energy design resources

design brief

CENTRIFUGAL PUMP APPLICATION AND OPTIMIZATION

A pump that is well-sized, -specified, and -selected, and is operated in a manner that reflects the design intent will be more efficient and often costs less than a pump that does not fit the needs of the system it serves.

Who Should Read This Brief?

This brief presents practical pump theory, selection and application principles. The concepts discussed will be of interest to:

- Design professionals who want to better understand how pump theory and design practice apply to day-to-day operating environments
- Commissioning providers who wish to better understand pumps, pumping systems, and the evolution of design intent for hydronic systems
- Facilities engineering professionals who want to better understand the fundamental design and application principles behind the equipment they operate

Key points for each of these areas of professional practice are emphasized in checklists at the end of each major topic.

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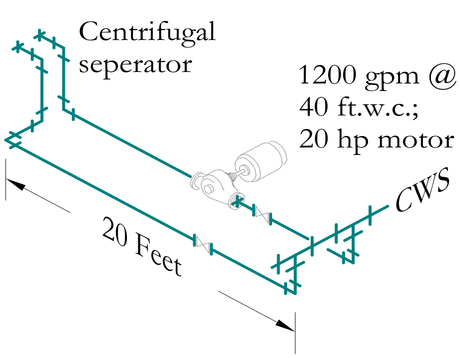
Summary

Centrifugal pumps perform many important functions to control the built environment. The physics and basic mechanics of pumps have not changed substantially in the last century. However, the state of the art in the *application* of pumps has improved dramatically in recent years. Even so, pumps are still often not well applied, and become common targets in retrocommissioning projects where field assessment and testing can reveal significant energy savings potential from optimizing pump performance. Typically, retrocommissioning finds that pump flow rates do not match their design intent and that reducing flow rates to match load requirements or eliminating unnecessary pressure drops can save energy. As the example below illustrates, decisions made during the design phase have implications throughout the operating life of the building.

Although fully optimizing any design will require some effort after installation, the prevalence and magnitude of the savings that are commonly found in retrocommissioning and ongoing commissioning begs the larger question: How much greater would the savings be if pumps were selected and applied optimally during the design phase?

A Common RCx Problem Reveals Missed Opportunity for Rightsizing

The pumping application shown below illustrates a typical less-than-optimal design that often translates into a retrocommissioning opportunity later on.



The pump selected for the application was rated for 40 feet water column (ft. w.c.) of head at the design flow rate, even though the piping circuit it served was very short with very few fittings. Field testing revealed that only 13 to 14 ft. w.c. of head was actually required. During RCx, the pump was optimized to the requirements of the system via an impeller trim and minor throttling. However, many other opportunities were missed because the system performance was not optimized during the design phase. If it had been, the pump selection would have more closely matched the needs of the system and would:

- Operate at its best efficiency point in the as-built system;
- Be smaller and cost less to purchase and pipe;
- Require a 10 hp motor and electrical service instead of 20 hp.

This brief tries to answer that question by exploring the theory and reality behind the selecting and applying centrifugal pumps in commercial buildings. You will learn that if a pump is carefully specified during the design process and is installed in a manner that reflects the design intent, it will be more efficient and will most likely cost less than a pump that does not fit the needs of its system. This is especially true when the "ripple effect" savings in the supporting drive and power systems are taken into account.

This brief describes how pumps use energy, and discusses various efficiency considerations. It then describes how the entire system affects the performance of the pump, and discusses how to select the best pump and components. A section on optimizing the performance of a pump system describes important considerations in sizing pumps, as well as how to adjust the pump's flow initially and over time to ensure optimal performance over a range of conditions.

Pump Energy

Pumps provide the energy to move water through piping systems including equipment, piping, and fittings, and through elevation changes in open systems. If the flow rate, head requirement, and efficiency for a pump are known, then the pump-power equation can be used to calculate the brake horsepower required to perform the task as follows:

$$BHP = \left(\frac{Q \times H}{3{,}960 \times \eta_{Pump}} \right)$$

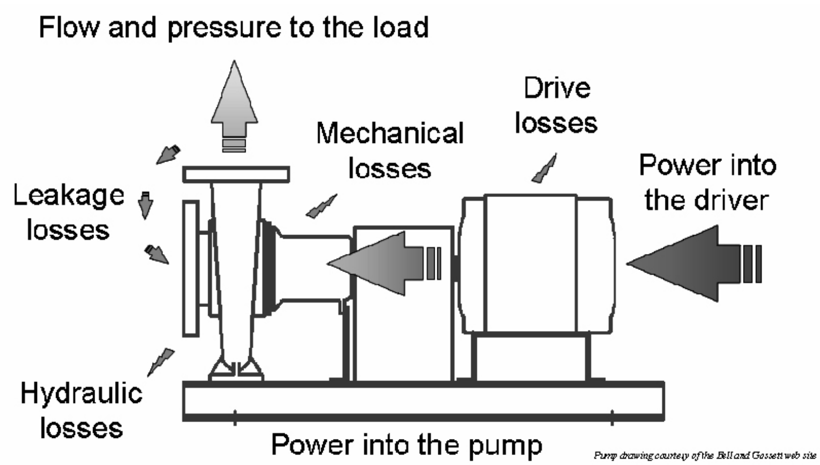
Where:

BHP	=	*Brake horsepower required at the pump shaft*
Q	=	*Pump flow rate in gpm*
H	=	*Pump head in feet of water column*
η_{pump}	=	*Pump efficiency (typical range = 0.45 to 0.85)*

Dividing the result of the pump-power equation by the motor efficiency converts the brake horsepower requirement at the motor output shaft into horsepower input to the motor. Multiplying the motor horsepower by 0.746 kW per horsepower converts this input power to electrical power. This value can then be multiplied by the hours of operation to determine how much energy the pump consumes in kilowatt-hours (kWh) for constant flow systems. The different terms used for power in pumps are summarized in **Sidebar 2**.

