PUMP ENERGY CONSERVATION TECHNIQUES

by

J. C. Cone

Senior Specialist Engineer

E. I. du Pont de Nemours & Company

Engineering Department

Wilmington, Delaware

J. Cary Cone is a rotating equipment specialist with E.I. du Pont de Nemours' Engineering Department, Design Division. He is responsible for the specification, design, installation, and startup of pumps, compressors, and turbines. Previously, he has worked in the Engineering Department's Service Division as a consultant specializing in vibration analysis, field troubleshooting, overhaul, and upgrading of rotating equipment. Mr. Cone's eight years experience with du Pont also includes assignments in plant R&D groups in Martinsburg, West Virginia and Nashville, Tennessee. He received his B.S.M.E. degree in 1971 and a M.S.M.E. degree in 1972 from Virginia Polytechnic Institute, is a registered professional engineer in Delaware and Tennessee, and is a member of ASME.

ABSTRACT

Thousands of dollars are wasted annually in a typical petrochemical plant through inefficient operation of pumps. Excess capacity, changing operating conditions, inefficient control, and inadequate maintenance are some of the more common sources of wasted energy. In many cases, significant energy savings can be made by systematically applying existing technology to reduce pump energy consumption. This paper reviews available pump energy saving methods and presents application guidelines for both new pumping designs and retrofit into existing installations. Detailed examples are given for each of the twenty pump energy savings techniques that are discussed.

INTRODUCTION

The typical petrochemicals plant has hundreds of pumps in operation, many of which run inefficiently and waste energy. While the loss per pump is relatively small, this wasted energy can often cost a plant several hundred thousand dollars annually. If energy costs continue to escalate at the present rate, equipment designed and installed today will operate at energy costs exceeding $1,000/hp-yr before the end of its useful life. Today we must take steps we would not have even considered five years ago to reduce pump energy consumption. This paper reviews proven pump energy saving techniques that are applicable for both new pumping designs and for retrofit into existing applications. By systematically applying these techniques, significant energy savings can be achieved.

GENERAL PUMP OPERATING CHARACTERISTICS

Before proceeding with a detailed discussion of pump energy saving techniques, let us review the general operating characteristics of a pump and see where energy can be saved. This discussion is aimed specifically at centrifugal pumps since they form the vast majority of petrochemical pumping applications. However, some of the energy conservation principles apply equally well to other types of pumps.

Figure 1 represents a general performance curve for a centrifugal pump as a plot of head (pressure) as a function of flow. Centrifugal pumps are variable capacity and variable head devices as represented by the curves D_1 and D_2. Curves D_1 and D_2 can represent two different pumps, the same pump at two different speeds, or the same pump with two different diameter impellers. Pump output is determined by the point where the system resistance (pressure or head) equals the pump head as shown by the points 1 and 2.

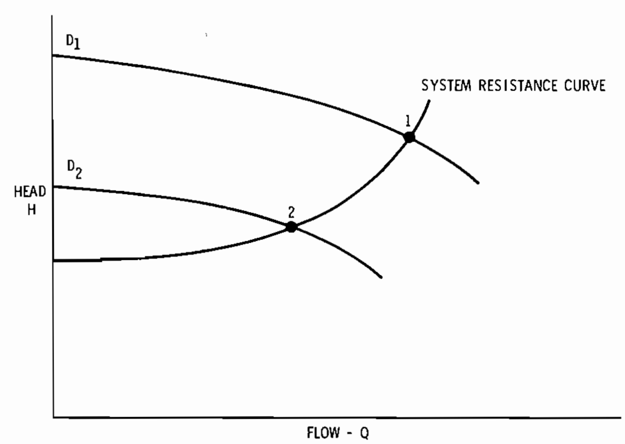


Figure 1. Characteristic Curve of a Typical Centrifugal Pump.

Pump energy losses are expressed in units of energy (horsepower, Btu's, or watts). In most cases, the pump energy losses cannot be measured directly and must be determined by analyzing the pump system power requirements, inspecting the pump curves, and making indirect measurements such as flow, pressure, fluid, or pump geometry. The most basic equation for evaluating pump performance is:

$$\text{Pump Horsepower} = \frac{\text{Flow} \times \text{Head} \times \text{Specific Gravity}}{\text{Pump Efficiency}}$$

From this equation, one can see that there are three fundamental causes of wasted pump energy:

1. excess flow.
2. higher than necessary head.
3. lower pump efficiency.

PUMP ENERGY SAVINGS TECHNIQUES

One need consider only three fundamentals to avoid wasting pump energy: avoiding excess flow, excess head, and low efficiency. However, these fundamentals can be expanded into many proven pump energy saving techniques. These techniques are summarized in Table I and are discussed in detail in the following pages. The discussion of each method includes advantages and disadvantages, when the method can be used, potential for energy savings, cost to apply, and a detailed example. These twenty energy savings methods are presented in three groups: primary design methods, secondary design methods, and field methods. The primary design methods are the techniques which have the broadest application and greatest potential savings in general pumping installations. The secondary design methods include techniques for use in more specialized, infrequently encountered applications and techniques with potentially smaller energy savings. The field methods are simple, low-cost techniques most readily applied to existing installations. To allow a limited comparison of one method with another and to simplify the examples, a power cost of \$.045/kWh, a utility of 91% (8,000 hr/yr), and a motor efficiency of 89.5% are used throughout the paper. This gives a power cost of \$300/hp-yr.

PRIMARY DESIGN METHODS

1. *Use More Efficient Pump*

Selecting a more efficient pump requires careful consideration of three factors: 1) system design, 2) type of pump for the job, and 3) proper sizing of the pump.

System design is the most important aspect of selecting a more efficient pump. Good system design starts with a complete understanding of how the system operates, the range of heads and flows required, and the fluid to be handled. For existing pumping systems, the pump requirements should be confirmed by actual pressure and flow measurements. Next, the system should be analyzed to determine what modifications will reduce the total pumping requirements. Such modifications can be as fundamental as allowing a larger temperature rise through heat exchangers to reduce cooling water flow or as simple as trimming an impeller to reduce excess head and flow.

Once the requirements of the pump are defined, the second step is to choose the best type of pump for the job [1, 2, 3, 4]. This means selecting a pump from a multitude of positive displacement (reciprocating, screw, diaphragm, etc.) or centrifugal (single-stage, multistage, high-speed, vertical, etc.) pumps. Centrifugal pumps are the best choice for most applications. Positive displacement pumps are generally more efficient than centrifugal pumps and should be considered whenever possible in low flow, high head applications and when handling viscous fluids. Of course, other factors such as maintenance cost, first cost, or materials of construction greatly affect the type of pump selected. To assist in narrowing the choice to the best type of pump, general application charts such as Figure 2 or Figure 3 are helpful.

Proper sizing of the pump is the third step in selecting an efficient pump. Probably more energy is wasted by oversizing pumps than by any other design factor. Even the most efficient pump can be grossly inefficient if oversized for the actual application. For pump installations, considerable effort should be made to use an efficient pump. A pump with 1% to 2% higher efficiency can save thousands of dollars in power costs over its useful life and can justify an initially higher cost pump. Coating or polishing "as cast" impeller surfaces and smoothing impeller contours can often increase pump efficiency about 1%. In existing installations, inefficiently operated pumps can be replaced with more efficient pumps.

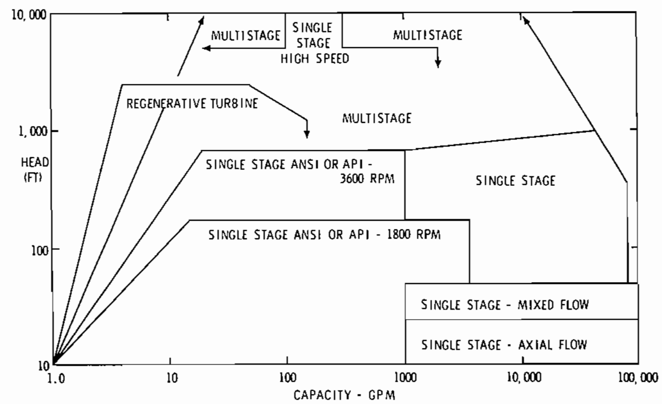


Figure 2. General Head and Capacity Limits for Centrifugal Pumps.

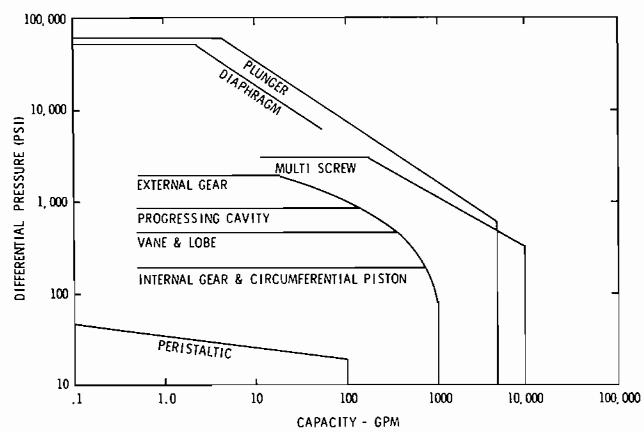


Figure 3. General Pressure and Capacity Limits for Positive Displacement Pumps.

<u>Example 1</u>

As an example of an inefficient pump, let us assume that a pump system has been properly sized and designed for minimum energy consumption and that the system requires 200 ft head and 800 gpm flow. Figure 4 shows two centrifugal pumps that are properly sized for this application. Pump A has a 7% higher efficiency than Pump B (74%

TABLE 1. SUMMARY OF PUMP ENERGY SAVING TECHNIQUE

METHOD	WHEN TO USE	ENERGY SAVINGS	COST TO APPLY	ADVANTAGE	DISADVANTAGE
A. PRIMARY DESIGN METHODS					
Use more efficient pump	Generally in design stage	Generally small	High, requires new equipment	Efficient pumps cost little extra	May increase spare parts inventory
Reduce system requirements in design	Used in design stage	Variable savings; can be large	Variable; can justify changes that cannot be made later	Greatest flexibility to make changes in design	Requires early analysis when design uncertainties are greatest
Avoid pump selection limitations	Used in design stage	Moderate savings	Low; often no cost penalty in design	Easily applied in design stage	Not generally applicable for retrofit
Use variable speed drives	For variable flow rates and extended operation at low flow	Large energy savings; best in frictional loss pumping systems	Moderate to high investment	Matches pump to system requirements, increases operating flexibility	More complex control
Pump control by throttling	For variable flow from 20-100% of max.	Moderate savings	Moderate cost	Simple, reliable, and widely used	Not effective for controlling low flows
Select low pressure drop control valves	Consider for any pump control valve	Variable savings; can be large for high-pressure drop	No cost penalty in design; moderate costs to replace existing valves	Reduces valve noise and maintenance	Requires well-defined pump system to minimize valve pressure drops
Eliminate fixed orifice bypass	Pumps with a continually open bypass line	Moderate, 10-25% of flow is bypassed	Moderate; requires automatic bypass control	Reduces valve noise and maintenance	Increased maintenance for bypass control loop
Replace oversized pumps	Constant operation below 50% pump capacity	Large; 50% or greater power savings	High; requires new pumps	Saves energy at all operating conditions	Not applicable to widely varying flows
Use multiple small pumps	For large variations in pumping demand	Large savings	High; requires additional pumps and control system	Increase operating flexibility and reliability	Requires careful pump control to achieve savings
B. SECONDARY DESIGN METHODS					
Use small booster pump	A low flow, high pressure flow path in a low pressure system	Large savings	High investment cost for booster pumps	Large reduction in total system pumping power	Retrofit in existing installations costly and difficult
Power recovery using a pump as a turbine	Use where high pressure fluid is let down to a lower pressure	Large savings, recovery of 40-60% total energy	High investment cost; best applied in design	Recovers energy otherwise lost	Requires relatively constant head and flow
Limit the use of lower efficiency specialty pumps	Generally used in design stage	5-40% energy savings with efficient standard pump	Frequently lower capital cost using standard pumps	Large energy savings	Special pumping conditions sometimes require less efficient pumps
Avoid gas entrainment	Where entrainment causes head or capacity losses	Variable; can be large	Moderate; generally to modify suction piping	Increases capacity and head; improves efficiency	Very limited application
Use more efficient motor	Avoid oversizing motors; use high efficiency motors	Small	Lower for smaller standard motors; premium for high efficiency motors	Saves energy continually	Cannot justify replacing existing motors
Eliminate pump seal cooling	Pumps handling hot liquid	Variable; reduces process heating & cooling water use	Low; requires high temperature seal such as metal bellows seal	Saves process energy needed to reheat cooled pumpage	Does not reduce pump power requirements
Minimize losses from mechanical seals and packing	Use in design, seals use less energy than packing	Very small	Low cost	Standard seals consume little power	Factors other than energy are more important in selecting seals
C. FIELD METHODS					
Shutdown unneeded pumps	Consider in multiple pump system	Large	Low; improved operating practices and controls	Little or no capital investment	Increased operations attention
Proper maintenance of existing pumps	Best in abrasive, corrosive or low flow and high head service	Small, typically 4-6% power savings	Low; can be a part of routine shutdowns	Uses normal spare parts	More extensive or frequent maintenance
Trim impellers to reduce excess head	Centrifugal pumps with excess flow or pressure	Variable; typically 10-15% power savings	Low	Matches pump to actual operating conditions	Does not work with rapidly varying flows
Select impellers to reduce excess capacity	Same as trimming impellers	Moderate; up to 20% power savings	Low; requires a new impeller	Matches pump to actual operating conditions	Does not work with rapidly varying flows

