Relating Centrifugal Pump Efficiency, Variable Speed Drives and Hydraulic Improvements to Energy Dollars

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Engineers, Operators, and Maintenance specialists all view their pump systems from different perspectives. The Engineer has traditionally been focused on pump equipment materials and sizing based on the most severe conditions that could be encountered at a facility. The Operator is typically interested in equipment flexibility, capacity and reliability. The Maintenance specialist concentrates on servicing the equipment, ensuring spare parts are readily available, proper installation and refurbishment, and of course pump system reliability. Even though many of us are focused on one segment of pump systems, we have all picked up ideas and practical knowledge from each of these areas.

The continuous pressure to reduce costs with less resources has forced many of us to learn more about pump system mechanical and electrical system efficiencies, and how to evaluate our pump systems to ensure we have done everything possible to reduce operational costs. The prime component of these operational costs is often the cost of energy.

This paper discusses the performance characteristics of the basic components of pumping systems. It also relates some factors that affect the components individually as well as how the individual component efficiency is interrelated with the other components.

Evaluating a pump system to increase system efficiency and reduce energy costs, can be segmented into the following major categories:

- ***Energy Costs and Utility Rate Structures***
- ***Hydraulic System Efficiency***
- ***Mechanical Efficiency***
- ***Electrical Efficiency***

Before even beginning the process of evaluating each of the above categories, the most important step is often to get the “big picture” on your pumping operation. Some of these questions may include:

Is a plant expansion or major process change expected soon? If this change is three years down the road it may still be worth making operational changes that can be implemented in the short term.

Can the process be changed to eliminate the need for pumping or reduce the flow rate or head? (In some cases a piping change will allow gravity flow for a process where a pump was previously specified)

Do the system flow rate/head requirements change with time? Are the changes in flow requirements continuously variable and spread over a relatively broad range, or at a few discrete flow rates?

It may seem like the above questions are obvious, However, it is surprising how often these issues are overlooked. One approach that is often used in characterizing system requirements is the development of a flow rate requirement[1] vs. time curve, sometimes called a flow duration curve. An example of duration curves for three different systems is shown in Figure 1. The total flow required (i.e., the area under each curve) is the same for the three systems, but the distribution is obviously quite different. This type of curve is a good starting point for overall system analysis. Further consideration, such as the relationship between system flow and head requirements is needed to fully understand system requirements and optimal design. But even a casual review of the three duration curves shown in Figure 1 reveals that the

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[1] It is important to make the distinction between system *requirements* and actual operation, in the case of pre-existing systems unless there is confidence that the two quantities are the same.

kind of pumping configuration that is suitable for System A (a single-size pump, selected to operate efficiently at the constant flow requirement) would be less than optimal for either System B or C.

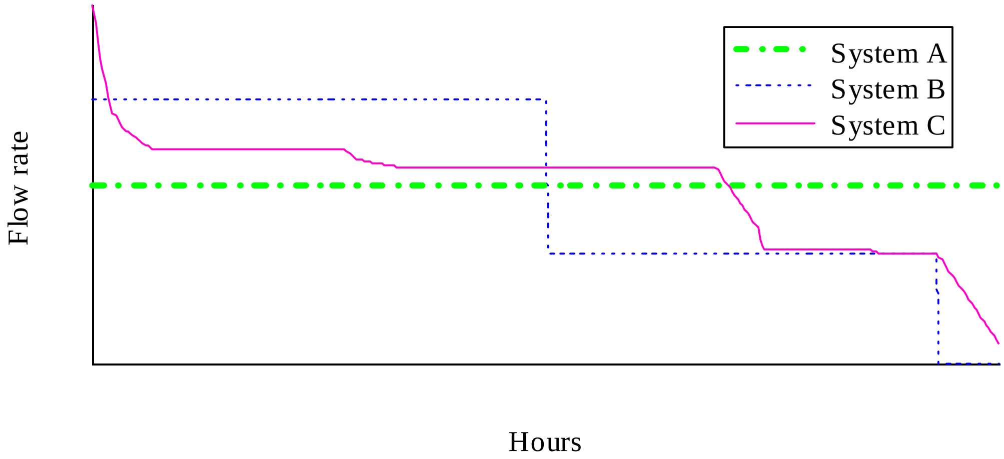


Figure 1. Example flow duration curves

Energy Costs and Utility Rate Structures

One of the first areas that deserves investigation before focusing on the efficiency details of a pump system is to understand energy rate schedules and how the cost of energy is determined at a facility.

Although many industries produce their own energy at very reasonable rates, most facilities purchase their energy from the local electric utilities. Typical utility energy costs include a consumption charge (kWh) and a demand charge (kW or kVA), and in some cases, a power factor penalty. Energy consumption charges range from 3 to 6 cents per kWh in the South and Midwest to 8 to 12 cents per kWh in the Northeast. Energy demand charges range from $4.00 up to $25.00 per kW of demand. For a 100-hp water pumping station that operates continuously, a sample monthly energy cost calculation may be as follows:

Element	Monthly value	Per unit cost	Cost ($)
Service charge	Fixed fee	$35/month	35
Energy use	57,600 kWh	$0.08/kWh	4,608
Peak demand	90 kW	$10/kW	900
Total			**5,543**

Note that there are other costs that are sometimes involved, such as fuel charges and power factor penalties that are not included here. As indicated in this example, demand cost can be 20% of the total energy bill and should always be considered when calculating savings.

