure measurements, except in marginal situations. The following equations (Equations (3) and (4)) for $NPSH_A$ take into account the velocity of the fluid at pump inlet:

$$\text{NPSHA (feet)} = \frac{[P_s+P_A-P_V].2.31}{SG} + \frac{[Vo^2 - V_i^2]}{2g} \quad (3)$$

or

$$\text{NPSHA (meters)} = \frac{[P_s+P_A-P_V].2.31}{9.81\ SG} + \frac{[Vo^2 - V_i^2]}{2g} \quad (4)$$

Where:

P_s is pump suction pressure: in psig for Equation (3), and in kPaG for Equation (4).

P_a is local atmospheric pressure: in psia for Equation (3), and in kPaA for Equation (4).

P_v is fluid vapor pressure at pump suction: in psia for Equation (3), and in kPaA for Equation (4).

SG is fluid specific gravity at pump suction

V_i is fluid velocity at pump inlet flange: in ft/sec for Equation (3), and in meters/sec for Equation (4).

V_o is fluid velocity at pump outlet flange: in ft/sec for Equation (3), and in meters/sec for Equation (4).

g is the gravitational constant: in ft/sec^2 in Equation (3), and in meters/sec^2 for Equation (4).

The terms *specific speed* (S) and *suction specific speed* (S_s), are key centrifugal pump characteristics that allow comparison of pumps that may be different in size, but exhibit similar hydraulic characteristics. Both terms are essentially nondimensional, but are offered in their customary form for ease of calculation and reference.

Specific speed is based upon head, flowrate and rotational speed, as shown in Equation (5). *Suction specific speed* is based upon $NPSH_R$, flowrate and rotational speed, as shown in Equation (6).

$$S = \frac{N \times [Q_{bep}]^{0.5}}{[H_{bep}]^{0.75}} \quad (5)$$

$$S_s = \frac{N \times [Q_{bep}]^{0.75}}{[NPSHR_{bep}]^{0.75}} \quad (6)$$

Where:

S is pump specific speed.

S_s is pump suction specific speed.

N is pump rotational speed in rpm.

Q_{bep} is the bep flowrate for maximum diameter impeller.

H_{bep} is the head at bep for maximum diameter impeller.

$NPSHR_{bep}$ is the $NPSH_R$ at bep for maximum diameter impeller.

Bep refers to best efficiency point.

While the relationship between *specific speed* (S) and *head and flow at bep* is nonlinear, it is clear from Equation (5) that the ratio flowrate/head will increase as S increases; decreasing the head at bep flowrate will result in a higher value of S, regardless of the size of pump. Similarly, while the relationship between *suction specific speed* (S_s) and *NPSH$_R$ and flowrate at bep* is also nonlinear, it is clear from Equation (6) that the ratio of flowrate/NPSH$_R$ will increase as S_s increases; decreasing the NPSH$_R$ at bep flowrate will result in a higher value of S_s, regardless of the size of pump.

PUMP BOUNDARY CONDITIONS

NPSHA and NPSHR

Net Positive Suction Head Available (NPSH$_A$)

The term NPSH$_A$ refers to the available net positive suction head at the pump suction flange (or inlet), for a specified flowrate. It is entirely a process system characteristic and is independent of the pump suction requirements.

Net Positive Suction Head Required (NPSH$_R$)

The term NPSH$_R$ refers to the minimum net positive suction head required at the pump suction flange (or inlet), for a specified flowrate, to avoid loss of performance due to cavitation. It is entirely a pump characteristic and is independent of the piping or flow characteristics of the process system in which the pump will operate.

Net Positive Suction Head Considerations

It follows that the NPSH$_A$ of the system must always equal or exceed the NPSH$_R$ of the installed pump to avoid performance decline and noticeable cavitation. Any demand for a flowrate greater than that which exists at the point where the NPSH$_A$ equals the NPSH$_R$ will not be met regardless of any further reduction in pump discharge pressure. A typical relationship between NPSH$_A$ and NPSH$_R$, defining the limiting flowrate due to insufficient NPSH$_A$ is shown in Figure (2) [1].

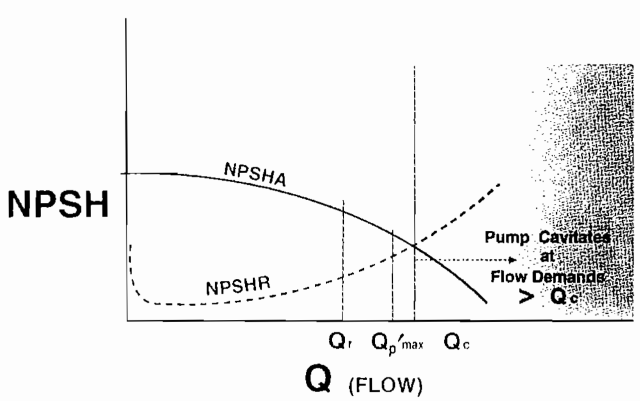


Figure 2. NPSH$_A$ Vs NPSH$_R$—Flowrate Limited to that Attainable where NPSH$_A$ Equals NPSH$_R$.

Developing a Pump NPSH$_R$ Curve

Pump manufacturers develop the NPSH$_R$ curve by testing the pump on their test loop, which incorporates precisely calibrated instrumentation for pressure, temperature, flowrate, and power measurement. Tests are almost always conducted with water as the pumped fluid (testing with a fluid other than water is costly and seldom specified). A performance test is performed prior to the NPSH$_R$ test, to determine the head/flowrate relationship for the pump. At a constant flowrate, the pump suction pressure is decreased to the point where the head at that specific flowrate, (as determined by the performance test), has fallen to 97 percent of the performance curve value: the corresponding calculated NPSH$_A$ at the pump suction flange is considered as the pump's NPSH$_R$ for that specific flowrate. This is repeated at various flowrates to develop an NPSH$_R$ curve. *Since the criteria for the development of this curve is based on a three percent drop in expected head, it is worth noting that a pump that is operating with an NPSH$_A$ that is equal to the NPSH$_R$ is already in partial cavitation.*

NPSH Margin

The differential between NPSH$_A$ and NPSH$_R$ is referred to as the NPSH margin. To avoid operation of a centrifugal pump under partial cavitation it is necessary to have an NPSH margin. There is no hard and fast rule which defines the numerical value of NPSH margin, which is sufficient to avoid partial cavitation and all of the three percent head drop accepted during the determination of the pump's NPSH$_R$ curve. One consensus holds that three ft (or one meter) is sufficient NPSH margin, and this value may be accepted as a general rule of thumb; any situation requiring the acceptance of a closer margin should call for close surveillance of the NPSH$_R$ test as confirmation.