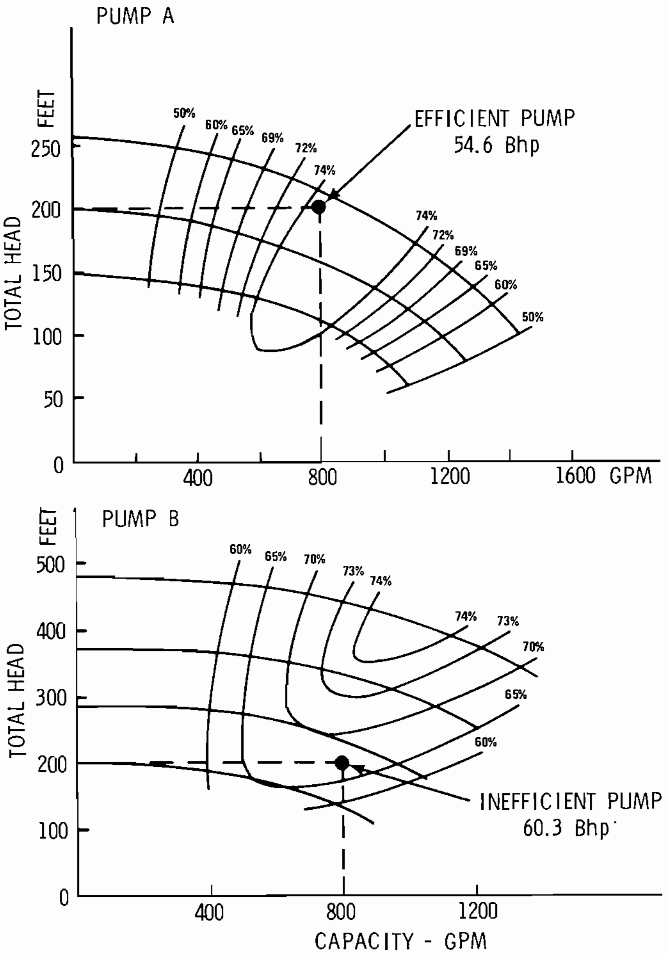


Figure 4. Comparison of Efficient and Inefficient Pump — Saves 5.7 Bhp.

vs 67%), which results in a power savings of 5.7 Bhp. With electricity costing $.045/kWh, this yields over $1,700/yr savings in power costs.

2. *Reduce System Pumping Requirements in the Design Stage*

Substantial reduction in pumping requirements can be made in the design stage by making basic process changes that could not be economically justified at a later stage. While good pump selection techniques will lead to an efficient pump and control system, this does not maximize energy savings. Even results from sophisticated pump and piping evaluation computer programs are only as good as the basic input data. Some common ways to reduce pumping system requirements in the design stage are:

a. Question the basic data. Are the maximum flow and pressure limits realistic? Will the maximum flow occur with the maximum pressure? Are large turndowns really necessary? Answers to these and other questions can reduce the oversizing of pumps and increase operating efficiency.

b. Increase NPSH available to the pump — Low NPSHA (net positive suction head available) can lead to the selection of an oversized, less efficient pump. NPSHA below 10 ft is considered low; NPSHA below 5 ft is very low and sometimes requires special low NPSH pumps. NPSHA can be increased in system design by raising the feed tank, raising the liquid level in the feed tank, pressurizing the feed tank, inserting the pump in the tank, or minimizing inlet pipe friction losses.

c. Allow a higher cooling water temperature rise across heat exchangers to reduce cooling water pumping requirements. The temperature rise across existing heat exchangers should be checked in the field to see if excess cooling water is being pumped.

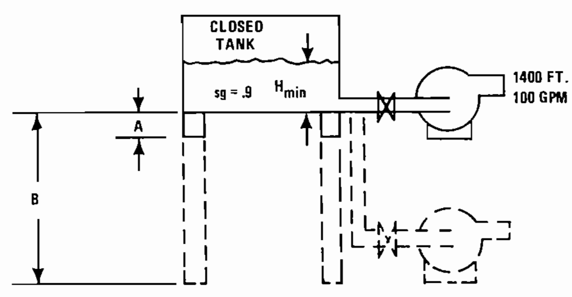
d. Heat a viscous process fluid prior to pumping to reduce viscosity. This reduces friction losses and lowers required horsepower. The cost of heating the fluid must not outweigh the savings in pumping energy.

e. Minimize pressure drops in piping, heat exchangers, valves, columns, and other process equipment.

f. Separate high pressure flow loops from low pressure flow loops. Use booster pumps where appropriate.

Example 2

The elevation above grade of feed tanks and other process equipment is often fixed before pumps are selected. To minimize first costs, the tank heights are kept low. As shown in Figure 5, if the feed tank is set too low, the savings in lower tank elevation can be offset by higher costs required to pump 100 gpm of a .9 sg liquid at 1400 ft head. With the tank set at height A, only 2 ft of NPSH is available at minimum operating level. Limited overhead space precludes the use of a vertical turbine pump. This application requires the use of a special low NPSH multistage regenerative turbine pump (Pump A) which is more expensive and less efficient than other pumps. If the tank height were increased 3 ft, NPSH available would be increased to allow use of the lower cost, more efficient horizontal multistage centrifugal pump (Pump B). This results in a $15,000 reduction in pump first costs and a $7,350/yr reduction in operating costs. These savings in pump first costs and pump operating costs easily justify the $20,000 additional cost to raise the tank 3 ft. However, once the tank height is set, it becomes increasingly costly



	LOW TANK A	HIGH TANK B
1. PUMP	REGENERATIVE TURBINE	MULTISTAGE CENTRIFUGAL
2. PUMP & MOTOR COST	$26,000.	$9,000.
3. NPSH REQUIRED	1 FT.	4 FT.
4. NPSH AVAILABLE	2 FT.	5 FT.
5. PUMP EFFICIENCY	42%	62%
6. POWER REQUIRED	75.8 Bhp	51.3 Bhp
7. POWER COSTS (300 $ - HR/YR)	$22,740 / YR.	$15,390 / YR.

Figure 5. Reduce System Pumping Requirements by Increasing NPSH and Using More Efficient Pump — Saves 24.5 Bhp.

to change the tank and piping elevations. If the tank height is fixed too low in the early design stage, the energy savings would not justify raising the tank at a later date. Even in later design stages, it may be too costly to raise the tank.

3. *Avoid Pump Selection Limitations*

Application of some pump selection procedures and rules-of-thumb results in many oversized, inefficient pumps. The common pump selection limitations fall into three general categories: 1) oversizing pumps, 2) limitations on pump efficiency and 3) excess factors of safety. They are often applied with little regard to the energy costs involved. Of course, other engineering factors can prevail over the energy savings involved; however, one should give full consideration to the potential energy costs. The most common pump selection limitations are:

a. Oversizing pumps — First, let's consider some of the underlying reasons for oversized pumps. Many of these reasons were once sound, conservative engineering practices used to "guarantee" performance. In view of present energy costs, many of these reasons must be reevaluated in terms of their true energy cost penalty.
   1) Poorly defined basic data — Since a pump must be sized for the worst case, increased attention should be used in selecting the extremes in head and flow. Unrealistic combinations of head and flow often result in pumps greatly oversized for normal conditions.
   2) Multiple effect of adding "fat" — Large capacity or head allowances applied by everyone from basic data preparation to vendor quotes lead to an oversized pump "guaranteed" to meet required heads and flows. For instance, if the plant basic data, basic design specifications, and vendor quotations each contained a 10% factor of safety, the resulting pump would have 33% excess capacity. Often the factors of safety are included in less obvious ways. Uncertainties in scaling basic data from other plants or flow sheets leads to some "fat". Other fat is applied as high control valve pressure drops to allow for control.
   3) Allowances for future capacity — Oversized pumps installed so plant capacity can be increased by opening a valve is an expensive operating flexibility. The energy wasted in the months or years before the expansion is needed can often pay for new pumps several times over, and sometimes the planned expansion never materializes.
   4) Plant operating conditions change from design conditions.
b. Subtle limitations on pump efficiency — Everyone says they selected the most efficient pump, but how often was it selected under one of these restrictions?
   1) "The pump must be a centrifugal". Many applications could be more efficiently handled by other types of pumps.
   2) "Only an ANSI pump will be used". Using ANSI pumps can simplify installation and design and minimize spare parts; however, other pumps may be better choices in some cases.
   3) "Use one vendor for all the pumps". While one vendor's selection may be a good choice for most of the pumps, he may have quoted some lower efficiency pumps. No vendor's line of pumps covers a wide range of pressure and flow with equal efficiency.
   4) "No 3500 rpm pumps". Fear of higher maintenance costs with 3500 rpm pumps can limit selection to low speed pumps and can lead to larger efficiency losses, especially in lower flow and higher head applications. These fears are no longer justified based on the current state of the art.
c. Excess factors of safety — Two widely used pump selection rules which lead to oversized pumps are [5]:
   1) "Pumps with constant speed drives shall be capable of at least a 5% head increase at rated conditions by installing a new impeller". This statement is part of API 610. It adds an additional 5% safety factor to a specification which usually already includes a large design margin. This rule leads to increased energy consumption when a pump casing with the maximum sized impeller would just fit the rated specifications. To allow a 5% head increase, the next larger pump casing would have to be used with a severely trimmed impeller. In addition to purchasing a larger, more expensive pump, the pump generally operates at lower efficiency. The head margin rule should be applied with caution, especially when large sizing factors are already included in the specifications.
   2) "Don't select a pump to operate to the right of the best efficiency point". This is an unwritten but frequently followed rule-of-thumb which ensures that a pump will not run out on the curve and also provides a margin of excess capacity. This rule eliminates almost half of the possible pumps from consideration (the lower capacity pumps) and results in the selection of larger capacity pumps. The losses are multiplied since the maximum capacity is usually larger than the normal operating point. With the maximum capacity point at or slightly to the left of the best efficiency point, the pump must be throttled back to a lower efficiency portion of the pump curve for normal operation. By allowing pumps to be selected to the limit of the pump curve, the maximum operating point can fall to the right of the best efficiency point. When throttled back to normal flow, the pump would operate closer to the best efficiency point.

<u>Example 3</u>

To illustrate two common pump selection limitations, consider the selection of a pump for a maximum flow of 1300 gpm at 140 ft head and a normal flow of 1000 gpm at 120 ft head. From Figure 6 we can see that Pump A will operate at the maximum flow condition at 78% efficiency and will use 58.9 Bhp. Pump A must be throttled to operate normally at 1000 gpm, 162 ft head, 79% efficiency, and 51.8 Bhp. While this pump meets the flow requirements, it violates two common pump selection limitations. The maximum rated flow of Pump A is to the right of the best efficiency point and a larger impeller cannot be used to increase the head an additional 5%. If either of these rules are applied, then the next larger pump casing size would be selected (Pump B with a 12″ impeller). At maximum flow, Pump B would operate at 1300 gpm, 140 ft head, 74% efficiency, and 62.1 Bhp. Pump B must be throttled well back on the curve to operate normally at 1000 gpm, 150 ft head, 65% efficiency, and 58.3 Bhp.

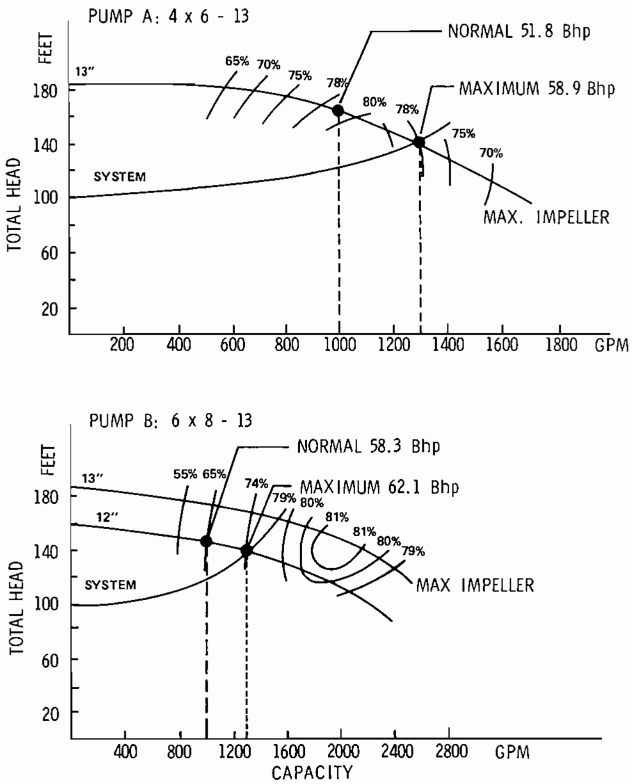


Figure 6. Avoid Pump Selection Limitation by Using Pump with Maximum Size Impeller and Operation to Right of Best Efficiency Point — Saves 6.5 Bhp at Normal Operation.