An important consideration in energy *cost* reduction is the option of applying Time-of-Use Electric Rates. Although the example in Table 1 shows a constant consumption rate of $0.080/kWh, some electric utilities reduce the cost of energy during off peak hours (typically evenings and weekends). Municipal water system that have been designed with adequate storage and efficient use of “pressure zones” can often pump only at night to replenish water storage tanks and rely on water storage capacity and elevation pressure to provide water service during the day. It is important to distinguish between overall energy consumption and energy costs here, since pumping at much higher capacity in off-peak hours (and consequently at lower capacity during on-peak hours) may actually require more energy consumption than if the system flow rate was maintained relatively constant.

Hydraulic System Efficiency

An important aspect of evaluating pump systems is to understand how the system is operated. This includes reviewing the system surrounding the pump rather than just the pump and driver. Some of these considerations include:

- What flow rate and pressure are *required* for the process? Is more of either being provided than is necessary?
- How is the pump being controlled? Can level or pressure setpoints be adjusted to higher or lower values?
- Are there restrictions in the piping system (e.g., throttled valves, pressure reducing valves, unneeded or oversized check valves, excessive elbows, corrosion and scale build-up in the piping system)? In a similar vein, have the system requirements changed significantly so that the existing pipe velocity is well beyond what was originally anticipated?
- Are multiple pumps operating in series or parallel being operated efficiently with control systems?

Answers to the above questions often provide the most cost-effective improvements that can reduce pumping energy costs by simple operational adjustments or piping improvements.

It is particularly important to first search for savings in the hydraulic system, since savings achieved at the system level are amplified at the pocketbook level. This is because the pump and motor are not 100% efficient; so for every horsepower reduction in fluid power requirements, there is about a 1.4-hp reduction in electrical power requirements, even for systems in which the pump and motor are operating efficiently.

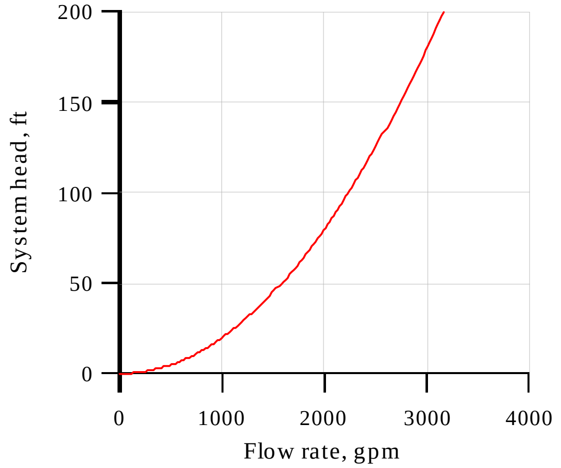
It is useful to develop system head-capacity curve(s) for the most common system alignment condition(s). This curve is useful for several reasons:

1) It, in conjunction with the pump head-capacity curve, defines where the system will operate. In order to gauge the effects of system or pump changes on overall performance, such a curve is essential.
2) It provides a baseline against which to compare future system performance. As the system ages and when components are replaced, the baseline curve can be used as a reference value from which potential savings can be estimated.
3) It helps identify situations where variable speed drives may be most effective (systems that are dominated by frictional head) or least effective (systems that are dominated by static head).

To help illustrate the merits of developing and trending system performance curves, an example system head-capacity curve (with no static head) is shown in Figure 2. The same curve, along with a second curve representing the same system after scale buildup in the piping is shown in Figure 3. To maintain a constant flow rate of 2500 gpm, an increase of system head of about 31 feet must be overcome. The increased head of 31 feet corresponds to an increase in hydraulic power (assumed fluid specific gravity = 1.0) of almost 20 hp, which is about 25% more than was required in the new, clean system.

Assuming a constant combined pump and motor efficiency of 70%, the frictional losses would translate into an electrical power requirement of about 21 kWe. If the system was operated continuously at this condition, and the per unit energy cost of electricity was $0.08/kWh, the annual cost of the increased friction would be almost $15,000.

Changes in the system curve affect the pump operating point (which in turn, affects the motor operating load), so the pump and motor efficiency would not, in a real-world application, remain fixed. But this simplified example helps to illustrate the merits of understanding and trending system performance.



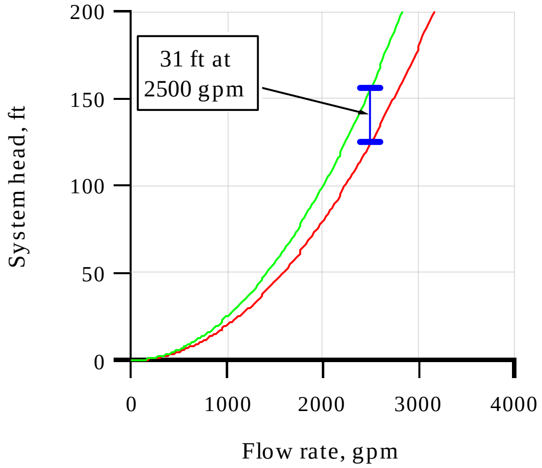


Figure 2. Example system head-capacity curve

Figure 3. Change in system curve with increased friction

Pump Efficiency

After developing an understanding of system operation and the operational cost components of your pump system, the mechanical/hydraulic efficiency of the pump should be investigated. One of the first places to start is to obtain a copy of manufacturers pump curve. Possible sources of pump curves are:

1) a test facility certified performance curve for the specific pump in question,
2) an in-situ performance curve (if good quality instrumentation is used, this is a preferred method, since it captures the actual motor and pump characteristics),
3) generic performance curve from the manufacturer's catalog, or
4) software packages that include manufacturer curves.