A typical example of the values of NPSH margin which may be necessary to avoid cavitation is shown in Figure 3 [2]. From this example, it becomes obvious that cavitation has more than one form, as the NPSH$_R$ clearly decreases with decreasing flow, whereas the NPSH margin increases with flows above and below bep flow. (The other forms of cavitation are discussed under *Suction Specific Speed and NPSH$_R$*.)

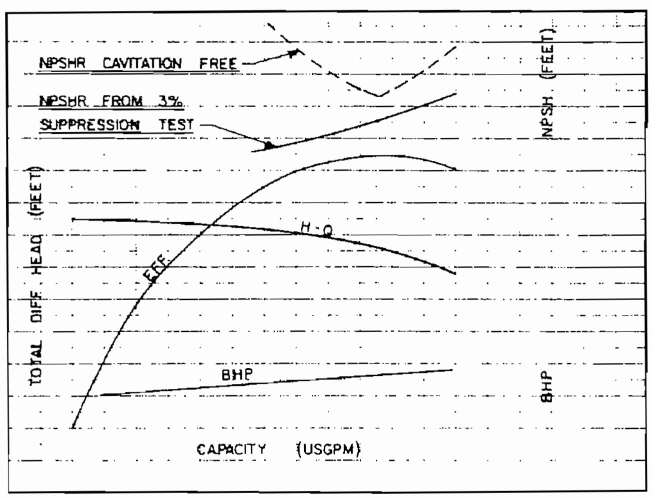


Figure 3. NPSH Margin Required to Avoid Cavitation. NPSH vs *flow (a comparison of NPSH$_R$ to avoid cavitation vs NPSH$_R$ based on three percent head drop.*

A typical relationship between flowrate and NPSH margin to avoid damage to pump components, due to flow instabilities within the pump, is shown in Figure 4 [3]. The shape of the curve of NPSH$_R$ to avoid damage will vary with other pump characteristics (e.g., suction specific speed), and this is discussed under *Suction Specific Speed and NPSH$_R$*.

The bep point is the flowrate at which the NPSH margin required to avoid damage to the pump impeller, bearings, and seals is least. While the actual NPSH margin increases as the flow falls below bep flow, the NPSH margin required to avoid damage also increases. Where the increased demand for NPSH margin, as flow decreases, is not met, more frequent pump failures will occur, and

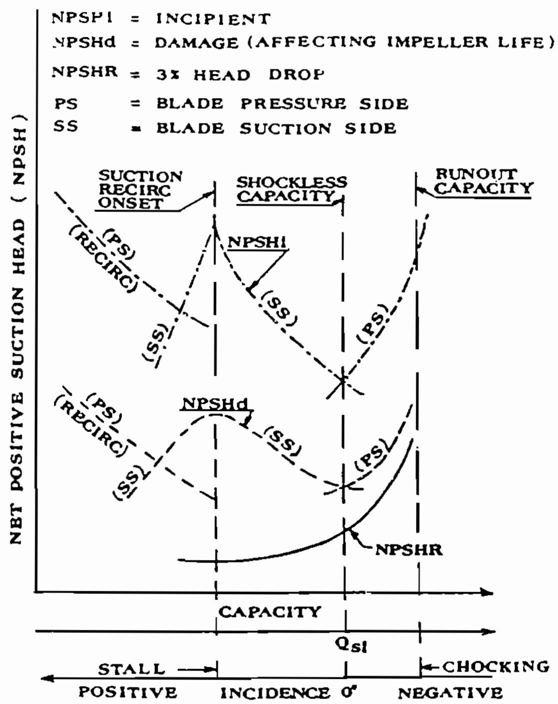


Figure 4. NPSH Margin Required to Avoid Damage and NPSH Margin at Which Incipient Cavitation Occurs.

pump hydraulic performance may decline due to impeller and casing erosion.

Suction Specific Speed (S_s) and $NPSH_R$

Suction Specific Speed (S_s) is a calculated value (Equation (6)), and is a characteristic of the pump casing and impeller design. The higher the value of S_s, the lower the NPSHR to avoid cavitation at a specific flowrate. The disadvantage is that the range of flow stability decreases as the value of S_s increases, although this disadvantage may be partially offset through careful attention to specific impeller characteristics [3]. This flow instability is associated with impeller suction and discharge recirculation. The point at which flow instability causes suction recirculation is the point at which the NPSH margin, to avoid damage and/or severe cavitation, begins to rise rapidly as flowrate decreases. A point is reached, at some flowrate below the onset of suction recirculation, where the severity of the flow instability is such that early pump failure is probable. This is the point of recommended minimum flowrate for the pump, and is dealt with under *Criteria for Determination of a Pump's Acceptable Continuous Minimum Flowrate.* The key principle here is that a lower specified $NPSH_A$, where it is marginal, will lead to a pump with a higher suction specific speed, and potentially lower flexibility in flowrate. Typical flowrates where damage may occur, relative to the onset of suction recirculation, are illustrated in Figure 5.

SYSTEM RESISTANCE

The *system resistance* curve must be clearly defined. How much static head is built into the pump discharge in terms of downstream pressure in a receiving vessel, or height which must be overcome to reach the vessel? How quickly does the system resistance increase with increasing flowrate? A quickly rising curve may preclude a maximum flowrate, expected periodically, which is considerably in excess of the normal flowrate. Control valve sizing will be affected by the rate of rise of the curve as will the size of pump. A larger than normal control valve may be required to

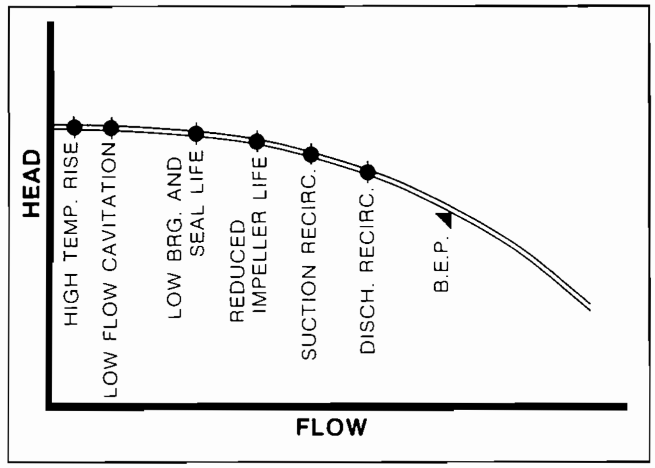


Figure 5. Head Vs Flow Illustrating Point of Onset of Events that Adversely Affect Pump Operation.

provide the artificial head loss at rated and minimum flowrates, while still accommodating the low loss it must provide at maximum expected flowrate. The effect of system resistance on maximum possible flowrate and required control valve head loss is shown in Figure 6 [4]. In new installations, pipe size may be increased to flatten a steeply rising system resistance curve to accommodate greater flow flexibility. The advantages of an increase in piping diameter are illustrated in Figure 7.