Compared to Pump A, the larger Pump B would require 5% more power (3.2 Bhp) at maximum flow and 13% more power (6.5 Bhp) at normal operation. If the pump operated at normal rates 75% of the time and at maximum flow 25% of the time, it would cost $1,700/yr (based on $.045/kWh) in increased power costs to follow either design selection limitation.

4. *Use Variable Speed Drives*

Variable speed operation is the most efficient means of matching pump output to varying system requirements in many pumping applications. Variable speed operation of centrifugal pumps can match the output of the pump to the system and save the energy that is normally lost as pressure drop across a control valve or as excess flow in a bypass system. Additional energy savings are obtained with variable speed operation due to increased pump efficiency. The pump's best efficiency point maintains its relative position on the pump curve as speed is reduced and this results in a higher pump efficiency at low flows than can be obtained through throttling. Further savings from variable speed operation result from reduced pump maintenance cost. Also, control valve maintenance costs can be eliminated by using variable speed control to replace control valves. Lower speed operation of pumps reduces vibration and wear and reduces seal and bearing problems. In some cases, process savings can result from variable speed pump operation. For instance, heat build-up and high turbulence from throttled high speed pumps can cause product degradation and/or breakup of crystals in some process applications.

When should variable speed pump operation be used? Since variable speed drives generally have a higher investment cost than bypass or throttle control systems, they should be used only where the total energy, process, and maintenance savings will pay for the increased investment. Pump systems where the head and/or flow requirements vary and are frequently operated below 75% of maximum design conditions are likely candidates for variable speed operation. Pumping systems with mostly frictional resistance (losses proportional to flow) offer greater potential energy saving than mostly static systems (losses independent of flow), and are thus easier to justify the use of variable speed control. Use of variable speed drives must be evaluated on a case by case basis after the pump system operation is fully analyzed at all flow rates.

There are many different ways to vary pump speed; the more common methods are listed below. It is beyond the scope of this paper to discuss in detail each type of variable speed drive:

- Steam turbines
- Gas turbines
- Internal combustion engines
- Variable frequency drives
- DC drives
- Wound rotor motors
- Traction drives
- Fluid drive coupling
- Magnetic clutch
- Variable pitch sheave belt drives
- V-belt drive
- Two-speed motors

Example 4

Variable speed control can match the system resistance curve and can save energy at reduced flow rates (see Figure 7). At the 100% capacity point, the pump performs equally well with variable speed, throttle, or no control.

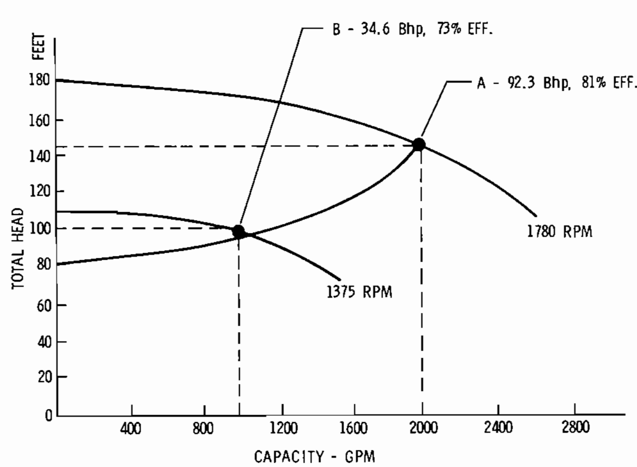


Figure 7. Variable Speed Control Matches Pump to System — Saves 57.7 Bhp vs No Control, Saves 36.7 Bhp vs Throttle Control.

The pump runs at 2000 gpm flow, 148 ft head, 81% efficiency, and 92.3 Bhp. However, at 50% capacity, the pump speed is reduced to 1375 rpm and operates at point B (1000 gpm, 100 ft head, 73% efficiency, and 34.6 Bhp). This is a savings of 57.7 Bhp compared to no control (point A) and 36.7 Bhp compared to throttle control (Figure 8, point B). With \$.045/kWh power and half time operation at 50% capacity, the power savings are \$8,600/yr and \$5,500/yr, respectively.

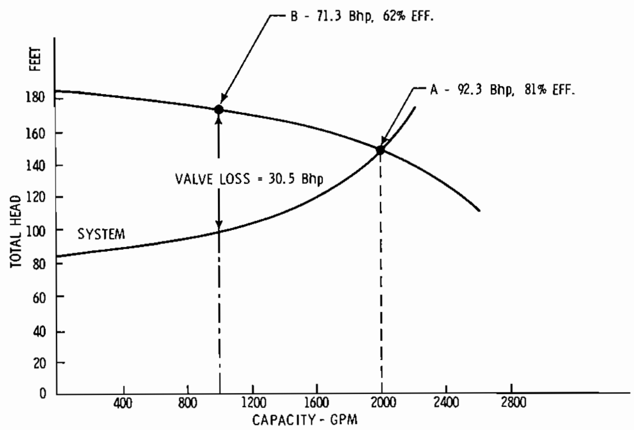


Figure 8. Pump Control by Throttling — Saves 21 Bhp vs No Control.

5. *Pump Control by Throttling*

Controlling a centrifugal pump by throttling the pump discharge is an energy wasteful practice. However, throttle control of a centrifugal pump is generally less energy wasteful than two other widely used pump control alternatives: no control and bypass control. As such, throttle control can represent a means to save pump energy. Also, throttle control is the most widely used and is often the lowest investment cost method to control the output of a centrifugal pump.

Throttling the discharge is a simple, effective method of controlling the output of a centrifugal pump. Since a centrifugal pump is a variable capacity device, it will operate at the intersection of the pump curve and the system curve. If the pump discharge is throttled by closing a valve, the pressure drop across the valve increases and causes the pump to operate back on the pump curve, thereby reducing the pump output. The throttling can be controlled manually or by an automatically actuated control valve.

There can be problems in using throttling control. First, the pressure drop across a control valve represents a loss of energy. Second, as a single volute pump is throttled back, hydraulic radial forces on the impeller increase and result in increased shaft deflection and vibration. This often leads to rapid wear and failure of seals, bearings, wear rings, impellers, and shafts. Third, a severely throttled pump operates with much internal recirculation which can lead to cavitation, increased vibration, high wear, and erosion. Fourth, pumps throttled back near shutoff rapidly generate heat which can boil the liquid and cause it to run dry. On multistage pumps, this can destroy a pump in minutes due to thrust bearing failure or heating and seizing of the pump rotor.

From an energy conservation viewpoint, throttle control should generally be used in preference to no control or bypass control. With no pump control, the pump will run out on the pump curve (Figure 8, point A). Any excess flow represents wasted energy. By throttling the pump discharge, the pump will operate further back on the system curve and will use less energy (Figure 8, point B). A bypass control system consumes energy like a pump system with no control; the pump always operates out on the pump curve at maximum flow. As a control system, bypass control generally does not save energy. Two exceptions are the use of bypass control for high specific speed pumps (mixed flow and axial flow) and for regenerative turbine pumps. These pumps require more power at low flow than at design flow, and throttle control would require more power than bypass control. Bypass control does offer a reliable way to control very low flow rates, a low cost way to maintain constant header supply pressure, and a means of controlling positive displacement pumps.

In addition to replacing bypass control or no control, there are additional considerations in using throttle control. Throttle control should be considered where rapidly variable flows are required. Good control can be maintained from 20% to 100% of the maximum flow. Throttling should be avoided at flows below 20% of the pump best efficiency flow on most single stage pumps. Some high horsepower or multistage pumps can be damaged if throttled below 75% of the design flow.

Example 5

Throttle control of a centrifugal pump can save energy over an uncontrolled system. The pump shown in Figure 8 recirculates a process solution (sg = 1.0) at a design rate of 2000 gpm. The system resistance curve is made up of 84 ft static head and friction head and friction head that varies with the flow. The process runs on hourly cycles and operates for half the time at 50% capacity. The energy savings from using throttle control to reduce recycle flow during the 50% capacity cycle needs to be determined. With no control, the pump operates constantly at the 100% design point A (2000 gpm, 148 ft head, 81% efficiency, and 92.3 Bhp). If throttled to 50% capacity, the pump will operate at point B (1000 gpm, 175 ft head, 62% efficiency, and 71.3 Bhp) for a savings of 21 Bhp. With \$.045/kWh power and 50% operation at half capacity, this saves \$3,150/yr in power costs.

While throttling saves horsepower compared to no control, considerable horsepower is still lost across the control valve. The control valve loss is over 30.5 Bhp at point B. Variable speed control, as previously discussed, is a more energy-efficient control alternative.

6. *Select Valves for Lower Pressure Drop*

Pressure drop across a control valve represents a waste of energy. Valve pressure drop can be reduced by matching the pump to the system requirements. Still further energy savings can be accomplished by minimizing the pressure drop selected to control the valve. Generally, 5 psi or less is an adequate pressure drop to maintain good control at the maximum design point in a single flow path system. When a control valve is selected with a higher pressure drop, the total cost penalty often involves more than the energy lost across the valve. The higher pressure drop requires a higher head pump which often results in a larger, more expensive pump and motor and reduced pump efficiency. Also, valve noise and maintenance are increased as the valve pressure drop is increased.

There are many reasons why pump control valves are frequently found with 5 to 50 psi pressure drops. The valves are undersized or oversized for the actual flows. High pressure drop valves such as globe valves were used where low pressure drop valves like butterfly valves could have been used. Rules of thumb (such as taking 50% of system pressure drop across the control valve or adding an extra pressure drop margin) were used in selecting control valves to avoid detailed system analysis. Oversized pumps and excessive basic data flow rates resulted in oversized systems. Wider than necessary flow control ranges resulted in high valve pressure drops. In defense of the designer, some extra control margin afforded by taking higher valve losses is necessary to allow for constantly changing basic data, for the uncertainties of scaling up process flow sheets, and to meet normal design timing.

By concentrating on energy losses at the design stage, control valves that combine low pressure drop and good control can be selected. The most useful parameter in sizing a control valve is the valve flow coefficient which relates the maximum flow through a valve, valve pressure drop, and liquid specific gravity:

$$Q = C_v \sqrt{\frac{P_1 - P_2}{sg}}$$

where C_v = valve flow coefficient from vendor catalog,

Q = flow in gpm,

$P_1 - P_2$ = pressure drop across valve in psi,

sg = liquid specific gravity.

The valve C_v depends on the type valve. For a given flow, gate valves have higher C_v's (lower pressure drop) than ball valves, and ball valves have higher C_v's than globe valves. The C_v also increases with valve size. Within a given size, valve C_v (and flow characteristics) can be changed by changing the valve trim. The key to selecting a low pressure drop valve is maximizing the valve C_v. Of course, there are many factors involved in selecting a control valve; therefore, valve selection should be left to a specialist. The following points in selecting low pressure drop valves should receive special emphasis:

a. Control valves must be equipped with good positioners.

b. Upstream pressure and temperature as functions of flow rate and downstream pressure as a function of flow rate should be specified for valve selection.

c. System turndown must be specified before control valves are selected. This includes maximum and minimum flow rates.

d. With large turndowns and small valve pressure drops, two parallel valves may be required to achieve control.

Pump energy savings can be achieved in existing installations by reducing excessive valve pressure drops. The first step is to identify potentially wasteful valves by looking for any of several factors:

a. Valves smaller than line size. Many flows could be controlled with line sized valves instead of smaller, higher pressure drop valves.

b. Valve positioners that show a heavily throttled valve. The system may be oversized and may operate between 25-50% instead of 50-95% capacity.

c. Noisy valves frequently indicate excessive throttling.

d. Valves requiring excessive maintenance.

e. Globe valves.

f. Single valves used for high turndowns. Beyond a 5:1 turndown, two valves in parallel may be a better choice.

The second step is to accurately measure the valve pressure drop and flow over the actual operating range. With this data in hand, one can work with a valve specialist to reduce pressure drop at maximum flow to 5 psi or less. This may be as simple as selecting a different valve trim. If the pressure drops are high enough, the energy savings may justify replacing the valve. After the valve has been selected for a lower pressure drop, the pump head must also be reduced in order to realize the energy savings.