The performance curves will provide a graphical understanding of the relationship of flow rate, head, efficiency, and shaft input power of the pump. In the case where an in-situ performance curve is generated, the electrical input power to the motor replaces the shaft power. An example set of performance curves for a pump that might be used in the system shown in Figures 2 and 3 is shown in Figure 4.

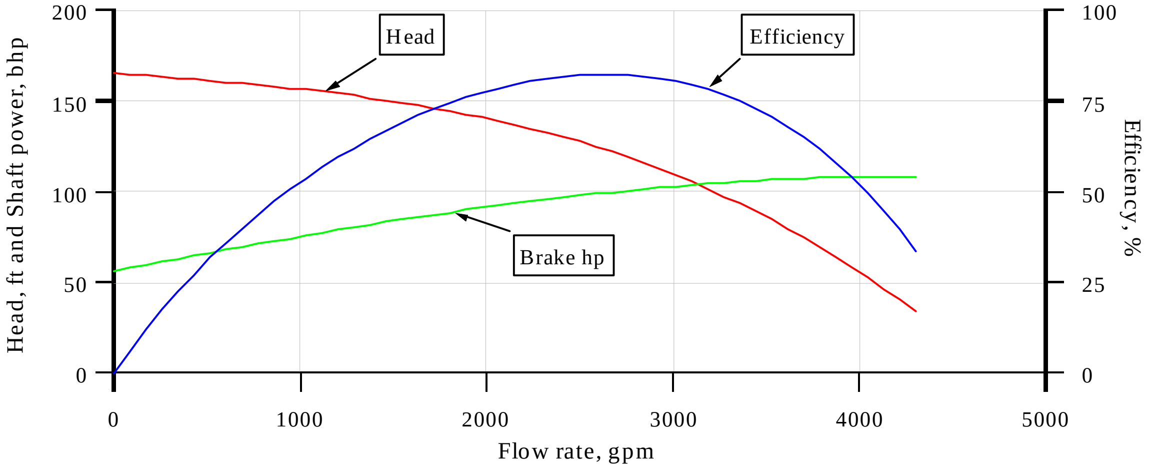


Figure 4. Example pump performance curves

A couple of key questions to be answered when considering pump performance curves are:

1) Is the pump operating reasonably close to its best efficiency point (BEP), at each required operational head and flow condition?
2) Has pump or system performance changed over time (for situations where a baseline set of pump performance curves are developed and compared against subsequent performance data)?

Once both the pump and system performance curves are available, the system and pump curves can be jointly plotted to identify the operating point(s). This exercise is especially important in answering the second question above.

Figure 5 shows the head-capacity curves for the system(s) and pump from Figures 3 and 4. The shift in flow rate *and* head that occur due to the increased frictional losses in the system are annotated. Figure 6 demonstrates possible pump performance degradation in a fixed system. The key point to be made is that baseline measurements of both pump and system performance characteristics provide the reference value against which subsequent measurements can be evaluated and thereby help in the identification of corrective actions.

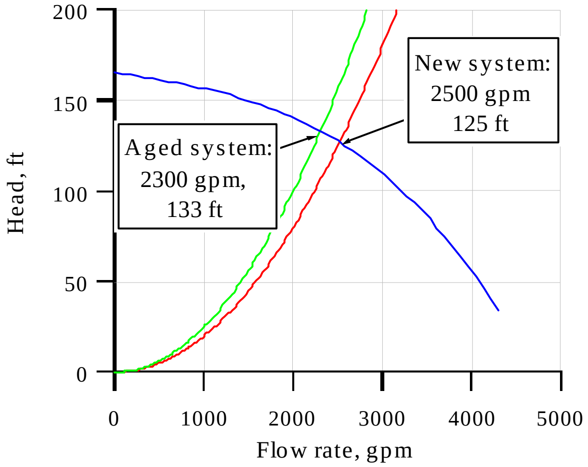


Figure 5. Fixed pump with system degradation

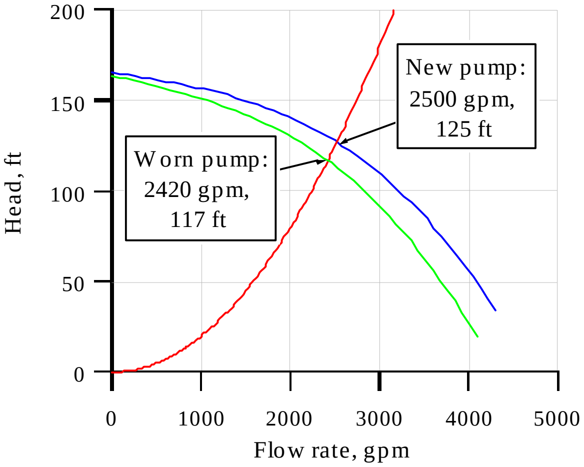


Figure 6. Fixed system with pump degradation

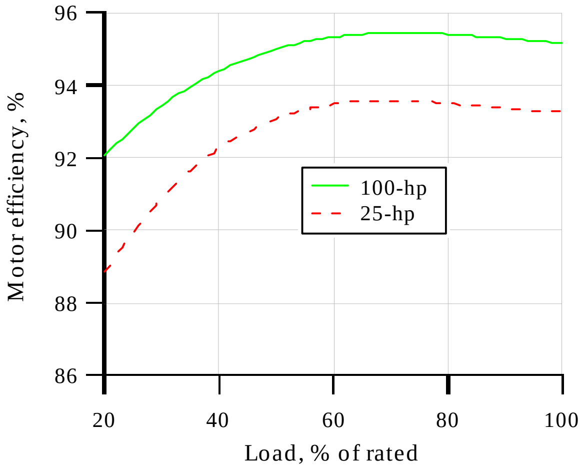
In cases where multiple pumps are operating in parallel or in series, graph paper or a pump-modeling program is useful for developing composite pump curves.