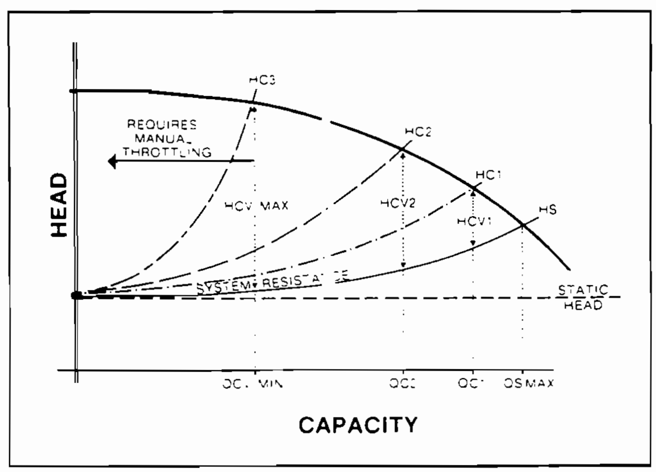


Figure 6. Effect of Control Valve on the System Resistance Curve.

The *differential* pressure that the pump sees will be derived from the system resistance curve. This must be converted to differential or total head (H). In arriving at H, the range of *specific gravity* (SG) expected must be reviewed, as any lowering of SG will require additional pump head to meet required discharge pressure conditions. (Maximum head requirement should be based on the lowest expected SG. Horsepower requirement should be based on the highest expected SG.) The potential for suction and discharge system resistance increases must be considered in arriving at a true value of maximum expected head for a given flowrate. Suction strainer plugging or heat exchanger fouling are typical of such increases in resistance and short cleaning intervals may be necessary where fouling is rapid. A new piping system, after chemical cleaning, will present the optimum cleanliness that is often not

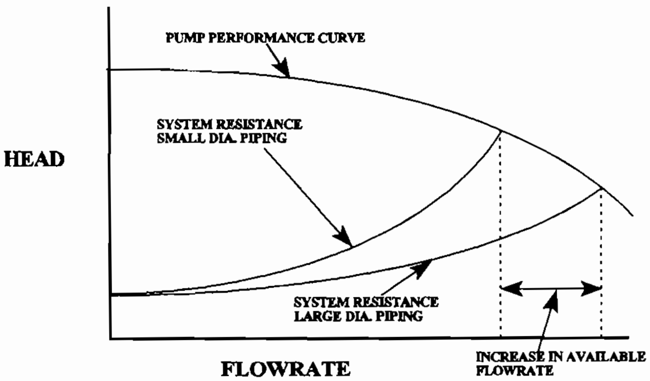


Figure 7. Effect of Pipe Size Increase on Flow Flexibility.

attainable thereafter, and this must also be factored into the initial calculations of resistance.

While defining system resistance to accommodate normal and changing conditions, it is also important not to be overly conservative. Imposing excessive head values on a pump specification for given flowrates will result in the pump operating much below its bep point, and in the lower efficiency region, when these heads prove to be lower than expected. Reliability and maintenance costs will suffer.

The static component of the discharge system resistance can also limit maximum capacity. Where the possibility exists of an increase in differential height between the liquid source and its delivery point, or an increase in the pressure of the receiving downstream vessel or a decrease in the pressure of the suction vessel, these must be looked at in determining rated conditions.

A simplified schematic diagram and head flow curve illustrating these points is shown in Figure 8 [4]. Rated flow must be possible at the greatest expected total discharge system resistance. Where

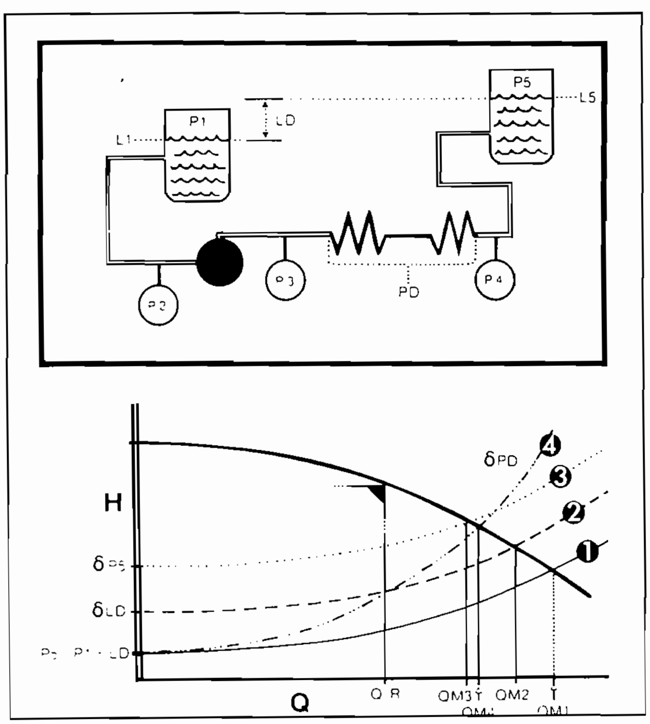


Figure 8. Effect of Variations in Static Head and System Resistance on Maximum Attainable Flow.

the long range outlook may call for step changes in total static head, the type and size of pump must be tailored to accommodate such, through possible increases in impeller diameter. Space flexibility may permit a more flexible pump to be offered (e.g., a double suction between bearings *vs* a single stage overhung or vertical inline design).

FLOW REQUIREMENTS

An approach that has been determined practicable and results in all necessary flow requirements being met is presented here. While there are many different opinions and papers on the subject of pump flow requirements, the following is based on personal experience with a workable approach utilizing some applicable industry findings. Improvements in impeller and volute designs have enabled some pumps to operate satisfactorily at flowrates below those minimum acceptable, arrived at through the approach offered by the author [3]. In critical situations it is always advisable to consult with a knowledgeable pump manufacturer's applications engineer.

DETERMINATION OF FLOWRATES

Flowrates in the petroleum industry are generally termed normal and rated, and these terms may be applied to any centrifugal pump application. The *normal flow* is the flow at which the equipment will usually operate. The *rated flow* is the guaranteed flow at specified guarantee point operating conditions.

When determining these design flowrates, care must be taken to avoid an extremely conservative approach. This is another area where higher than expected flow requirements will result in a larger than required pump (as in head considerations). This may be further complicated where the size and the design of pump may be altered to comply with these high flowrates. A more simple single stage, overhung pump application may require a double suction between bearings design under increased flow requirements.

The rated flow should reflect the maximum flowrate the system can envisage under current consideration, but also must consider the long range outlook. Minimum flow requirements can conflict with rated requirements and recirculation facilities may be required.