Example 6

To illustrate the use of lower pressure drop control valves, consider the typical application of a 1½ × 3-13 centrifugal pump as shown in Figure 9. For a 200 gpm flow, the system consists of a 60 ft static head, 30 ft of friction head, and the control valve pressure drop. The system was initially selected with a low C_v (high pressure drop) control valve which required a pump to operate at point A (200 gpm, 145 ft, 53% efficiency, and 13.8 Bhp). The 13 in. impeller and 15 Bhp motor were selected to meet these conditions.

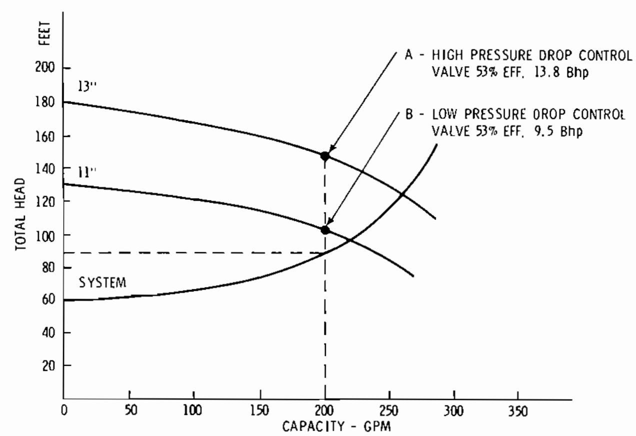


Figure 9. Use Low Pressure Drop Control Valve — Saves 4.3 Bhp.

Field measurement has found 23.8 psi pressure drop across valve A. How much savings could be achieved if this valve drop were reduced? Good control could be achieved with as little as a 10 ft (4.3 psi) valve pressure drop. The lower pressure drop valve would allow the pump to operate at point B (200 gpm, 100 ft, 53% efficiency, and 9.5 Bhp). This reduces the pump requirements by 4.3 Bhp and saves $1,290/yr in power costs with $.045/kWh electrical costs. This savings could be achieved by changing the valve trim from a $C_v = 41$ to a $C_v = 96$ and then reducing the impeller diameter from 13 in to 11 in. Replacement valves could also be justified if the energy savings were great enough. If this were a new design instead of a retrofit, a smaller, cheaper 10 Bhp motor could have been selected instead of a 15 Bhp motor for additional investment savings.

7. *Eliminate the Use of Fixed Orifice Bypass Flow for Pump Dead Head Protection*

A fixed orifice wastes energy when in continually bypasses flow from pump discharge to return. It is found on most boiler feed pumps and on many multistage process pumps handling volatile liquids near their boiling point. Typically, 10% to 25% of the process flow is bypassed. In addition to wasting energy, the continuous bypass increases capital costs by oversizing pumps and motors.

A bypass is essential to protect multistage pumps at low flows. For instance, boiler feed pumps are normally handling water near the boiling point. As the pump is throttled back near shutoff, the power taken by the pump goes into heating the water. At shutoff, only a few minutes operation will flash the water in the first stage to steam and cause the first stage to run dry. In a close tolerance multistage pump, this will destroy the pump by overheating and seizing. To prevent this damage, a minimum flow through the pump is used to prevent the temperature rise through the pump from exceeding 10° to 15° F. At low flows, a bypass is necessary to maintain the required minimum flow (typically 10% to 25% of the design flow). However, at process flow rates above the pump minimum flow, the bypass becomes unnecessary.

If the bypass is necessary only at low flows, why is a continously open bypass orifice used? The reasons usually offered are: 1) the orifice is low cost, 2) it's reliable, and 3) it's failsafe (always open). Although an orifice has a low initial cost, its energy cost is very high.

There is no question that an orifice is reliable and failsafe; but, automatic bypass control can be designed failsafe also. An instrumented flow control loop can be designed to bypass liquid at low flows and to close the bypass once the process demand exceeds minimum flow. This system bypasses only the flow necessary to assure the total flow exceeds the pump minimum flow requirements. A typical system consists of a check valve, recirculation valve, orifice, controller, and flow sensor. While complex, the flow control loop can be made both reliable and failsafe. An attractive alternative to an instrumented, flow control loop is an automatic recirculation control valve. These valves function as a complete, self-contained bypass system. The valve body actually contains a check valve, flow sensor, and bypass controller. The energy savings from eliminating a continuous bypass will often justify either system of automatic bypass control.

<u>Example 7</u>

A four-stage boiler feed pump is supplying 450 gpm of 210°F water (sg = .96) to make 216,000 pph of 600 psig steam. The pump is bypassing 25% of the total flow through a fixed orifice (Figure 10, point A). Presently, the pump is operating at 600 gpm, 1500 ft head, 76% efficiency, and 287 Bhp. If the fixed orifice were replaced by an automatic bypass control system to close the bypass

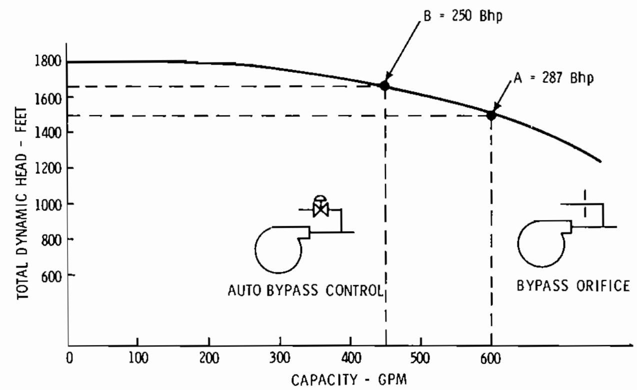


Figure 10. Automatic Control Closes Continuously Open Bypass Orifice — Saves 37 Bhp.

line, the pump would operate at point B (450 gpm, 1650 ft head, 72% efficiency, and 250 Bhp). This saves 37 Bhp or \$11,000/yr with \$.045/kWh power. The savings will easily justify the cost of a bypass control loop. Note that the savings are large even though the pump is throttled back slightly so it runs at a higher head and lower efficiency. If the pump had been sized initially with automatic bypass protection, additional power savings would have been gained by selecting a pump for more efficient operation.

8. *Replace Oversized Pumps*

Oversized pumps represent the largest single source of wasted pump energy. An oversized pump operates at a higher head or flow than required and if throttled, it operates at a lower efficiency. Each of these conditions waste energy. Under certain circumstances, an oversized pump can be replaced with a smaller more efficient pump. First, consider pumps that operate at a fixed speed and have a relatively fixed capacity. Highly variable flow rates are better handled by variable speed control, throttle control, or multiple pump operation. Next, look for pumps suspected of pumping more flow than required. These pumps are often throttled back by either a manual or an automatic valve. A quick check of the vendor's pump curve will show the range of impeller sizes for the particular pump. In most cases, limited reductions in excess flow or head (typically 10% to 50%) can be achieved by trimming impellers. Capacity corrections greater than impeller modifications may justify a replacement pump. This must be determined case-by-case.

Large energy losses from greatly oversized pumps are often not the only problem. Excessive throttling from oversized pumps can result in high vibration and noise and lead to increased pump and valve maintenance costs.

<u>Example 8</u>

Oversized pumps can be replaced with smaller pumps. For example, a process pump was specified for 800 gpm flow at 300 ft head. This included a 2X flow allowance (based on original design maximum flow) for a future capacity increase. A standard ANSI 3×4 ductile iron pump with 9 in impeller was originally chosen (Figure 11, Pump A). When the actual process operating conditions were measured, they were found to be only 200 gpm at 150 ft head (one-fourth the original design flow). Pump A was throttled and operated at 200 gpm, 360 ft head, 44%

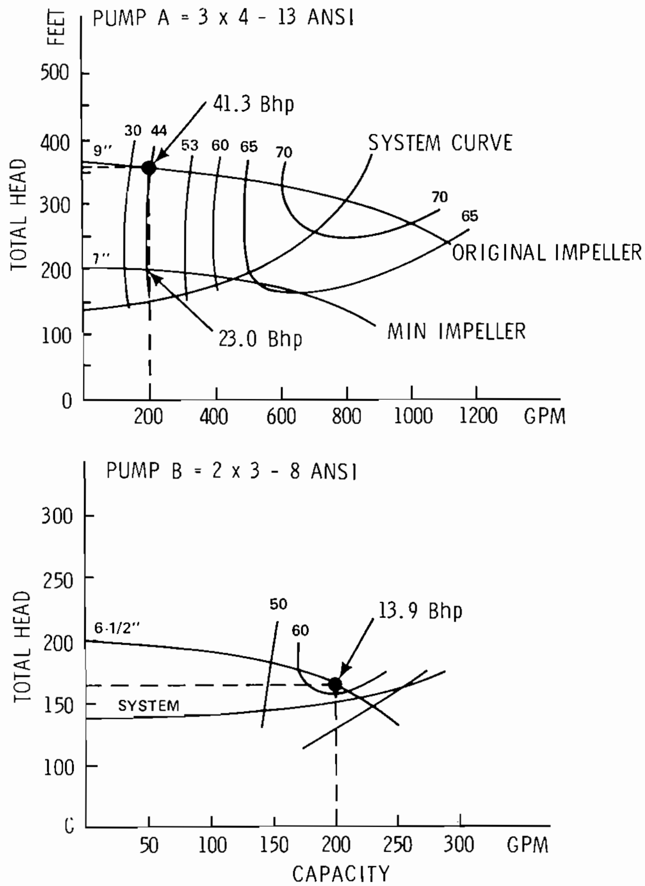


Figure 11. Replace Oversized Pump — Saves 27.4 Bhp vs Original Impeller, Saves 9.1 Bhp vs Minimum Impeller.

efficiency, and 41.3 Bhp. A look at pump curve A shows the actual operating point cannot be met even with the minimum size impeller and a replacement pump should be considered. By limiting the replacement pump selection to standard ANSI pumps, installation costs can be minimized. Note that Pump B ($2 \times 3 - 8$) will fit in the same 23½ in space as Pump A, thus minimizing installation costs. Pump B has been sized with some extra head and flow capacity and will operate slightly throttled at 200 gpm, 165 ft head, 60% efficiency, and 13.9 Bhp. Power consumption is reduced 27.4 Bhp compared to the original impeller and 9.1 Bhp compared to the minimum impeller. With \$.045/kWh power, the respective savings are \$8,220/yr and \$2,730/yr. These savings justify a replacement pump.

9. *Use Multiple Small Pumps in Place of a Single Large Pump*

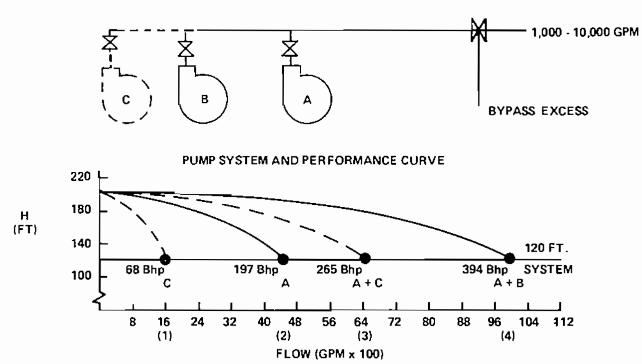
Substantial energy savings can be realized where there are large variations in pumping demand by using multiple small pumps in place of a single large pump. Multiple pumps offer an alternative to variable speed, bypass, or throttle control. The savings results from shutdown of one or more pumps at low system flow while the remaining pumps operate at high efficiency. Multiple small pumps should be considered when the minimum pumping load is less than half the single large pump maximum capacity. Multiple pumps are commonly used in chilled water systems, for boiler feed, and to meet cyclic production demands. Of all the methods of pump energy conservation, this method leads to some of the largest savings. Multiple pumps can be used: 1) in parallel to provide a widely ranging flow at relatively constant head, or 2) in series to provide a range of pressure at relatively contant flow.

Control of a multiple pump system is vital to realizing energy savings. Maximum energy savings are only obtained when the minimum number of pumps are operated. Manual pump control can be used where capacity variations are slow. Production line or shift demands and seasonal cooling or heating loads are slow demand changes. Automatic start-up/shutdown and control of pumps are needed for more rapid demand variations. Also, automatic control can simplify pump operation and eliminate power losses from improper system operation.

Proper control is not the only potential problem for multiple pumps systems. Check valves and block valves can leak a large percentage of the flow back through a shutdown parallel pump. A second potential problem is the selection of centrifugal pumps to operate in parallel. If nonidentical pumps having different head characteristics are used, one pump could assume the entire load or even pump in reverse through the pump with the lower head. These potential problems point out that using multiple pumps requires more careful consideration than using a single pump. However, if carefully applied, multiple pump systems can pay off with large energy savings.