If the existing pump is operating well away from its BEP flow rate, there is prima facie evidence that efficiency and operational cost reduction opportunities exist. The suitability of alternative courses of action depends on a variety of factors, including:
1) the range of flow requirements, and whether the flow varies continuously or in discrete steps;
2) the distribution of the system static and frictional head components;
3) the distribution of time operating at the various flow rates;
4) whether there is evidence that the pump is degraded.

Electrical Efficiency

The electrical efficiency of a pump system is, in essence, the motor output mechanical shaft power divided by the power measured at the utility meter. It includes the electric motor, switchgear, and supply leads, and where applicable, variable speed drive, filters, and transformers. Overall, the efficiency of these electrical components is usually very high. While there may be opportunities to gain a few efficiency points in the electrical components themselves, the primary improvements usually achieved in the electrical area are to improve the efficiency – or, perhaps more importantly reduce the power requirements of either the pump or the fluid system.

Typical motor performance efficiency vs. load and power factor vs. load curves for two motor sizes are shown in Figures 7 and 8. The curves shown are for four-pole, energy efficient ac-induction motors.



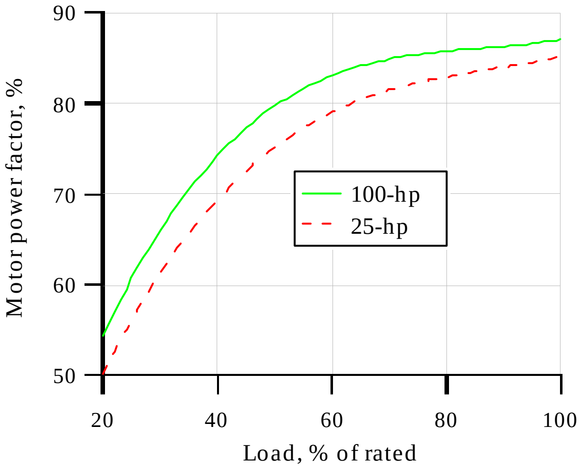


Figure 7. Typical motor efficiencies vs. load **Figure 8. Typical motor power factors vs. load**

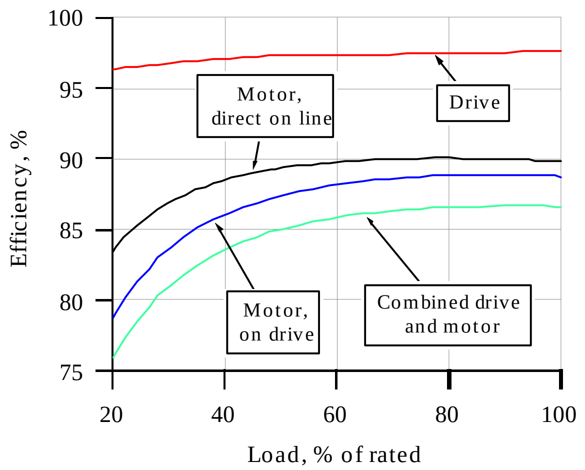
An important point to recognize from Figure 7 is that efficiency for induction motors is essentially constant over the normal load range. It is only if the motor is extremely lightly loaded that sizing is an issue. In fact, if a power factor penalty is part of the rate structure (either explicitly or implicitly, based on historical experience), the low power factor at light load might be a more significant cost factor than the slight reduction in efficiency.

When an adjustable speed drive is used, the motor speed can be reduced to accommodate the system flow requirements with reduced energy consumption over these alternative means:

1) Not controlling flow at all that is, running the system at a higher flow rate than is necessary to meet the system requirements,
2) Controlling flow in a batch mode that is, starting and stopping the pump for limiting conditions, such as when filling or depleting a tank or reservoir,
3) Controlling flow by valve throttling, or
4) Controlling flow rate by recirculating a portion of the flow.

Of course, like any other active component, the drive efficiency is not 100%. Figure 9 shows motor, drive, and combined efficiency for a modern pulse width modulated drive operated at rated speed conditions (on a two-pole, 50-hp motor). The motor efficiency alone, when driven directly from the power supply (i.e., without the drive) is also shown for comparison. As can be seen, the drive efficiency is in the upper 90-percent range. The drive also causes the motor to operate at a slightly lower efficiency than when the motor is driven directly across the line.

Figure 10 shows the same motor efficiency curve as Figure 9, but for the combined drive and motor curve, uses data that represents centrifugal loads (i.e., output power is proportional to the speed cubed).



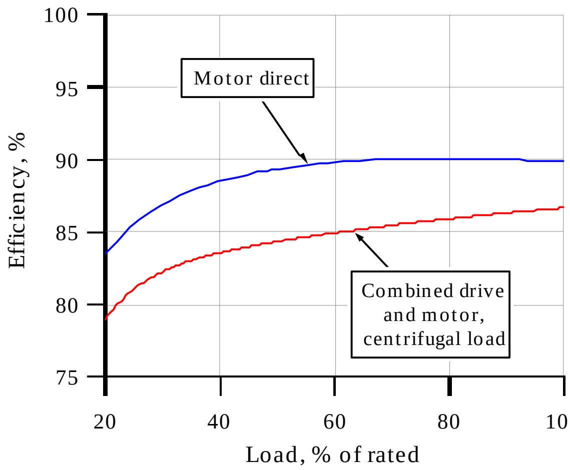


Figure 9. Motor and adjustable speed drive efficiencies at rated speed

Figure 10. Motor (rated speed) and adjustable speed drive (centrifugal-type load speeds) efficiencies

But simply considering efficiency alone doesn't capture the essence of variable speed drives. The real saving derives from the drop in power that accompanies speed reduction. For centrifugal loads, such as pumps, the shaft power is approximately proportional to the cube of the speed. Figure 11 contrasts the difference in shaft power between fixed and variable speed driven when the pump whose performance curves are shown in Figure 4 is used in the system whose head-capacity curve is shown in Figure 2. The shaft power for the variable speed case was calculated using the pump affinity laws (discussed below).