While it is of prime importance to define maximum and minimum flow requirements properly, it is also important to clarify the percentage of time over which the pump will operate at minimum, normal and rated (or maximum) flowrates. Where a pump is used for two very different services, the lower flowrate may require excellent turndown while the higher flowrate will impose more stringent $NPSH_A$ restrictions. Longterm operation at the lower flowrate can mean higher maintenance costs due to higher bearing loads and shaft deflections, and may result in high energy consumption due to prolonged operation at low hydraulic efficiencies. The relationship between radial bearing load and flowrate is shown in Figure 9. (Note: A general rule for rolling element bearings is that bearing life is inversely proportional to the cube of load.)

PROPOSED METHOD FOR DETERMINATION OF A PUMP'S ACCEPTABLE CONTINUOUS MINIMUM FLOWRATE

The possibility of physically or hydraulically shutting off the pump at its discharge must be considered. Recycle facilities may again be required to protect the pump.

Where complete *shutoff* (discharge isolation) of a pump is an expected occasional occurrence, provision must be made to recycle flow to prevent the pump from vapor locking, due to overheating of the trapped fluid. The minimum recycle flowrate required to protect from shutoff is a function of the time over which shutoff of actual delivered process flow will be maintained and the ability of

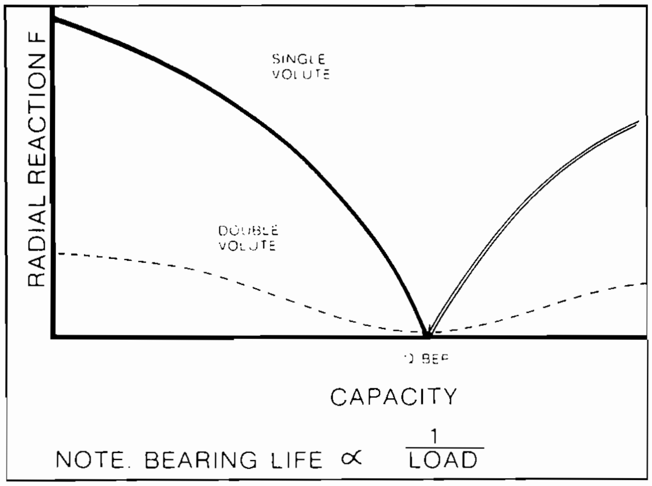


Figure 9. Relationship Between Flowrate and Radial Bearing Load.

the pump to accommodate low flow conditions. The recycle requirements to protect the pump during shutoff conditions will generally be much less than required to protect the pump when operating at minimum continuous flow. Where shutoff will be for a short interval of minutes rather than hours, a recycle flow of 10 percent of bep will normally suffice. For minimum continuous flow, a total flow of 30 to 40 percent of bep is more realistic, although this can be much higher for high S_s, and/or high head (H) pumps. Capital cost of recycle facilities is a major consideration here and the desirability of specific low flow (turndown) capabilities must be highlighted.

Expected minimum continuous flowrate from an operational or process viewpoint may be less than is recommended for reliable, low maintenance service. Various hydraulically related factors and phenomena display themselves, and may be listed as:

- Suction recirculation.
- Discharge recirculation.
- Reduced impeller life.
- Reduced bearing and seal life.
- Low flow cavitation.
- High temperature rise.

These effects were shown graphically in Figure 5. Generally, the first four listed will determine what minimum flow is considered acceptable.

The percentage of bep flow at which discharge and suction recirculation occur within the impeller is a function of pump design and impeller geometry. For a given pump design, the flows at the onset of discharge and suction recirculation move closer to bep as the suction specific speed (S_s) increases. This means that, for a specific pump design, pumps have low values of $NPSH_R$, and, consequently, have higher S_s values, will experience unstable flow patterns at a higher percentage of bep flowrate.

The effects of the localized cavitation due to impeller recirculation will increase in severity as flow is further reduced. A point will be reached where normal impeller life is significantly reduced with performance decline showing up after a short run time.

For any specific impeller design the effect of lower NPSH requirements may be shown as follows:

Lower NPSH Required
!
Higher S_s Value
!
Larger Impeller Eye Diameter
!
Higher Capacity at Suction Recirculation
!
Higher Minimum Flow
!
Narrower Range of Trouble-free Operation

Note: Where $NPSH_A$ is very low, a deepwell pump is often considered as an alternative, where the depth of the outer casing below the suction flange centerline allows the first stage impeller to be submerged, adding to the NPSHA.)

A graphical method for estimating the onset of suction recirculation is offered in Figures 10 (5). Both S and S_s should be known.

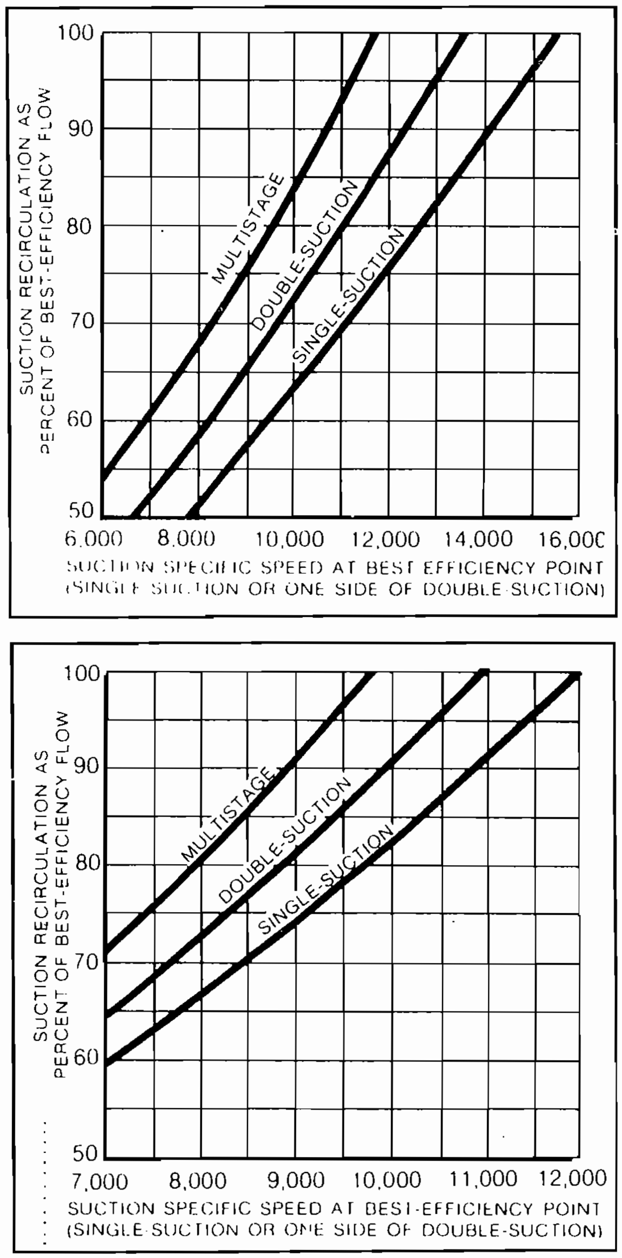


Figure 10. Recirculation Flow Vs *Suction Specific Speed for Two Specific Speed Ranges. a) from 500 to 2500; b) from 2500 to 10,000.*

The location of suction and discharge recirculation within an impeller are shown in Figure 11.