Example 9

Multiple small pumps can be used to reduce pump energy requirements. Figure 12 shows a pumping system designed to provide 1,000 to 10,000 gpm of cooling water. Bypass control is used to maintain a constant 120 ft header pressure. The system was originally designed with two identical pumps rated for 5200 gpm at 120 ft head. After a year's operation, the system flow rates were found to vary



POINT	SYSTEM FLOW (GPM)	OPERATING TIME MONTHS	ORIGINAL SYSTEM PUMP A AND B: PUMPS RUN	Bhp	POWER COST*	SYSTEM WITH SMALL PUMP C: PUMPS RUN	Bhp	POWER COST*
1	1,600	6 MO	A	197	29,550	C	68	10,200
2	4,500	3 MO	A	197	14,775	A	197	14,775
3	6,500	2 MO	A+B	394	19,700	A+C	265	13,250
4	10,000	1 MO	A+B	394	9,850	A+B	394	9,850
TOTAL					\$73,875			\$48,075

* POWER COST = 300 \$/HP - YR = 25 \$/HP - MO

Figure 12. Use Small Pump in Multiple Pump System — Saves \$25,800/yr.

as shown in Figure 12. Even with two pumps, considerable water was being bypassed and the power costs totaled $73,875/yr. By adding the smaller capacity Pump C in parallel with the existing pumps, much smaller control increments could be obtained. This led to the $25,800/yr reduction in power costs shown.

SECONDARY DESIGN METHODS

10. *Use Small Booster Pumps to Reduce System Power Requirements*

Overall system energy requirements can be reduced by the use of a booster pump if two conditions are met. First, the pumping system must have more than one user or flow path. Second, one of the flow paths must be at a lower flow and substantially higher pressure than the remainder of the system. The energy savings are obtained by using a small booster pump to provide the high pressure flow and to allow the remainder of the system to operate at lower pressure and reduced power. Plant water systems are the most common multiple path pumping system. A single low-flow, high pressure user might force the entire plant water system to run at a higher pressure than otherwise required. For examples, a tall building might require a 100 ft higher head pressure than the rest of the plant, or one process might require higher pressure water than the rest of the system, e.g., a process using high pressure water jets.

Substantial reductions in energy costs can be achieved in systems meeting the above conditions. The multiple user flow systems are generally the plant's larger pumping systems where even small head reductions result in large horsepower savings. The best time to apply this method is in the design stage. Often, high pressure users can be grouped on the same flow path to reduce total system pumping requirements. Field installation of a booster pump is less common, but can result in large energy savings.

Example 10

Booster pumps reduce total system power requirements as illustrated in Figure 13. Two identical pumps supply 8000 gpm of 50 psi water to the powerhouse and 4000 gpm of 100 psi water to the plant. These two pumps operate at point A (6000 gpm, 240 ft head, 82% efficiency, and 443 Bhp) and use 886 Bhp. Because the pumps must operate at the maximum system pressure to supply the plant, the powerhouse water is pressure reduced to 50 psi. This pressure reduction represents a 296 Bph energy loss. To reduce the power loss across the reducing valve, booster pump B is used to increase the plant water pressure from 120 ft head to 240 ft head. Pump B operates at 120 ft head, 4000 gpm, 82% efficiency, and 148 Bhp. This allows the primary pumps A to operate at the lower pressure point A′ by reducing the impeller diameter and by removing the pressure reducing valve. Each Pump A′ will operate at 6000 gpm, 120 ft head, 75% efficiency, and will draw 242 Bhp. Total power for the system with the booster pump will be 632 Bhp compared with 886 Bhp for the original system. This 254 Bhp reduction represents $76,200/yr power saving with $.045/kWh power. The savings will justify a field retrofit of a booster pump and new smaller impellers for Pump A. If the system had been designed originally with a booster pump, additional savings would have been achieved by selecting a more efficient pump to operate at point A′.

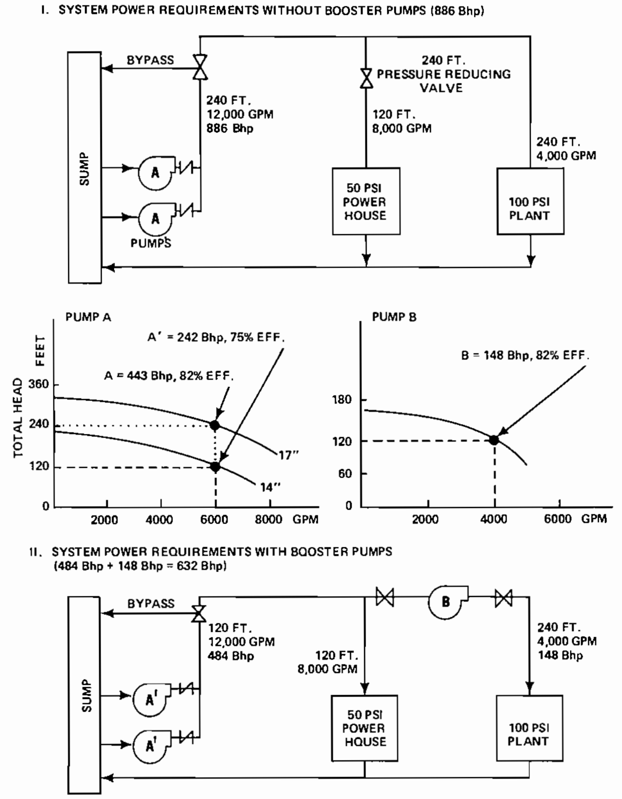


Figure 13. Small Booster Pump Reduces Total System Power — Saves 254 Bhp.

11. *Recover Power Using a Pump as a Turbine*

High pressure water or process fluid let down to a lower pressure through an orifice or control valve is a source of frequently wasted energy. If this high pressure liquid were passed through a hydraulic turbine, much of the wasted energy could be converted to shaft horsepower. Many standard centrifugal pumps will operate efficiently in reverse as hydraulic turbines. This allows low cost standard centrifugal pumps to be used as power recovery turbines. The turbine output can be used to directly drive pumps, other process equipment, or an electrical generator. Since the pump has no speed regulating mechanism when used as a turbine, speed will vary with the head and flow. However, by slightly undersizing the pump/turbine and by using an auxiliary power source such as an electric motor to drive the hydraulic turbine through a double-ended shaft, a constant output speed can be obtained for variable inlet head and flows. Hydraulic power recovery turbines are best applied for capacities greater than 100 gpm and pressure greater than 100 psi. For smaller horsepower applications, the reduced efficiency of the hydraulic turbine makes power recovery difficult to justify economically. Since turbine efficiency and output fall off the head or flow are reduced, this technique is best applied to relatively constant head and flow situations. Turbine power output typically falls to zero as flow is reduced to 35% to 40% of best efficiency capacity. As a first approximation in selecting a pump to operate as a turbine, one may assume the pump will operate with the same efficiency as either a pump or turbine for a given head and flow. However, the pump vendor should be consulted before making a final selection. Potential energy savings are large for this method. Because of the large equipment investment required for a turbine/pump, motor/generator, and controls, this energy saving technique is best applied in the design stage.

Example 11

Standard centrifugal pumps can be used as hydraulic turbines to recover power typically lost as pressure drop across a valve. Figure 14 shows such a power recovery technique applied to a high pressure scrubbing system in a chemical process. The chemistry requires that 2250 gpm process fluid be supplied to a scrubber at 400 psi. The fluid exits the scrubber at 365 psi, where it can be pressure reduced through either a turbine or a control valve. The process is a constant pressure and flow operation which can be met by a pump operating at 2250 gpm, 900 ft head, 82% efficiency, and 624 Bhp. A 700 hp electric motor was selected to drive the pump. By using a pump with a double-ended shaft as a power recovery turbine, 57% of the total pumping power can be recovered from the recycled process fluid. The electric motor provides full start-up power, and as the turbine picks up the load, the motor load is reduced. The motor also controls the speed of the turbine and pump. In this case, the pump running as a turbine will supply 354 Bhp from 820 ft head and 2250 gpm flow. At \$.045/kWh power cost, the power recovered represents savings of over \$106,000/yr.

12. *Limit the Use of Lower Efficiency Specialty Pumps*

Pump energy consumption can be reduced in the design stage by limiting the use of specialty pumps. Many special purpose pumps have lower efficiencies than standard centrifugal pumps. Occasionally, unusual pumping system requirements lead to the use of a nonstandard pump. However, each use should be carefully evaluated to determine if the special pump is necessary and worth the loss in efficiency. The more common types of special pumps are discussed below.

a. Self-priming pumps are 10% to 20% less efficient than comparable horizontal or vertical centrifugal pumps. The self-priming pump has built-in chambers in front of the impeller which trap sufficient liquid to allow the pump to start repeatedly. Friction loss and recirculation in these chambers lead to the high loss in efficiency. Standard horizontal pumps with priming systems or vertical pumps should be considered in place of self-priming pumps.

b. Solids handling pumps for use with slurries, paper stock, sewage, etc., are typically 2% to 10% less efficient than standard centrifugals. Slower running speeds, larger clearances, and nonclog impellers account for the efficiency loss. Standard centrifugal pumps will handle some solids, and it may be more economical to unclog an efficient pump occasionally instead of using a less efficient solids-handling pump.

c. Canned motor pumps are typically 5% to 15% less efficient than standard centrifugal pumps. They are used where zero leakage is demanded, where a pump

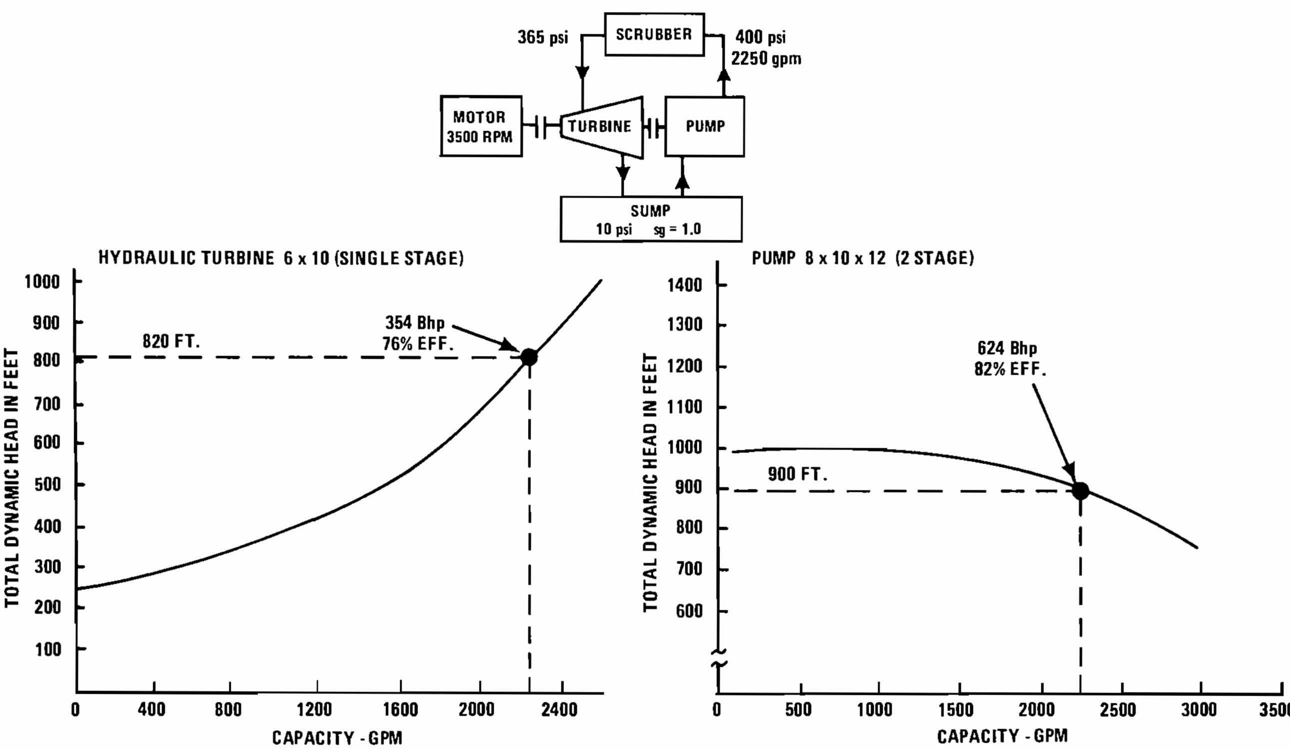


Figure 14. Recover Power Using Pump as a Hydraulic Turbine — Saves 354 Bhp.

seal failure could cause disaster, or for high suction pressure appications.

d. Air operated pumps, either air operated diaphragm or air motor driven, tend to have higher operating costs because plant air is more costly per horsepower than electricity.

Limiting the use of specialty pumps is not a widely applicable energy conservation method and is best applied at the design stage. However, sizeable energy savings can be made under the proper conditions. Occasionally, even a retrofit of existing equipment may be justified.