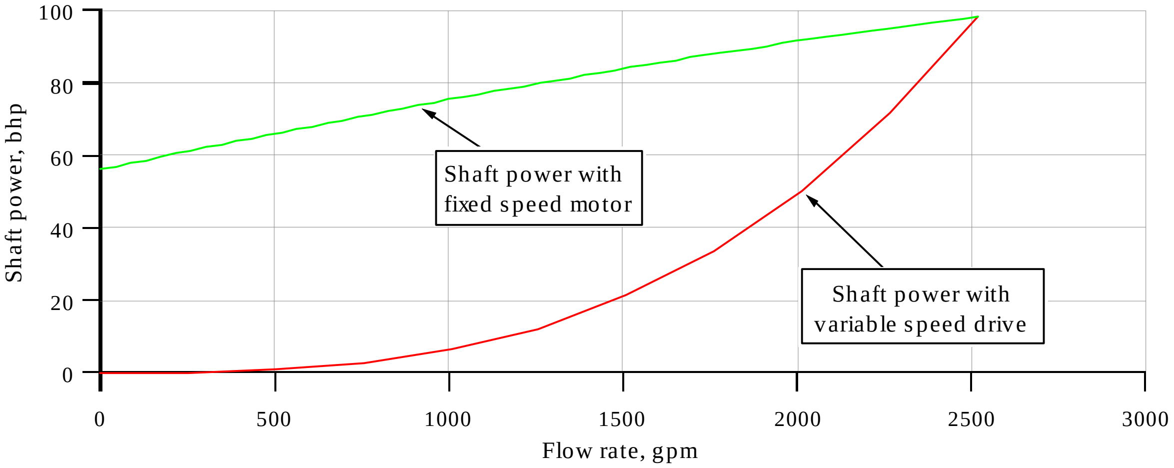


Figure 11. Shaft power requirements for a fixed and variable speed driven pump

Overall Pumping System Calculations and Field Measurement Considerations

Pumping system electric power and energy consumption can be calculated as follows:

$$P = \frac{Q\,H\,\gamma}{5310\,\eta_p\,\eta_m\,\eta_s} \quad \text{and} \quad E = \frac{Q\,H\,\gamma\,T}{5310\,\eta_p\,\eta_m\,\eta_s}$$

Component	**Abbreviation**	**Units**
Power	P	kW
Energy	E	kWh
System flow rate	Q	gpm
System head	H	ft
Fluid specific gravity	γ	Dimensionless
Run time	T	hours
Pump efficiency	η_p	Dimensionless fraction
Motor efficiency	η_m	Dimensionless fraction
Electrical supply efficiency	η_s	Dimensionless fraction

Since it is none of the efficiency values can be practically measured in the field, the efficiency terms can be combined to a single overall value, η_T, resulting in the following relationships:

$$P = \frac{Q\,H\,\gamma}{5310\,\eta_T} \qquad \text{and} \qquad E = \frac{Q\,H\,\gamma\,T}{5310\,\eta_T}$$

Some notes regarding the field measurement of these parameters:

1. In many cases (for example, where ambient temperature water is the pumped fluid), the specific gravity can be assumed constant; but it is important to note that other situations, such as where variable consistency slurries are being pumped, the specific gravity should be measured.
2. By definition, the head and flow rate for the system and the pump are equal, since the intersection of the pump and system head-capacity curves define the operating points for both. Measurement of the pump head requires that suction and discharge pressure measurements be made (and gages referenced to a common elevation). True pump head measurement also requires an accounting of differences in suction and discharge velocity heads, but as a general rule in field measurements, the differential velocity head can be disregarded because it is such a small component of head.
3. The two parameters that are usually most readily measured are pump head and electrical input power. If an *accurate* pump performance curve is available, the head can be used to graphically determine flow rate. It is important to emphasize the word *accurate* here, since it is not unusual for pump performance to deviate from generic manufacturer curves; furthermore, it is all too commonly found that the user doesn’t possess even a generic performance curve.
4. It is important to measure rotating speed (e.g., with a portable strobe light). Pump performance curves (even certified curves) are usually performed at a particular speed. The test speed may be one or two percent different from the actual speed. This has a significant impact, and the performance curve needs to be adjusted for actual speed conditions (see discussion on pump affinity laws below).
5. When the only two parameters that can be measured are head and motor power, and a generic pump performance curve is available, there is a great deal of value to using these two measurements to do a confirmatory cross check. For example, use the measured pump head and the head vs. capacity curve to estimate the flow rate. Next, estimate electrical train efficiency. Unless a variable speed drive is being used, the motor is the only component that usually needs to be considered. On newer motors, the motor nameplate will include a NEMA nominal and/or guaranteed motor efficiency. Multiply the measured motor input power by the estimated motor efficiency to estimate shaft power. Then compare this value to that derived for the estimated flow rate the power vs. capacity curve. If there is a significant difference, further consideration is warranted. Some possible reasons are:
    a. Particularly if the measured power suggests that the pump is not as efficient as the curve indicates (i.e., if the measured power is greater than the curve-based power at the estimated flow rate), it may be safe to assume that the pump is simply not operating at the manufacturer curve

efficiency. This could be due to poor pipe layout (and resultant fluid geometry) in the field; it could also be the result of pump wear.

b. If the reference curve is simply a generic performance curve for the particular pump model, it is important to recognize that there may be significant variation in performance from pump to pump.

c. The head or power measurements (or estimated motor efficiency) were erroneous. If permanently installed pressure gages were used for the head determination, their accuracy should be verified. In the authors' experience, field gages often go for years without being calibrated or even checked. If at all possible, temporary test gages known to be in calibration should be used if at all possible.