As a general rule, the following acceptable minimum flowrates are recommended (5):

• Water pumps operating at below 2500 U.S. gpm and 150 ft head may operate satisfactorily at minimum flowrates of as low as 50 percent of the suction recirculation values shown.

• For hydrocarbon operation, flows as low as 60 percent of the suction recirculation values shown may be accepted as satisfactory minimum continuous flows.

Where acceptable minimum flowrate, as determined by these criteria, is below that planned for the process, a more detailed review may be called for. This is particularly the case where vessel elevations cannot be altered or pumps recessed, or where recycle flow costs would be exorbitant.

Further review of criteria associated with NPSH margin, specific gravity, percentage of time planned at minimum flow, and power density, as these factors relate to acceptable minimum continuous flowrate, are offered by Gopalakrishnan [6]. They are worthy of consideration in most applications.

Before calling for construction modifications to improve $NPSH_A$ or add or increase recycle flow, a detailed review by the pump manufacturer's applications engineer is recommended, as some pump designs are more tolerant of low flowrates than others with similar suction specific speeds.

In summary, the simple method of using 50 percent (water) or 60 percent (hydrocarbon) of the flowrate at the onset of suction recirculation, as the criterion for minimum continuous flow, will provide for acceptable pump operation in a large majority of centrifugal pump applications. Where cost factors are large, and/or physical plant limitations exist, further analysis may be warranted, in the determination of a more precise value for minimum acceptable continuous flowrate.

Operation below recommended minimum acceptable continuous flowrate can cause severe damage.

Knowledge of the foregoing considerations in regard to minimum flowrate will permit process designers to optimize design parameters for a pump to balance costs of surrounding structures and piping against expected pump performance. This is typical of an area where teamwork between the process designer, operations personnel, and the machinery specialist is essential.

GENERAL FLOW CONSIDERATIONS

Pumps with drooping head/flow curves, that result in a falloff in maximum head towards shutoff, are best operated well out towards the bep point for adequate flow control stability.

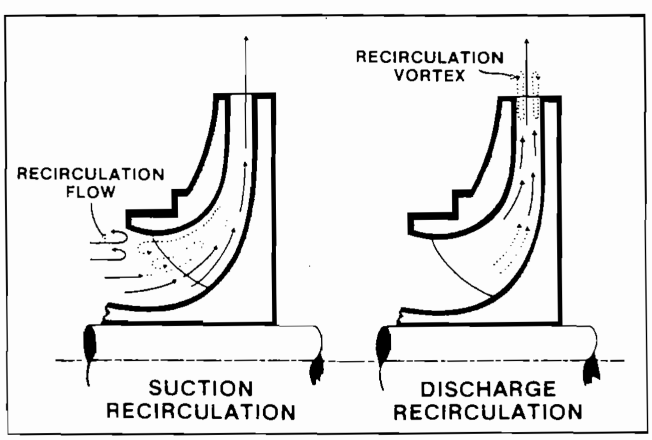


Figure 11. Position of Points of Recirculation Within an Impeller.

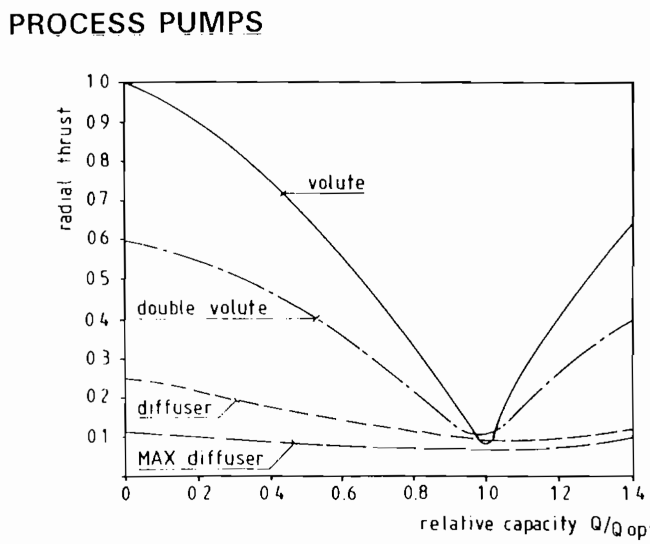
The *type of flow control* must be considered. Level control if it fails, resulting in a fully open control valve, may allow a pump to run out on its curve. A pump driver and $NPSH_A$ should be able to accommodate this and allow the pump to assume normal operation via manual control without motor trip or vapor locking. Flowrate control may be less likely to create similar problems, particularly where system resistance is a major part of the pump head. In any case, all types of flow control must consider what might happen to pump suction and discharge conditions under control failure.

Where a pump is required to provide a *dual service*, the lower flow may again require a *controlled recycle flow* to maintain the flowrate within the acceptable range.

KEY MECHANICAL/HYDRAULIC INTERRELATIONSHIPS

Radial and Axial Thrust

Impeller radial thrust is at a minimum at the bep flowrate (this is normally very close to the flowrate for shockless entry into the impeller). Increasing or decreasing the flowrate above or below this bep flowrate will result in an increase in radial thrust, with the degree of increase being proportional to the increase or decrease in flowrate. The effect of off-bep operation on radial thrust can be considerably lessened by providing a double volute casing. Radial thrust can be further lessened through installation of a diffuser ring. Radial thrust *vs* flow, for single and double volutes and diffuser designs, is illustrated in Figure 12.



Comparison of radial forces

Figure 12. Radial Thrust Vs *Flow for Single Volute, Double Volute, and Diffuser Designs (Courtesy of Stork Pumps).*

Impeller axial thrust is generally at a maximum at zero flowrate (shutoff). Increasing flowrate will result in a decrease in axial thrust, since axial thrust is primarily generated through differential pressure acting on the impeller geometry.