Example 12

Compare the preformance of two specialty pumps (self prime and slurry) with two standard centrifugal pumps (horizontal and vertical) in identical service. Figure 15 shows these four pumps sized for 500 gpm and 120 ft head. The standard ANSI horizontal and vertical pumps operate equally at 73% efficiency and 20.8 Bhp. The self-prime pump operates at 53% efficiency and draws 28.6 Bhp. This represents a 38% increase in power consumption and would cost $2,340/yr (power at $.045/kWh) more to operate than a vertical pump. The slurry pump operates at 63% efficiency and requires 16% more power to operate than a similar horizontal pump. This extra 3.3 Bhp represents an additional $1,000/yr to operate the slurry pump.

13. *Avoid Gas Entrainment*

Gas entrainment in a pump can lead to large capacity, head, and efficiency losses. Entrainment can occur from air or gas being introduced into the pump at the suction in a number of ways: vortexing, insufficient submergence depth, improper location of a bypass line, poor sump design, gas released from the process, or air leaks in the piping system. For whatever the reason, the results are the same. As little as 1% to 2% gas entrained in a centrifugal pump can lead to a 3% to 15% reduction in head and a large efficiency loss.

Measuring the amount of entrained gas is very difficult; however, the problem is usually readily apparent from a large reduction in head. If analysis of a problem pump system shows that the pump is not developing the rated head and inspection of the pump reveals no mechanical problem, suspect gas entrainment. Correction of the problem usually involves modification of the pump inlet piping and inlet tank by increasing submergence, changing sump design, using vortex breakers, and other means.

Example 13

To illustrate the effect of entrained gas on pump performance, consider a pump designed for handling 16,000 gpm of 1.3 specific gravity liquid at 40 ft head. The liquid will contain 10% gas released from a chemical reaction. From actual performance tests, the pump was found to operate at 16,000 gpm, 117 ft head, 82% efficiency, and 750 Bhp

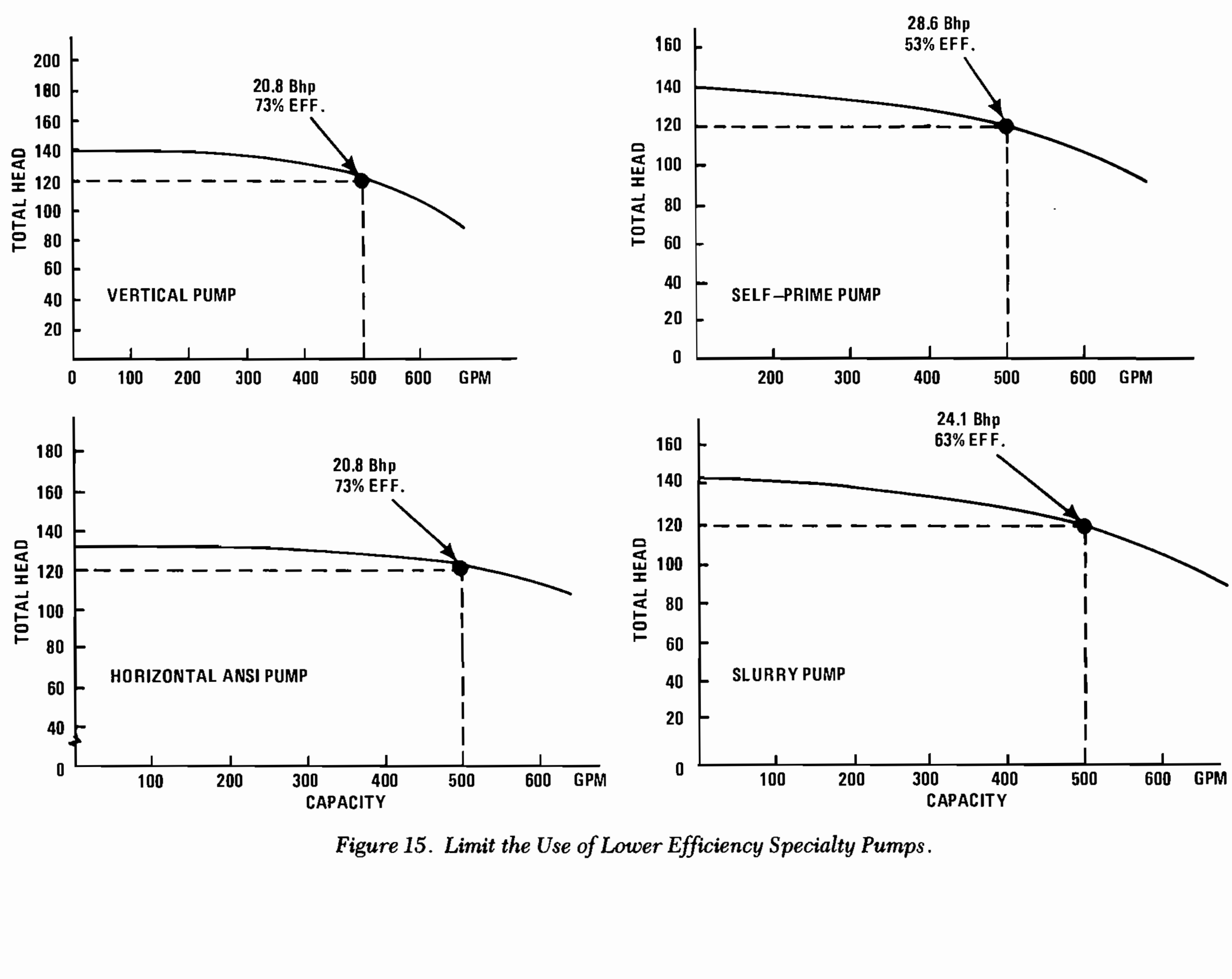


Figure 15. Limit the Use of Lower Efficiency Specialty Pumps.

when there is no entrained gas. When running on 10% entrained gas, the pump performance was reduced to 40 ft head, 16,000 gpm, 35% efficiency, and 540 Bhp. With a smaller impeller and no entrained gas, the pump could operate at 16,000 gpm flow, 40 ft head, 80% efficiency, and 263 Bhp. The effect of the entrained gas increased the power required by 277 hp (263 hp to 540 hp) and increased power costs by over $83,000/yr.

14. *Use More Efficient Motors*

Since most pumps are motor-driven, energy savings can be made by avoiding inefficient operation of pump drive motors and by using more efficient motors. Motor efficiency can be improved by avoiding part load operation. First, variable flow rates may lead to extended operation at reduced flow rates and reduced motor loads. Second, oversized pumps that are throttled back or pumps with smaller impellers may have been sized for a motor to be nonoverloading at the maximum possible pump power requirement. Both cases can result in an oversized motor operating at part load. For part load operation between half and full power, no motor efficiency penalty is suffered by using the oversized motor instead of a smaller motor. The increase in motor efficiency as motor size increases offsets the decrease in motor efficiency at half load (see Figure 16). Thus a 20 hp motor can operate as efficiently at 10 hp as a 10 hp motor. However, motor efficiency falls off rapidly below half load so that oversized motors running at less than half load are much less efficient than properly sized motors (note the decreased efficiency of the 40 Bhp motor operating at 10 Bhp in Figure 16). Although reduced motor efficiency becomes a factor below half load operation, the energy savings can rarely justify replacement of existing motors.

Many recent publications point out the potential savings that could result from using "high efficiency" motors. In reality, only modest energy savings can be realized from using the higher efficiency motors manufactured today. Figure 16 represents the typical efficiencies of NEMA B motors suitable for chemical industry service. The greatest potential for improving these efficiencies by using "high efficiency" motors is in the 1 hp to 20 hp size range. Above 20 hp, the anticipated efficiency gains grow smaller; existing motors over 200 hp are already relatively efficient.

A high efficiency motor does not substantially differ from a conventional electric motor. Efficiency improvements result from reducing motor losses by: 1) adding more copper in the windings to reduce I^2R losses, 2) using higher quality steel and thinner laminations to reduce core losses, 3) using higher quality components to reduce windage and friction losses, and 4) using optimal slot and air gap design to reduce stray load losses. In addition to improving efficiency, these improvements can increase manufacturing costs, decrease reliability and decrease other motor performance characteristics.

Example 14

To illustrate the savings available from using a more efficient motor, let us look at a fully loaded, 1800 rpm, 20 hp motor. An existing 20 hp motor has a 90% efficiency and a high efficiency motor has a 92% efficiency. Using $.045/kWh power and 8,000 hr/yr operation, the existing motor has an annual power cost of $5,968/yr and the high efficiency motor has an annual operating cost of $5,838/yr. In this case, the high efficiency motor can save $130/yr which can justify a premium price for the high efficiency motor, but it will not justify replacing the existing motor.

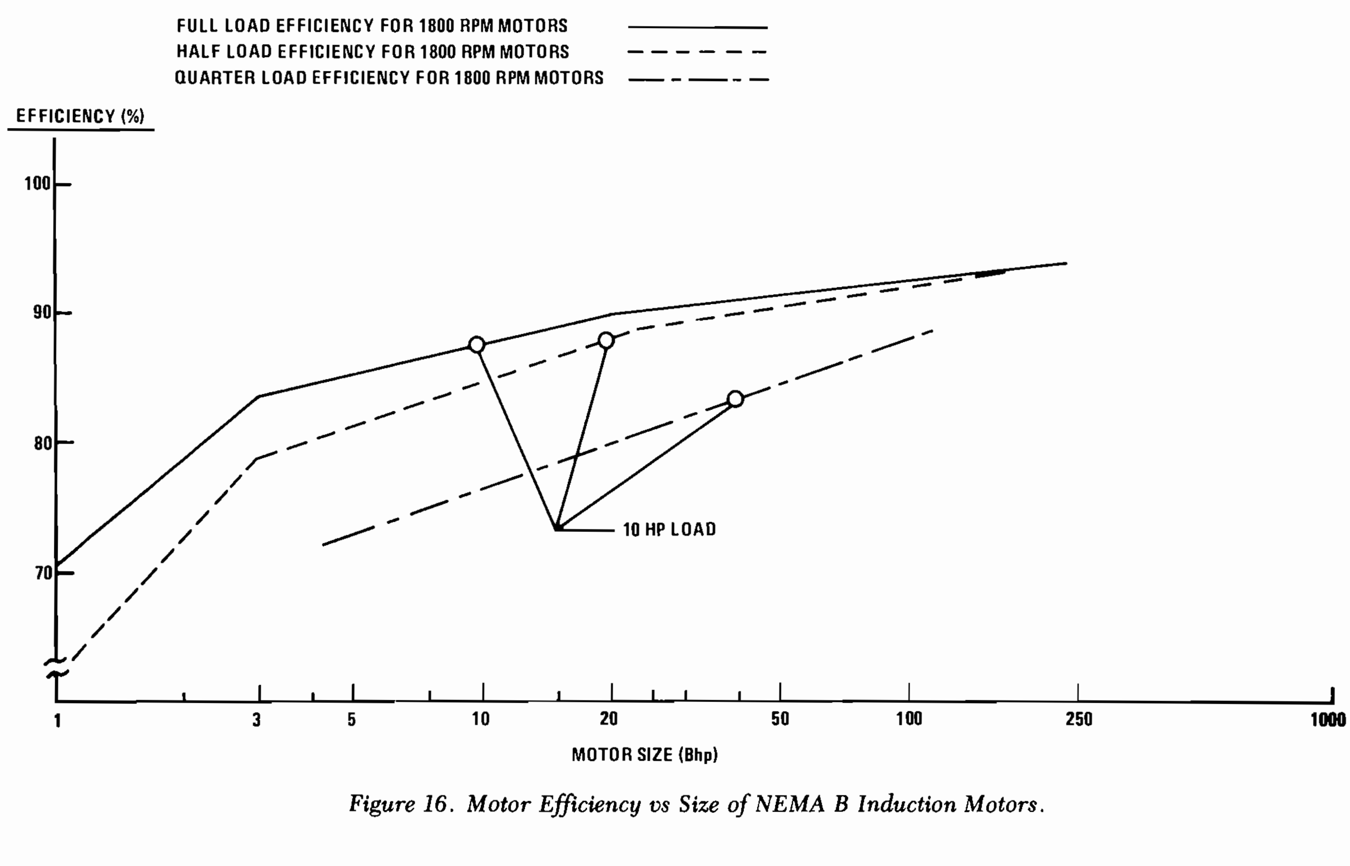


Figure 16. Motor Efficiency vs Size of NEMA B Induction Motors.

15. *Eliminate Pump Seal Cooling to Save Energy*

Energy savings can be achieved under certain conditions by eliminating pump seal cooling and/or flushing. Energy costs associated with pump seal cooling and flushing generally fall in these two categories: 1) cost of reheating the process fluid cooled by seal cooling or flushing, and 2) cost of providing the water for pump jacket and seal cooling. When a cool seal flush (either water or cooled pumpage) flows into the process, heat must be added to maintain constant process temperature. Additional heat must also be added to the process to make up for the energy removed by the pump jacket cooling water temperature rise.