It is usually necessary to apply the pump affinity laws to account for differences in actual operating speed and the speed at which the pump curves were developed. The pump affinity laws are as follows:

$$Q_2 = Q_1 * \left(\frac{N_2}{N_1}\right) \qquad H_2 = H_1 * \left(\frac{N_2}{N_1}\right)^2 \qquad P_2 = P_1 * \left(\frac{N_2}{N_1}\right)^3$$

where Q = flow rate, N = rotational speed, H = head, and P = power. The subscripts 1 and 2 represent two different speeds.

Another form of the affinity laws relates to impeller diameter, D:

$$Q_2 = Q_1 * \left(\frac{D_2}{D_1}\right) \qquad H_2 = H_1 * \left(\frac{D_2}{D_1}\right)^2 \qquad P_2 = P_1 * \left(\frac{D_2}{D_1}\right)^3$$

The subscripts 1 and 2 represent two different impeller diameters. The impeller diameter affinity scaling relationships have proven useful in some field-based measurement experiences in that the impeller diameter used as the basis for the performance curves can be modified iteratively to a point where the curve-based flow estimates from measured head and power are in agreement.

Opportunities for Improving Pump System Efficiency

After a preliminary review of the pump system as been performed, some quick calculations can be performed to identify the magnitude of savings that each opportunity may provide prior to more extensive field testing. Some examples of preliminary calculations may include:

> *Using the pump equation shown above and the manufacturers pump curve to identify the impact of lowering the TDH on a system by raising the level in the pump suction (e.g., raising the average level of the wetwell in a waste water application), reducing pressure on the discharge side of the pump, or decreasing the piping system head losses.*
>
> *Using the affinity laws to determine how a reduction in flow by shaving the pump impeller size or reducing the speed of a variable speed drive will effect energy use.*

Performing some preliminary estimates of savings is useful prior to a detailed evaluation to understand what kind of testing should be performed.

As each pump system is reviewed, several potential improvements may provide savings in one area and reduce efficiency in other areas. One of the most common improvements that this occurs is when variable speed drives are applied.

The primary advantages of AC variable speed drives (also called VSDs, VFDs or ASDs) is to control the pump flow rate by adjusting pump motor speed. Figure 12 illustrates the change in pump head-capacity curves as motor speed is changed (using the affinity laws).

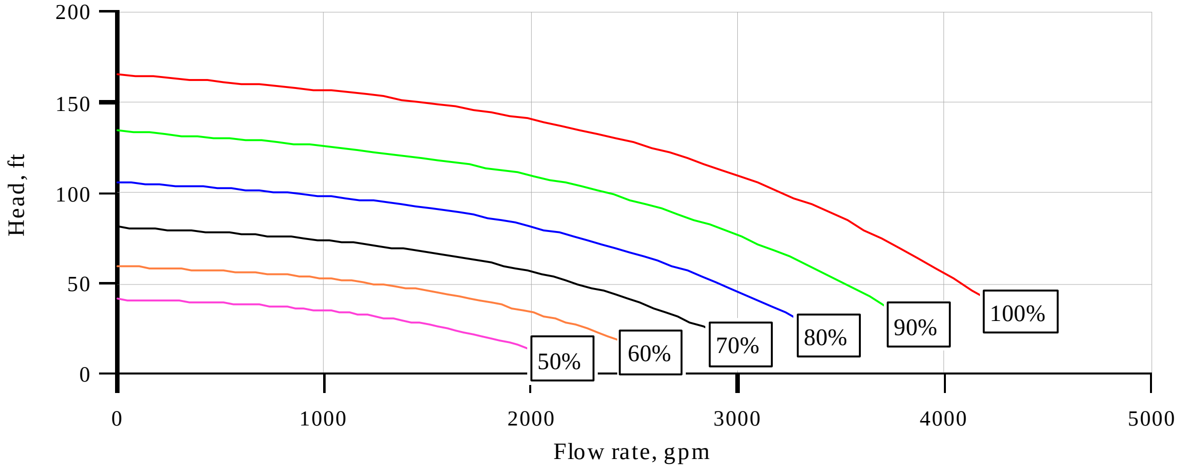


Figure 12. Head-capacity curves at varying speeds

It is important to understand, however, that the change in actual flow rate doesn't necessarily follow the affinity laws in fact it only follows the affinity law scaling for systems with no static head. This can be illustrated by overlaying two system curves on the series of pump curves, as shown in Figure 13. For the system with 100 ft of static head, reducing motor speed to 80% of nominal causes a reduction in flow rate to about 25% of that for full speed. This is in stark contrast to the all frictional (no static head) system, for which operation at 80% speed results in the flow rate dropping to 80% of that at full speed.