Physical Damage Due to Impeller Recirculation

The following are physical evidence of either discharge or suction recirculation:

Discharge Recirculation

• Cavitation damage at the vane's discharge on the pressure side of the vanes

- Volute tip or diffuser tip cavitation damage
- Axial shaft movement
- Shaft failure on the outboard end of double suction or multistage pumps
- Damage to impeller shrouds at outer diameter can extend to complete impeller failure

Suction Recirculation

- Cavitation damage at the vane's inlet on the pressure side of the vanes
- Damage to suction stationary vanes
- Suction surging
- Random suction crackling noise (instead of steady crackling noise as associated with low NPSH cavitation)

The points of low flow cavitation and *high temperature rise* are only valid considerations where extremely low flowrates are considered probable for short periods that may cause severe cavitation and eventual vapor locking of the pump. Such events may lead to rapid mechanical seal failure and require protection against for even a short duration of one to two minutes, where volatile liquids close to their vapor point are being pumped.

Localized damage areas within an impeller due to various types of cavitation are shown in Figure 13.

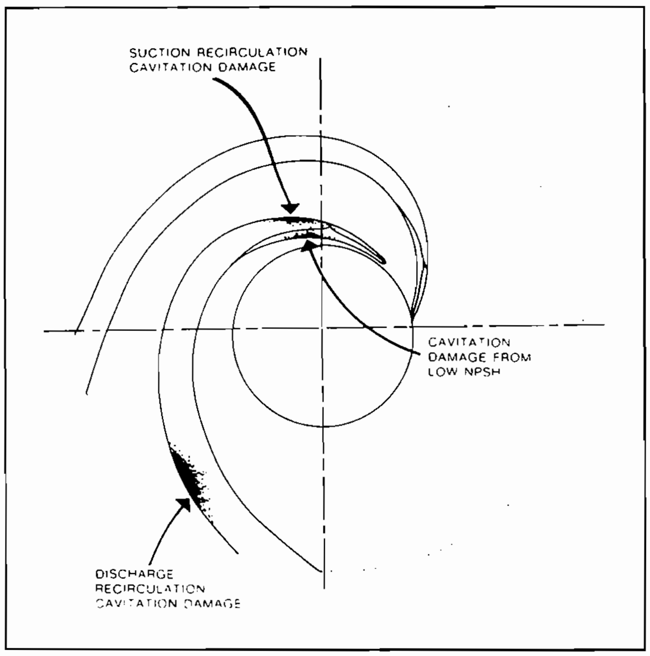


Figure 13. Areas of Impeller Damage Due to Cavitation Caused by Discharge Recirculation, Suction Recirculation, and Low $NPSH_A$.

High Flow or Very Low Flow Cavitation Damage

High flow or very low flow cavitation is distinct from cavitation due to impeller recirculation, as it occurs entirely due to insufficient $NPSH_A$, rather than flow instabilities. At high flowrates, the pressure loss due to acceleration of the fluid between the pump suction flange and the impeller eye, coupled with a degree of friction loss, can result in a drop in $NPSH_A$ to the point where the $NPSH_R$ for the specific flowrate is no longer met.

Very low flow cavitation is the result of most of the driver power causing an increase in fluid temperature, rather than in the delivery of process fluid. The resultant increase in temperature increases the fluid vapor pressure, with a resultant reduction in $NPSH_A$ to the point where the $NPSH_R$ is no longer met.

Either high flow cavitation or very low flow cavitation result in erosion of the impeller vanes (normally on the visible low pressure side), increased vibration levels, and increased potential for vapor at the seal faces. The result is reduced pump performance, and more frequent bearing and seal failures.

Flowrate vs *Shaft Deflection and Bearing Life*

The radial force acting on the impeller due to reduced flowrate (below bep) results in shaft deflection. This translates to increased movement between the rotating and stationary mechanical seal faces, that increases emission rates and reduces seal life.

The added load on bearings due to increased radial force acting on the impeller will have a pronounced effect on bearing life: the life of a ball bearing is inversely proportional to the cube of the load imposed upon it. Even double volute pumps, designed to reduce bearing load at off-bep operation, will exhibit reduced bearing life a low flowrates.

Effect of Running Clearances on $NPSH_R$

Increased wear ring clearance, (usually due to erosive wear), while having an impact on pump hydraulic performance, will also result in an increase in $NPSH_R$ for a specific flowrate. Typically, doubling of the wear ring clearance can result in up to 50 percent increase in $NPSH_R$. This relationship is represented in Figure 14 [7].

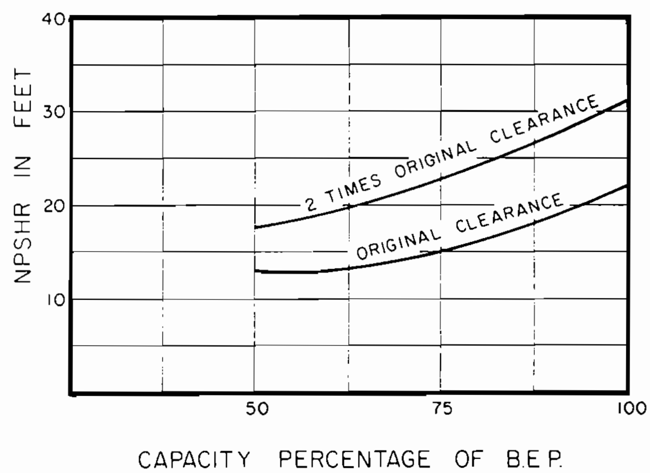


Figure 14. Typical Influence of Wear Ring Clearance on $NPSH_R$.

Effect of Running Clearances on Critical Speed

The calculation of dry critical speed of a multistage pump, by finite element analysis, is not truly indicative of the critical speed of a rotating assembly when operating in fluid. The dampening effect of fluid in the running clearances will alter this calculated critical speed [8], especially in multistage pumps. Changes in running clearances will alter the dampening coefficients, with resultant changes in the critical speed of the rotating assembly. This may prove to be very important in variable speed applications, or where the calculated critical speed is within 20 percent of operating speed.

Potential Effect of Changes in Specific Gravity

An increase in specific gravity will result in an increase in differential pressure for a specific flowrate, since a centrifugal pump produces a fixed head for a specific flowrate, and differential pressure is directly proportional to the product of head and specific gravity. This will cause an increase in thrust load and radial load for a specific flowrate. Driver power demand will also increase, since driver power is directly proportional to specific gravity, as illustrated in the following formulas:

$$\text{Driver Power (HP)} = \frac{H.Q.SG}{3960.E}$$

where: H = Pump Head (ft)
Q = Flow rate (U.S. gpm)
SG = Specific Gravity of Fluid
E = Pump Hydraulic Efficiency

or

$$\text{Driver Power (kw)} = \frac{9.81\ (H.Q.SG)}{3600.E}$$

where:
H = Pump head (meters)
Q = Flowrate (M^3/hour)
SG = Specific gravity of fluid
E = Pump hydraulic efficiency

The effect of specific gravity on driver horsepower is shown in Figure 15.