Where pump seal cooling or flushing is not essential, such as in pumping clean, high-boiling liquids, it can be shut off and the pump run hot [6]. With proper materials, conventional mechanical seals can run to 500°F, and metal bellows seals can operate to 800°F without cooling. If several of the following conditions pertain, further analysis should be made to determine if seal cooling and/or flushing can be eliminated:

a. Pump handles hot liquids above 300°F.
b. Process pumpage is maintained at an equal or greater temperature downstream of the pump.
c. The liquid pumped is not near its boiling point.
d. Pump, seals, or bearings are water cooled.
e. The pump seal is cooled and flushed with pumpage cooled by an external heat exchanger.
f. The pump seal is flushed with an external flush.
g. Seal flush liquid must be removed from the process later.

Flushing and/or cooling is necessary in many cases and should not be eliminated without a detailed analysis. Flushing with clean pumpage or with an external source of clean liquid is used to keep dirty, abrasive process liquid away from the seal faces to extend seal life. Cooling is usually required when handling liquids near the boiling point to keep the temperature at the seal face below boiling. If the liquid boils in the pump seal cavity, the seals would run dry and rapidly fail.

Example 15

Figure 17 shows a pump handling Dowtherm (sg=.8, C_p=.625) at 600°F. 2 gpm of pumpage is cooled to 400°F and used to flush a conventional mechanical seal. 20 gpm of cooling water is used with a 10°F temperature rise in the seal flush heat exchanger. An additional 4 gpm of jacket cooling water increases 15°F as it cools the seal chamber. It is desired to determine the potential savings resulting from: 1) the replacement of the conventional cooled seal by a metal bellows seal, 2) flushing with hot pumpage, and 3) turning off the seal chamber cooling water. Assume 8,000 hr/yr operation, $5.00/$10^6$ Btu fuel cost, 80% heater efficiency, and cooling water cost at $.05/1,000 gal.

1. Cost to reheat process fluid cooled for seal flush.

$$E = mc_p\,(T_1 - T_2)$$

where
E = energy, Btu
m = mass of seal flush pumpage
c_p = specific heat of pumpage
$T_1 - T_2$ = temperature drop of pumpage
sg = specific gravity of pumpage

$$m = 2\,\frac{gal}{min} \times 6.67\,\frac{lb}{gal} \times 60\,\frac{min}{hr} \times 8000\,\frac{hr}{yr} = 6.4 \times 10^6\,\frac{lb}{yr}$$

$$E = 6.4 \times 10^6\,\frac{lb}{yr} \times .625\,\frac{Btu}{lbm°F} \times 200°F = 800 \times 10^6\,\frac{Btu}{yr}$$

$$Cost = 800 \times 10^6\,\frac{Btu}{yr} \times 5\,\frac{\$}{10^6\,Btu} \times \frac{1}{.8}\text{ heat eff.} = \$5{,}000/yr$$

2. Cost to reheat process fluid cooled by seal jacket cooling water.

Note the quantity of heat removed from the pumpage equals the heat added to the cooling water. Cooling water temperature rise = 15°F.

$$m = 3\,\frac{gal}{min} \times 8.33\,\frac{lb}{gal} \times 60\,\frac{min}{hr} \times 8000\,\frac{hr}{yr} = 12 \times 10^6\,\frac{lb}{yr}$$

$$E = m\,c_p(T_1 - T_2) = 12 \times 10^6\,\frac{lb}{hr} \times 1.0\,\frac{Btu}{lbm°F} \times 15°F = 180 \times 10^6\,\frac{Btu}{yr}$$

$$Cost = 180 \times 10^6\,\frac{Btu}{yr} \times 5\,\frac{\$}{10^6\,Btu} \times \frac{1}{.8}\text{ heat eff.} = \$1{,}125/yr$$

3. Cost of cooling water for seal jacket (3 gpm) and heat exchanger (20 gpm).

a. Use filtered raw water which is reused.
b. Use .05 $\frac{\$}{1000\,gal}$ cost of pumping water.
c. Annual cost $= .05\,\frac{\$}{1000\,gal} \times 60\,\frac{gal}{hr} \times 8000\,\frac{hr}{yr} = 24\,\frac{\$}{gpm - yr}$

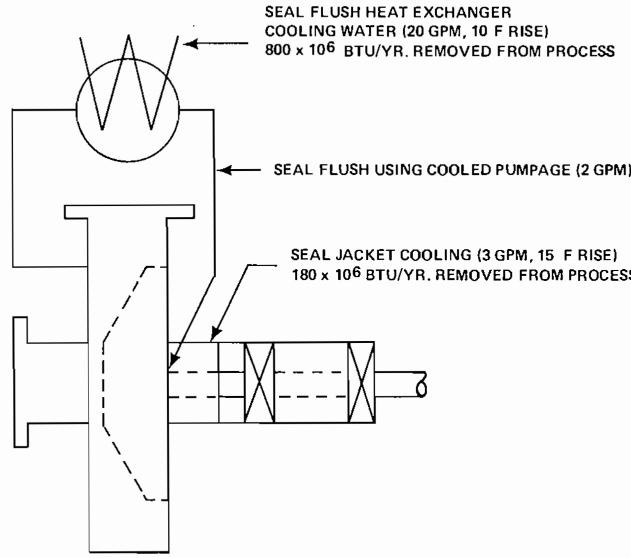


Figure 17. Eliminate Pump Seal Cooling — Saves 980 × 10^6 Btu/yr in Process Heating, Saves 11 × 10^6 GPM in Cooling Water.

d. Annual water cost $23 \text{ gal} \times 24 \frac{\$}{\text{gpm} - \text{yr}} = \$552/\text{yr}.$

In this example, the total energy savings (sum 1, 2, and 3) by eliminating seal cooling and flushing comes to over $6,600/yr.

16. *Minimize Losses from Mechanical Seals and Packing*

Mechanical seals and packing represent minor power losses in a pump; therefore, there is little potential to reduce pumping energy. Except for specialty pumps such as diaphragm pumps, canned motor pumps, or sealless magnetic drive pumps, all pumps are sealed with either mechanical seals or packing. Figure 18 shows the typical horsepower losses from single unbalanced mechanical seals [7]. These losses are generally small in comparison to total pump power and can usually be neglected in a pump energy saving analysis. Note that seal losses increase with increasing size, speed, and stuffing box pressure. Using balanced seals reduces the horsepower required at higher pressures. Using double seals increases the power losses. The power losses from packing are extremely variable and depend on many factors such as material, number of rings, size, tightness, lubrication and temperature. With very tight packing, the sealing power losses can be up to six times greater with packing than with a balanced seal. Because of the extreme variability of packing power losses, energy saving alone cannot justify changing from packing to mechanical seals. However, the other advantages of mechanical seals over packing (less leakage, longer life, increased reliability) have led to their use in almost all pumping applications.

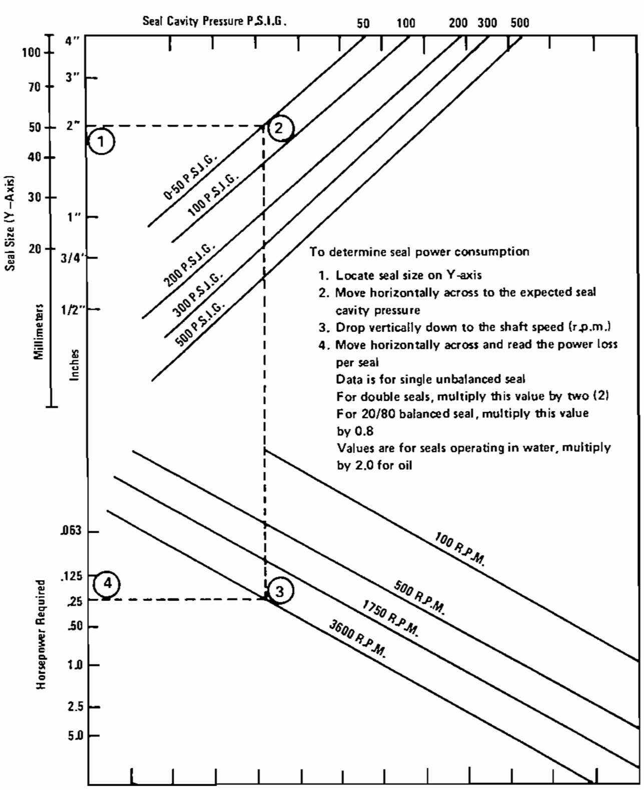


Figure 18. Power Losses from Pump Seals.

FIELD METHODS

17. *Shut Down Unneeded Pumps*

This energy-saving technique seems obvious, but it is often overlooked. For instance, consider a process fed by parallel pumps that were run throttled back so far that one pump alone could handle the flow; or four multiple cooling tower pumps being run where three pumps would meet the demand; or running a single pump to maintain header pressure when the process is down for a shift or the weekend. This method often costs nothing to apply except a conscious effort on the part of the operations to know what their actual process demands are. It can be applied wherever multiple pumps are found and can result in potentially large savings.

Example 17

In a process requiring 500 gpm at a constant 200 ft head, two identical pumps in parallel were originally supplied, one operating and one installed as a spare. In actual practice, both pumps were run throttled back to 250 gpm at 235 ft head so that if one pump failed, the other pump could instantly take up the load. However, the plant was paying a very high price for the flexibility of not even a momentary interruption in flow. As shown at point A, Figure 19, a single pump delivering 500 gpm at 200 ft head would operate at 70% efficiency and require 36 Bhp. With both pumps throttled to half capacity (point B), each pump operates at 250 gpm, 235 ft head, 57% efficiency, and 26.0 Bhp for a total of 52 Bhp. Thus, 16 Bhp could be saved by shutting down one pump whenever possible. With electric power costing $.045/kWh, this represents a potential savings of $4,800/yr. Additional savings would result from improved pump life since a pump throttled at reduced capacity will generally have a shorter life than one operating near best efficiency.

18. *Properly Maintain Existing Pumps*

Substantial energy savings can be obtained by replacing worn pump parts to minimize internal leakage and to maintain pump efficiency. Pump capacity, head, and efficiency are reduced as pump internal leakage increases from excessive backplate and impeller clearances and worn impeller wear rings and impellers.

Efficiency loss from wear depends on impeller type and specific speed. For equal wear clearances, open impeller pumps show a larger loss in efficiency than closed impeller pumps. Open impeller pumps typically have initial clearances of .015-.020 in. However, as these clearances increase from wear over the initial clearance, pump efficiency is reduced. Head and capacity are also reduced as more and more of the pump's capacity is recirculated through the clearance. Figure 20, from Karassik [8], shows that as specific speed decreases, leakage losses increase. This is not surprising since low specific speeds indicate high head and low flow. Because pump clearances are nearly the same on big pumps as on small pumps, internal leakage is a much greater percentage of the small pump output and this leads to a corresponding reduction in pump efficiency.

Although the savings are not the same for every case, pumps with the following characteristics will benefit the most from special attention to maintenance:

- Pumps with open impellers.

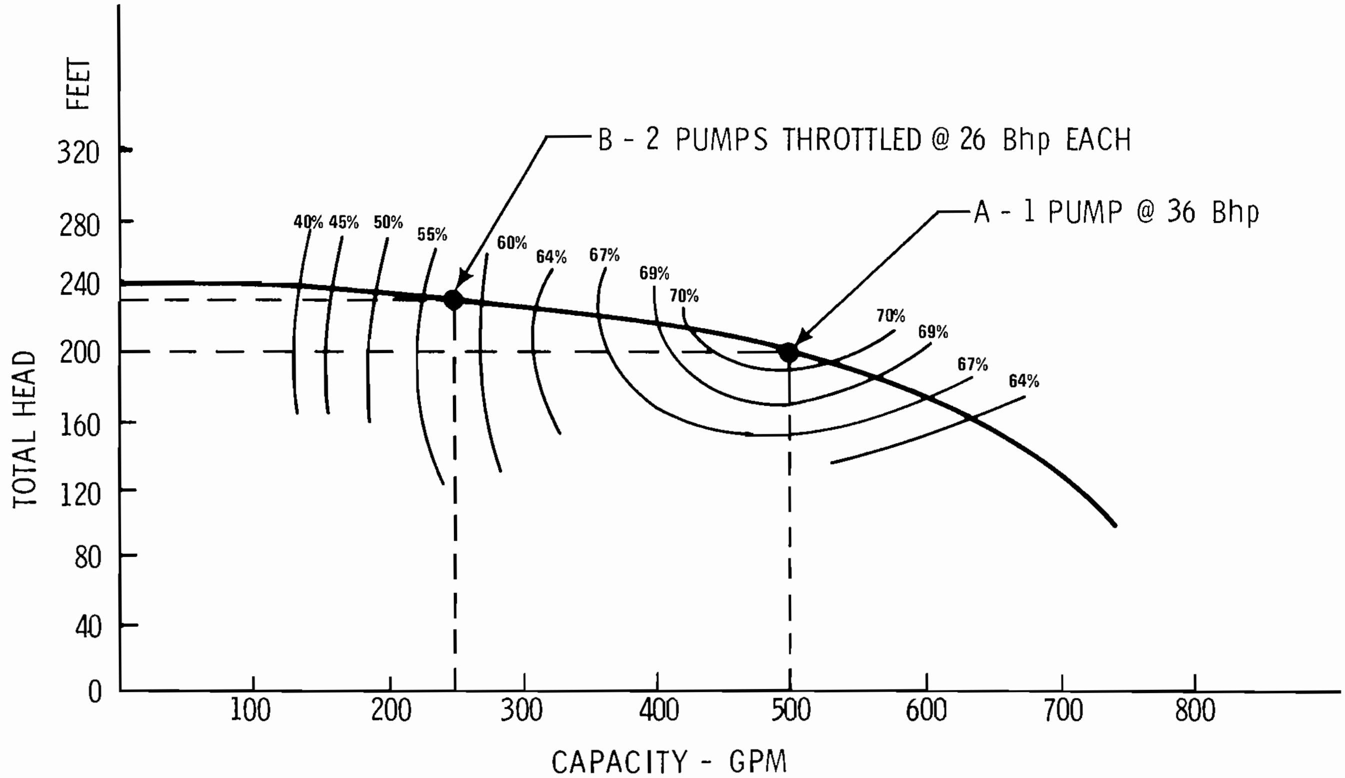


Figure 19. Shut Down Unneeded Pumps — Saves 16 Bhp.