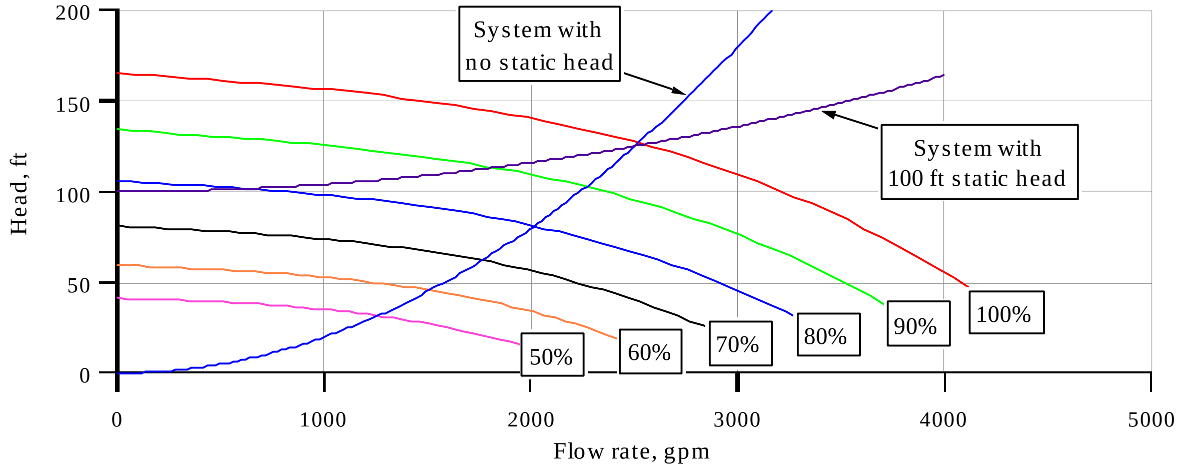


Figure 13. System and pump head-capacity curves at varying speeds

Several factors that should be weighed when considering application of drives are:

1) A fundamental requirement for considering variable speed drives should be that flow (or in a more limited sense, head) requirements change over time. If the system flow and head requirements are essentially fixed, efforts should concentrate on ensuring a properly sized pump (which should also be a consideration, regardless of whether a drive is employed).
2) The greater the proportion of the system head that is static, the less effective will be the variable speed drive. This is particularly true if the flow requirements vary significantly. Parallel pumps are, on the other hand, most effective in systems that are dominated by static head. Parallel pumps need not all be identical; in fact there are some systems, such as waste treatment plant applications, where

a small pump could meet system requirements 80-90% of the time, and supplemented by a parallel, larger pump for the occasional higher flow condition.

3) The pumps must operate enough hours during the year to justify applying VSDs (when based on a simple payback, a pump station with multiple pumps may only have the lead pump equipped with a VSD).
4) In situations where there is significant static head, the effective range for a variable speed driven pump can be increased by selecting the pump such that the pump operating point is to the right of (i.e., greater than) the BEP when operating at the highest anticipated flow requirement. This allows the pump efficiency to initially increase as speed is reduced.
5) The motor and electrical system should be verified to be suitable for drive applications. This is particularly important if a VSD is being added as a retrofit.
6) If a variable speed drive is being retrofitted to systems with switch-based controls, such as a float-type level control system that causes the pump to be started or stopped at distinct levels, continuously variable instrumentation, such as a level indicator, will need to be added for control purposes.
7) In a similar vein, systems that currently employ control valves with automatic valve positioners can usually be more easily retrofitted by using the existing valve positioning signal source as the controlling signal. It might be noted that valves that are normally significantly throttled are a good first indicator of a system for which a drive may be effective.

Evaluating Pump Systems

Evaluating pump systems begins with an understanding of how the pump is operated. Often it is useful to assemble a table with several flow intervals that can be used to determine how often the pump operates at various flows. Several data collection or estimating options for each aspect of operation are shown in Table 1.

Table 1. Measurement considerations

	Method	Notes
Flow rate	Estimate flow rate from measured head and/or power	- Depends on accurate pump performance curve; - For many pumps, shaft power varies relatively little with flow rate; - Motor efficiency must be assumed.
	Measure, using permanently installed flow meter	- Usually preferred; calibration/accuracy must be verified
	Measure, using portable flow meters, such as ultrasonic	- Ensure that a proper flow profile exists (suitable length, particularly downstream, of elbows or other flow disturbances)
	Calculate, using thermodynamic flow meter	- Accuracy Reduced for Low Head Pump Systems
	Measure, using pump down test	- Head changes must be accounted for.

	Method	Notes
Head	Estimate head from of measured flow and/or power	- Depends on accurate pump performance curve; - For many pumps, shaft power varies relatively little with flow rate; - Motor efficiency must be assumed.
	Estimate pressure loss from components, pipe data	- Not verified with field data; large uncertainties, particularly with older systems, unless most of the head is static
	Measure using suction and discharge taps	- Gage accuracy must be verified. Use of portable test gages is recommended. - If significant loss components are between the pump and the pressure gage, an estimate of the associated head loss must be made.

	Method	Notes
Efficiencies	Estimate motor efficiency from nameplate, manufacturer data, or software sources	- Although there may be some inaccuracy, the errors will generally be small relative to other error sources
	Estimate electrical system efficiency from power measurements on each side of transformer, VSD, etc..	- Accuracy of many power meters is suspect when used between the VSD and motor. Monitor on the line side of the drive to minimize this problem. - Measurements on the high side of transformers (above 600V) will require the availability of already installed instrumentation.
	Calculate overall pump and electrical train efficiency using measured input power, head and flow rate	- Accuracy depends on individual measured component accuracies. This is the authors' preferred approach.

Table 1. Measurement considerations (continued)

	Method		Notes
kW	Calculated from head and flow rate measurements, pump performance curve and assumed electrical efficiencies	-	Not recommended unless there are no alternatives.
	Estimated from measured current	-	Can be reasonably accurate, provided that representative motor performance curves are available.
	Estimate from measured speed	-	Not recommended. Errors can be particularly large on newer, low-slip motors.