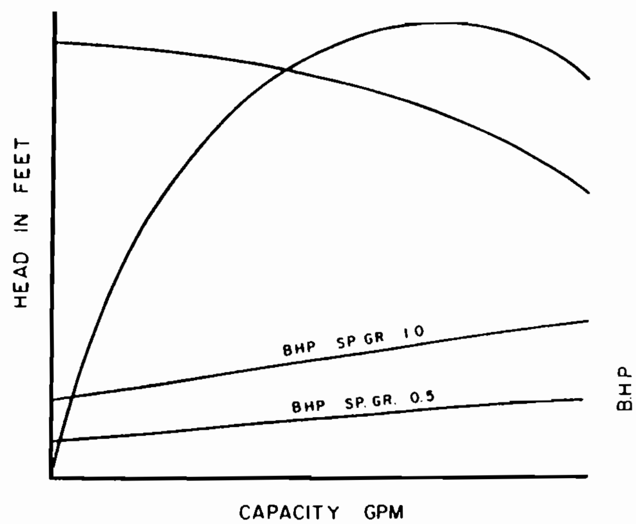


Figure 15. Effect of Specific Gravity on Power Requirements.

PARALLEL AND SERIES PUMP OPERATING CONCERNS

When a pump is called upon to operate in *parallel or series* with another pump, additional care must be taken in defining each pump's boundary conditions. The effect of two identical pumps operating in parallel or series is illustrated in Figure 16 [3].

Parallel operation requires that the minimum stable flowrate of all pumps, that are operating in parallel (two or more), be satisfied. Where pumps operating in parallel are not identical, the difference in shutoff heads may result in one pump being hydraulically shut off at a low flowrate within the operating range. A similar problem

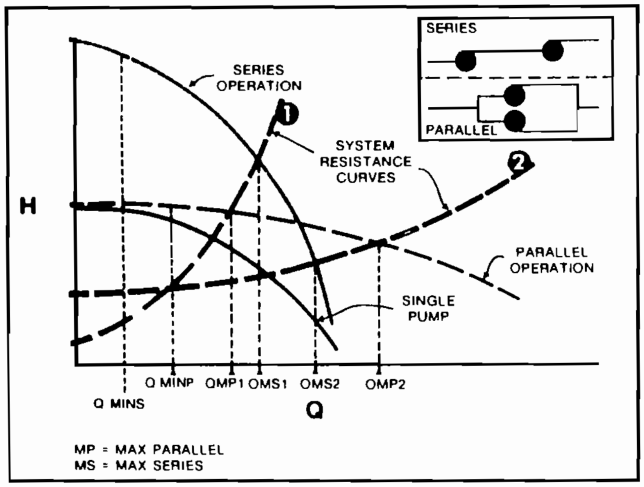


Figure 16. Pump Performance Curves for Parallel and Series Flow Operation, Showing Effect of System Resistance.

may occur at even higher flowrates where the head/capacity characteristic is very flat and shutoff heads differ.

While lower flow operation may not result in hydraulically shutting off one pump, it may result in one pump operating below its minimum stable flow point.

API 610 (7th edition) calls for one-stage and two-stage pumps operating in parallel to have head rises of 10 to 20 percent of the head at rated capacity. This will protect identical pumps, but may endanger different sized pumps operating in parallel whose shutoff heads can differ while complying with this requirement. While the head rise requirement must be complied with, the agreement of parallel pump shutoff heads is equally important.

Pumps of the same size (same rated point) operating in parallel may have a minor difference in shutoff head, within the 10 to 20 percent head rise criterion, as shown in Figure 17. This may require the pump of the higher shutoff head (pump 2) to deliver close to its rated flowrate before the continuous minimum stable flowrate of pump 1 is satisfied. The result is a much higher combined continuous minimum stable flowrate (Q_{min}CP in Figure 17 [4].

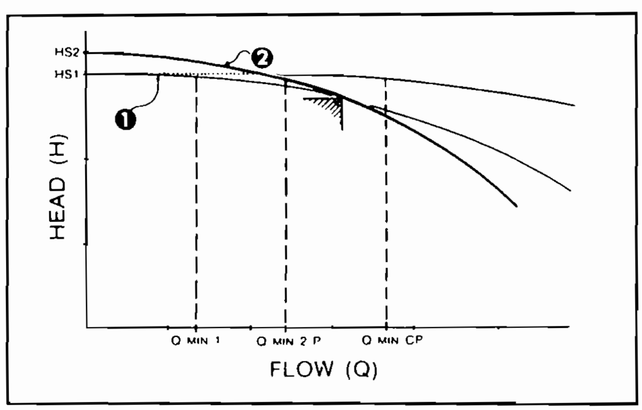


Figure 17. Parallel Operation of Same Size Pumps.

Parallel operation of different size pumps (same rated head, different rated flows) may also have minor differences in shutoff head within the API 610 guidelines, as shown in Figure 18. This will also result in a high parallel continuous minimum stable flowrate (Q_{min}CP in Figure 18) to satisfy the continuous minimum stable flowrate of the smaller pump, pump 1.

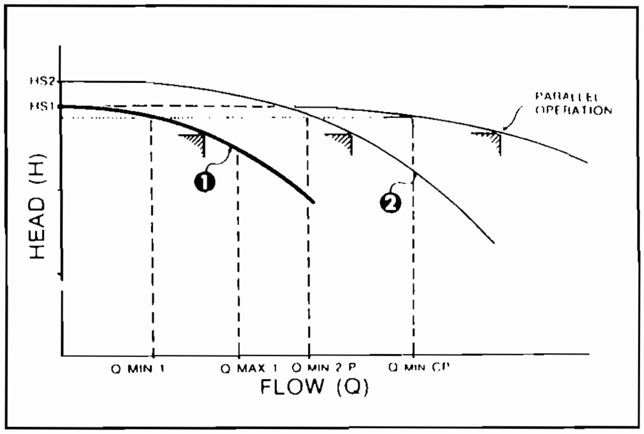


Figure 18. Parallel Operation of Different Size Pumps.

Pump 2 in Figure 18, would be required to establish a flowrate of Qmin 2P before pump 1 can be satisfactorily operated. A decision to operate pump 1 alone during periods of low flow requirement to gain the advantage of its lower continuous minimum stable flowrate, would only be possible to a flowrate of Qmax 1 (Figure 18) [4]. At flowrates above this it would be necessary to switch to pump 2 alone, as parallel operation above Q_{max} 1 requires a step rise in flowrate to avoid hydraulic shutoff of pump 1.

The threat of premature or frequent pump failure and/or the inconvenience of proper parallel operation under either of the two examples shown in Figures 17 and 18 are easily avoided by paying close attention to the agreement in shutoff heads, as well as rated point, in pumps required to operate in parallel. As a general guideline, a five percent maximum difference in shutoff heads, with a common rated point, will result in satisfactory parallel operation over a wide range of flowrates. For best results, however, a study of system flow requirements and the pumps' curve form will determine more precisely the acceptable variations in shutoff heads for two or more pumps operating in parallel.