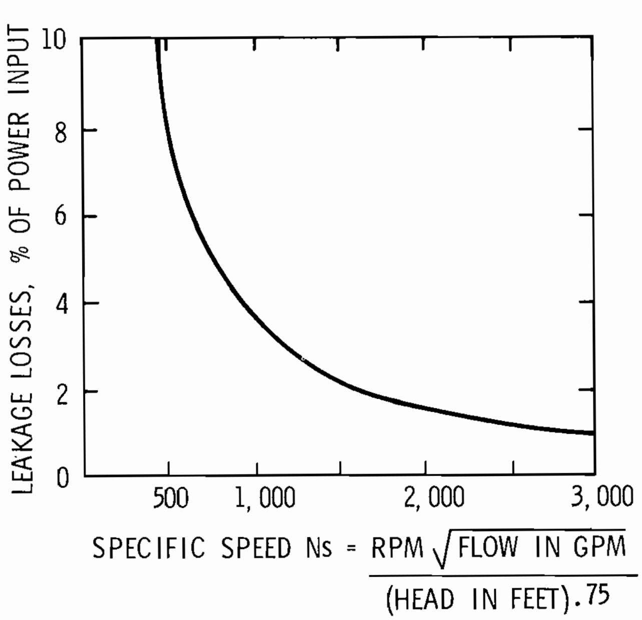


Figure 20. Energy Losses from Internal Clearances Vary with Specific Speed.

- Pumps which operate at high heads (over 100 ft) and low flows (below 500 gpm).
- Pumps on abrasive or corrosive service.

By concentrating on pumps with these factors and limiting impeller and wear ring clearance, power reductions of 4% to 6% per pump can be expected. Wearing parts (impellers, wear rings, bushings, and bearings) should be adjusted or replaced before the clearances become excessive. The maintenance interval should be determined by experience on each particular installation.

Example 18

To show the cost of excessive pump internal clearances, consider the single stage, open impeller ANSI pump supplying a process requirement of 300 ft head and 300 gpm flow (see Figure 21). With an initial .015 in impeller

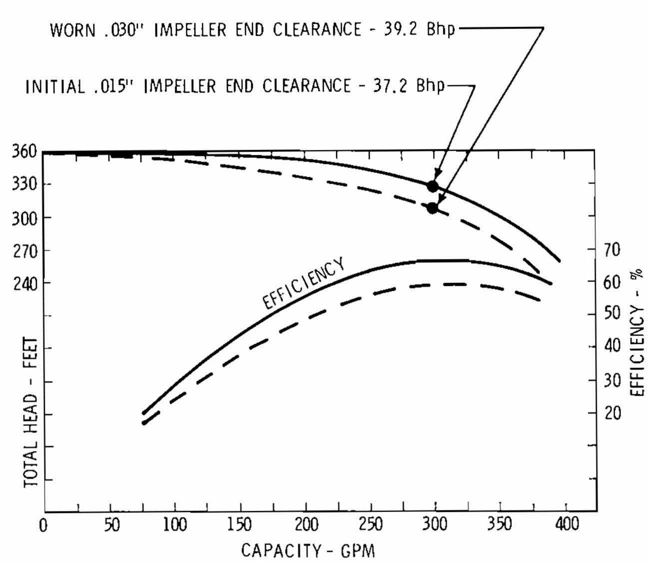


Figure 21. Maintain Pumps by Restoring Impeller Clearances — Saves 2.0 Bhp.

end clearance, the pump operates slightly throttled at 325 ft head, 300 gpm flow, 66% efficiency, and 37.2 Bhp. As impeller clearances wear, the head, capacity, and efficiency are reduced as shown. With .030 in impeller clearance, the pump operates at 305 ft head, 300 gpm, 59% efficiency, and 39.2 Bhp. Restoring the impeller clearance to .015 in would reduce power consumption by 5% and save 2.0 Bhp. With $.045/kWh power, this would save $600/yr in power costs.

19. *Reduce Excess Head by Trimming Impellers*

Trimming centrifugal pump impellers is the lowest cost method for correcting oversized pumps. Oversizing of pumps occurs for many reasons: 1) a wide range of pressure and flow requires a pump sized for the worst case, 2) allowances are made for future capacity, 3) large factors of safety are applied to "guarantee" a pump will provide required pressure and flow, and 4) operating conditions are different than design conditions. Whatever the reason for oversizing, the results are the same. The pump puts out more flow at a higher head than is required and this wastes energy. When a pump impeller diameter is trimmed, the flow is reduced proportional to the impeller diameter, the pump head is reduced as the square of the impeller diameter, and the power is reduced as the cube of the impeller diameter.

When can trimming be applied? Trimming pump impellers should be considered for any centrifugal pump. Look first for pumps suspected of pumping more flow or are operating at a higher pressure than is required. Second, look for pumps with excess capacity that are throttled back by a valve (either a manual or an automatic control valve). A quick look at the vendor's pump curve will show the range of impeller diameters available for the particular pump.

Typically, a 10% to 50% reduction in head can be achieved by changing pump impeller diameter within the vendor's recommended size limits for the pump casing. Either the existing impeller can be trimmed to a smaller diameter or a new smaller diameter impeller can be ordered and the original impeller stored for future use should the system resistance increase. Since the cost of trimming and balancing an impeller is small (generally under $1000), payback is generally measured in months instead of years. Where demand is seasonal, like a cooling water pump, it often pays to use a small diameter impeller in the winter and a larger diameter impeller in the high demand summer months. One can apply the same principle of reducing excess head to multistage centrifugal pumps (horizontal or vertical) by removing excess pump stages.

Example 19

Let us see how energy can be saved by simply trimming an impeller. A double suction centrifugal pump with a 13.75 in diameter impeller is used to pump process water. Demand is a constant 2750 gpm, and the pump is controlled by a manual throttle valve. A total head of 164 ft is measured and the pump is found to operate at 164 ft head, 2750 gpm, 83% efficiency, and 137 Bhp (Figure 22, point A). Pressure measurements show a 37 ft (16 psig) pressure drop across the partially closed throttle valve at 2750 gpm flow and only a 6 ft drop across the fully open valve. If the pump were exactly matched to the system requirements, only 127 ft of head would be required without the valve. Since even the fully open valve has a 6 ft pressure drop, the minimum head required is 133 ft. To this head, a 5% allowance should be added as a tolerance for the accuracy of the field measurements and impeller trimming operation. This brings the minimum total head required to 140 ft. Applying the pump affinity laws gives the trimmed impeller diameter to be 13 in. Note that both the head and flow are reduced as the impeller is trimmed.

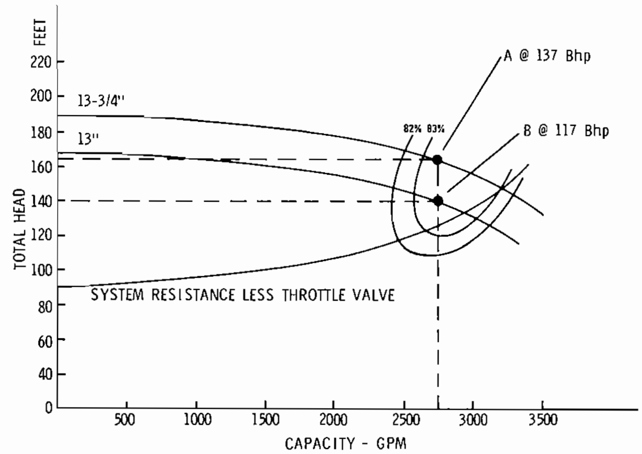


Figure 22. Trim Impeller to Reduce Excess Head — Saves 20 Bhp.

With a trimmed 13 in impeller, the pump will operate slightly throttled at 140 ft head, 2750 gpm, 83% efficiency, and 117 Bhp (Figure 22, point B). With $.045/kWh power, the trimmed impeller reduces power consumption 20.0 Bhp and saves $6,000/yr. With trimming and balancing an impeller, typically costing less than $1,000, the corrective action will pay for itself in less than two months.

20. *Reduce Excess Flow With Proper Impeller Selection*

Often different pump impellers can be obtained which will change the pump operating characteristics. These impellers can extend or reduce a given pump's head or flow range as much as 10% to 50%. The different impellers are generally designated high head or low head impellers or high flow or low flow impellers. These impellers are used to match system pumping requirements using existing pumps. The use of special impellers is much more limitd than changing impeller diameters. Presently, special impellers are available for only a limited number of pumps such as vertical turbine pumps and very large centrifugal pumps. However, on pumps where different impellers are available, their use can extend the pump operating range and better utilize existing equipment. In addition to improved efficiency, a different type impeller may offer lower NPSH or a steeper head-flow slope for improved control. Use of high or low capacity impellers should be considered on a case-by-case basis as a means of matching the pump to the system.

Example 20

To demonstrate the effect of proper impeller selection, consider the large cooling water pump in Figure 23 which was initially designed to deliver 30,000 gpm at 140 ft head (point A). A high and low flow impeller are offered for this

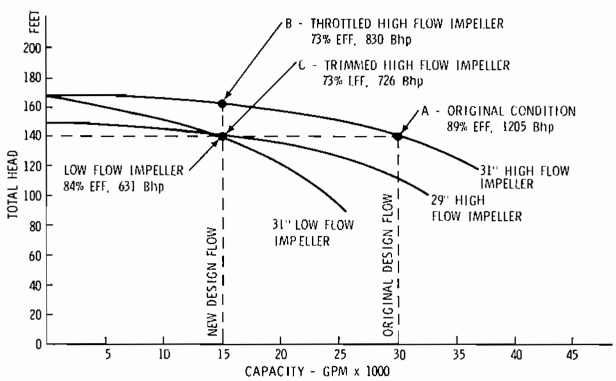


Figure 23. Reduce Excess Capacity with Proper Impeller Selection — Saves 199 Bhp vs Throttled High Flow Impeller, Saves 95 Bhp vs Trimmed High Flow Impeller.

pump. The operating conditions have changed, and now the pump must operate at 15,000 gpm and 140 ft head. Continued operation at point A (30,000 gpm, 140 ft head, 88% efficiency, and 1205 Bhp) while bypassing the excess flow would be too energy wasteful to consider. Some energy could be saved by throttling the pump back to point B (15,000 gpm, 150 ft head, 73% efficiency, and 830 Bhp). Additional energy could be saved by trimming the high flow impeller diameter to operate at point C (15,000 gpm, 140 ft head). However, the trimmed high flow impeller would operate at only 73% efficiency and use 726 Bhp. Maximum energy savings can be achieved by using the low flow impeller operating at point C. The low flow impeller would operate with 84% efficiency and 631 Bhp. The low flow impeller saves 95 Bhp over a properly trimmed high flow impeller and hundreds of horsepower over throttling or bypassing with the original impeller. At $.045/kWh, the 95 Bhp reduction represents $28,500/yr power savings.

SUMMARY

Significant energy savings can be made by systematically applying existing technology to reduce pump energy consumption. All the pump energy conservation techniques described in this report are not applicable in every case, but each technique has been used successfully in numerous applications. Collectively, these techniques provide a comprehensive package that can be successfully applied to pump energy conservation problems.

Many of the energy conservation methods are applicable at the design stage, and energy savings principles are most appropriately considered as a part of the design process. Often, energy savings measures which are expensive as field retrofits can be justified in the initial design at little or no extra cost.

Other corrective measures are simple, low cost techniques that can be readily applied to existing installations. Since the potential energy savings are distributed over a large number of pumps, a systematic plantwide survey is suggested as the best way to apply these techniques to existing installations.

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