After selecting the methodology of data collection, a table can be developed for each flow interval as shown below in Table 2. Only 5 flow intervals are shown, however each pump system reviewed may have more or less intervals depending on the application

Table 2. Flow Interval Data Collection Table

Interval	Pump Flow (gpm)	Head (ft)	Pump Efficiency	Motor Efficiency	Calculated kW	Annual Hours of Operation	kWh
1							
2							
3							
4							
5							
Totals	**--**	**--**	**--**	**--**	**--**	**8760**	

From the collected data, it is often useful to plot head and flow measurements from field collected data compared to the original pump curve data. This can be done with pump modeling programs, or with the graphing option of spreadsheet programs. This modeling is especially useful to determine how output and efficiency performance is effected in a multiple pump system.

Examples of Cost Saving Opportunities

Case Study A: Municipal Wastewater Pump System

Ten 3,500-hp pumps equipped with A.C. variable speed drives were evaluated to identify potential areas of savings. A review of pump system operation revealed the following:

- The suction tank (wetwell) average operating level could be increased to reduce total system head
- Multiple pump system efficiency could be improved with automatic control systems

A review of the pump system operation provided the following data:

Existing Pump Operation (one flow interval shown over 36 performed for full analysis of this system)

Pump Type:	Vertical single stage centrifugal pumps
Total Head (TDH):	85 ft.
Flow:	110 million gallons per day (76,000 gpm)
Pump Efficiency:	90% (from certified curve)
Combined motor and electrical supply efficiency:	92% (from electrical calculations)

The product of the pump and combined electrical efficiencies is 82.8%. From the pump equation, electrical power, in kW (P) can be calculated with the following equation:

$$P = \frac{Q\,H\,\gamma}{5310\ \eta_T}$$

For one of the existing flow intervals:

$$P = \frac{76{,}000 \text{ gpm} * 85 \text{ ft} * 1}{5310 * 0.828} = 1.47 \times 10^3 \text{ kW}$$

Proposed Operation

The potential of raising the level on the suction side of the pump by 2 feet provided an excellent opportunity to reduce the total head on the pump system. Although the adjustment seems small, the ability to ramp down the variable speed drive slightly to produce the same flow at a lower drive speed provided the following savings (pump and electrical efficiencies did not change significantly):

$$P = \frac{76{,}000 \text{ gpm} * \boldsymbol{83} \text{ ft} * 1}{5310 * 0.828} = 1.43 \times 10^3 \text{ kW}$$

Power, energy, and cost savings were calculated:

Reduction in motor power:	0.04×10^3 kW = 40 kW
Reduction in annual energy usage[2]:	40 kW * 8760 hours/year = 3.5×10^5 kWh
Annual cost savings:	3.5×10^5 kWh * \$0.070/kWh = \$24,500

This simple control system adjustment did not require an investment and paid for itself immediately.

Case Study B: Municipal Water System

An existing 200-hp finish water pump (with the motor driven directly off the line) was reviewed to determine the cost effectiveness of repairing or replacing the existing pump. Repair estimates to refurbish the worn pump was approximately $10,000.

A review of the pump system provided the following generic nameplate information:

Pump Type:	Split case horizontal pump	**Design flow rate, head:**	1,300 gpm, 415 ft
Pump efficiency from curve:	70%	**Nameplate motor efficiency:**	94% (2-pole motor)

Other electrical system losses were assumed negligible.

The nameplate pump efficiency of 70% was compared with information included in Hydraulics Institute (HI) standard ANSI/HI 1.1-1.5-1994, which indicated an achievable efficiency of 80%. To determine the potential savings of replacing the pump with a higher efficiency unit, the 70% pump efficiency was confirmed by taking flow, head and kW measurements (electrical efficiency losses were assumed as noted above). Rearranging the pump curve confirmed the efficiency noted on the pump curve as shown below:

$$\eta_p = \frac{Q\,H\,\gamma}{5310\ P\eta_m}$$

The motor power, measured at the nameplate conditions, was 155 kW. The pump efficiency was then estimated to be:

$$\eta_p = \frac{1{,}300 \text{ gpm} * 415 \text{ ft} * 1}{5310 * 157 \text{ kW} * 0.94} = 69\%$$

After a review of energy rate schedules and system configuration, the pump efficiency was reviewed in more detail to determine if other manufacturers could provide better efficiency at the some flow

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[2] To simplify this example, the operating condition is assumed to be constant throughout the year.

conditions. Based on a few phone calls, several pumps with the same configuration were found that could achieve the same flow and head at 80% efficiency (which, incidentally, was consistent with the HI standard noted above). Combining this improved pump design with a premium efficient motor (96% efficiency) produced the following:

$$P = \frac{Q\,H\,\gamma}{5310\,\eta_p\,\eta_m}$$

$$P = \frac{1{,}300 \text{ gpm} * 41.5 \text{ ft} * 1}{5310 * 0.80 * 0.96} = 132 \text{ kW}$$

Reduction in motor power:	25 kW
Annual kWh Savings:	25 kW * 8760 hours/year = 219,000 kWh
Annual cost savings:	219,000 kWh * $0.090/kWh = $19,710 annual savings

This project paid for itself within 2 years (not including the avoided $10,000 maintenance expense on the old pump).

Summary

It is our hope that this paper has successfully presented useful information to help operators, engineers and maintenance staff identify cost saving opportunities for their pump systems. As demonstrated in the case studies above, some quick calculations and data collection can help discover improvements that are cost effective and help justify investing in new pump equipment, control systems and efficient electrical systems.