As a general rule, *parallel operation* of centrifugal pumps *to increase flow* is most beneficial where the system resistance curve is relatively flat (or shallow) with respect to flow.

Series operation by nature, enforces an identical flowrate through each pump where the discharge head of the pair (or more) is the sum of the heads developed by each pump. On occasion, however, side stream flows may break this rule with the upstream pump delivering more flow than the downstream pump. The presence of resistive components and side streams has a major impact on setting pump boundary conditions where series operation is required. The split in head between the two pumps must reflect each pump's system resistance. Pressure limitations on system components such as heat exchangers may limit the maximum permissible pressure (or head) at a pump's discharge and may demand an uneven split in the pump's total head (or differential pressure). Series operation may also require the specification of a high pressure casing on the downstream pump, that may also require loss of flow protection in the event that it is unable to maintain a minimum flow if the upstream pump fails to deliver sufficient supply pressure.

As a general rule, *series operation* of pumps to *increase flow* is more beneficial than parallel operation where the system resistance curve is steep with respect to flow. Head/flow characteristics for simple series and parallel applications are shown in Figure 16.

Reliable operation, continuous satisfactory performance, and low maintenance costs are only possible when such flow considerations are reviewed in a team framework at the system analysis stage.

PUMP SPECIFICATION—KEY ELEMENTS

Pump specification requires that the vendors (or prospective bidders) be informed of which requirements must be fulfilled and which options they have in certain areas. A list of "musts" is provided and these are defined in a clearly displayed pump specification sheet—typically the API 610 standard centrifugal pump data sheet. Optional areas may be left blank, or a range of acceptable alternatives listed separately, to avoid unknowingly penalizing a particular vendor for quoting an unacceptable item.

Narrative statements should accompany the centrifugal pump data sheet, to qualify in more detail those areas of importance that are only briefly described in the data sheet. A separate sheet for the mechanical seal specification is strongly recommended. The API 610, 7th Edition includes a pump seal data sheet that may be used for this purpose.

There are a number of prime areas of importance in specifying a centrifugal pump and the preparatory work done on system analysis will enable many of these areas to be defined confidently. These prime areas include:

- Liquid specification.
- Operating and site conditions.
- Performance.
- Construction.
- Mechanical seals.
- Auxiliary piping.
- Lubrication and bearings.
- Inspection and test.
- Vertical pump details.
- Weights.
- Additional information.

The discussion will be limited to the first three areas; *liquid specification, operating and site conditions, and performance.*

Liquid Specification

In addition to the parameters outlined on the API 610 data sheet, comment must be made on solids *content, toxicity, and setup temperature*. These latter three qualities of a liquid will play a large part in determining mechanical seal selection and auxiliary piping requirements as will many of the other liquid specifications.

It may be necessary to include an additional comment in the narrative statement to fully define special qualities of the liquid.

Operating and Site Conditions

The capacity is now defined to represent normal and maximum (or rated) conditions. Minimum expected continuous flowrate must also be included here. By defining these three flowrates, maximum, normal and minimum, vendor constraints are imposed, which must be considered in light of the other hydraulic specifications. Remember to include a table showing the percentage of time the pump is expected to run at each of these three flows.

Suction pressure, maximum, rated and, in particular, the minimum that may be experienced, will be given very serious consideration by the vendor when considering capacity requirements. (Excessive drop in pressure at the impeller eye at high flowrates forces designers to increase impeller eye diameter, or consider a pump impeller design with a better blade cavitation factor, to accommodate low $NPSH_A$.)

The maximum *discharge pressure* that will be encountered under conditions of maximum flow and minimum suction pressure will heavily influence the size and type of pump that a vendor must offer and may limit the choice.

It is necessary to be realistic in writing the pump specification. After full system analysis, the boundary conditions (fluid conditions at the pump suction and discharge flanges) and flow requirements may preclude a vertical inline pump, even though the plot space calls for such a pump to fit a limited space. A low flow, high head requirement may not fall within the range of a conventional centrifugal pump and may require a high speed, two (or multiple) stage or series pump operation.

Completed fully, the previously conducted system review will have considered the operating flexibility and space requirements of various pump designs in defining boundary conditions and flow requirements. The optimization performed under the system analysis will result in clear and easily definable pump hydraulic parameters.

Site conditions will influence items such as electrical or steam tracing requirements, lubricant quality, type of lubrication, motor protection, etc.

Performance

This relatively small area of the data sheet is of prime importance when bids are reviewed. The vendors (bidders) have an opportunity here to convey much of the important performance variables that will affect the selection. Strangely, this section is often partially neglected by bidders or the information submitted is erroneous, or too subjective. The critical parameters of minimum acceptable continuous flowrate and suction specific speed are often neglected or treated lightly. It is necessary to reinforce the requests for these details by being more descriptive of the performance needs in the section on *Operating Conditions*. In particular, where calculated values of $NPSH_R$ are considered to be more important than the normal test values, they must be specifically requested. Some debate is continuing on the correct formulae for calculating $NPSH_R$ values, and it may be some time before such a request can be considered standard.

REFERENCES

1. Shiels, S., "Hidden Dangers in Centrifugal Pump Specification," World Pumps (Jan/Feb 1995).
2. Ross, R., "Theoretical Predication of Net Positive Suction Head Required ($NPSH_R$) for Cavitation Free Operation of Centrifugal Pumps," United Centrifugal Pumps publication, San Jose, California.
3. Schiavello, B., "Cavitation and Recirculation Field Problems," *Proceedings of the Ninth International Pump Users Symposium*, Turbomachinery Laboratory, Texas A&M University, College Station, Texas (1992).
4. Shiels, S., "Centrifugal Pump Specification & Selection—A System's Approach," *Proceedings of the Fifth International Pump Users Symposium*, Turbomachinery Laboratory, Texas A&M University, College Station, Texas (1988).
5. Fraser, W. H., "Flow Recirculation in Centrifugal Pumps," *Proceedings of the Tenth Turbomachinery Symposium*, Turbomachinery Laboratory, Texas A&M University, College Station, Texas (1981).
6. Gopalakrishnan, S., "Minimum Flow Criteria," Pacific Energy Association Meeting, Irvine, California (1986).
7. Lobanoff, V. S., and Ross, R. R., *Centrifugal Pumps—Design and Application*, Second Edition, Houston, Texas: Gulf Publishing Company (1992).
8. Bolleter, U., and Frei, A., "Shaft Sizing for Multistage Pumps," *Proceedings of the Tenth International Pump Users Symposium*, Turbomachinery Laboratory, Texas A&M University, College Station, Texas (1993).