Figure 1 shows that the amount of power that exits a pump in the form of flow and pressure is less than the power that enters the pump from the utility system, due to losses in the drive system and the pump. These losses include drive-system losses, mechanical losses, leakage losses, and hydraulic losses. These losses, and opportunities to minimize them through proper selection, application, and specification are described below.

Figure 1: Efficiency Losses Through a Pump and Its Drive



Power Terms Used in Pump Systems

Several related terms are used to describe the power used by or generated by a pump:

- *Brake Horsepower (BHP):* Brake horsepower is the power required as an input to the pump shaft to deliver the indicated flow and head. Brake horsepower takes pump efficiency into account but not motor efficiency. To account for losses in the motor and drive system, it is necessary to take the brake horsepower requirement and divide it by the motor and drive efficiencies at the designated operating point.
- *Motor Power:* The output power delivered by a motor shaft to a driven device. This is the motor "size", e.g., "1/2 horsepower motor"
- *Input power to motor:* The amount of electricity that is drawn by the motor, which shows up on the electric meter in kW.

Drive System Losses

The components most commonly found in the drive system serving commercial building pumps are the motor (or other prime movers, such as a steam turbine) and a variable-speed drive. The drive system is the first place that efficiency losses occur as power moves from the utility system to the fluid stream.

Motor Efficiency Losses

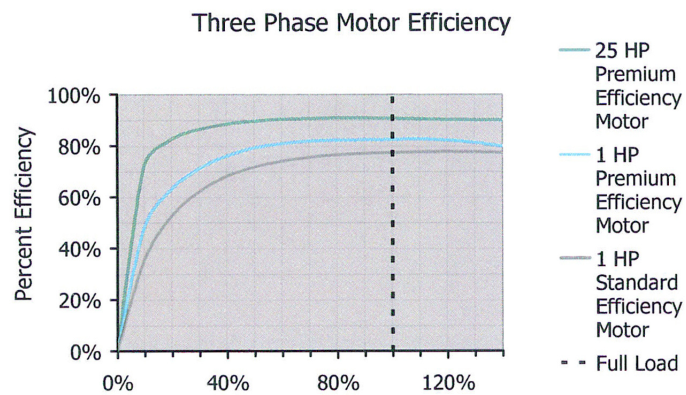
Several phenomena cause losses in motor efficiency:

- Spinning shafts must be supported by bearings and even the best bearings offer some resistance to motion.
- Current flowing through the windings in an electric motor experiences resistance, generating heat that performs no useful work. A similar effect occurs with the flux flowing in the motor's magnetic circuit.
- Efficiency losses in electric motors become heat which must be removed. The cooling fan mounted on the end of the motor shaft performs work on the air used to cool the motor, extracting power from the shaft beyond that used by the driven machine.

The efficiency of a motor depends on several factors.

- Efficiency Varies with Load: For most motors, the peak efficiency occurs at conditions other than full load; for example, 110% of full load for a common 1-hp motor and 90% of full load for a common 25-hp motor. The shape of the efficiency and other characteristic curves associated with a motor is a function of the details of its design. **Figure 2** shows how the efficiency of a motor varies with load.
- Efficiency Varies with Motor Size: Generally, larger motors are more efficienct than smaller motors if all other things are equal. This is because some of the factors affecting efficiency are not directly related to motor size. For example, bearing resistances for 1-hp and 25-hp motors are similar. But, when compared to the rated power, bearing resistance is a bigger percentage of the output for the 1-hp motor. **Figure 2** shows how the relationship between efficiency and oad is different for two different-size motors.

Figure 2: Motor Efficiency Varies with Size



- Efficiency Varies with Motor Quality: The added cost associated with a premium-efficiency motor corresponds to an added value. Most of the improvement is achieved through higher-quality materials in the electric and magnetic circuits that reduce their resistance and thus improve efficiency. Improvements in the magnetic circuit also improve the power factor, which affects efficiency as discussed below.

- Efficiency is Relatively Constant from 50% to 100% Load: While efficiency varies with load, it does not vary significantly until the load drops below 40-50% of full load. This means that running a motor at less than full load does not have a major effect on overall efficiency until the load drops significantly.

Power factor; Stored Energy with Efficiency Implications

Power factor is a number used by electrical engineers to represent the phase shift that occurs between voltage and current when an alternating current power supply (AC) is connected to a capacitive or inductive load.

The motors commonly found on pumps represent an inductive load on a typical utility system and thus have a power factor associated with them. While most of the current drawn by a motor becomes work or real power at the output shaft, some of it is used to create the motor's magnetic fields and is termed reactive power. Power factor is simply the ratio of real power to apparent power, which includes both the real power and the reactive power. The number will never exceed one, and a higher number is more optimal than a lower number.

Reactive power is never really consumed. Rather, it is simply "shuffled" around the distribution system as the inductive and capacitive fields expand and collapse with the AC cycle. Because conductors and transformers have a resistance associated with them, some of the reactive and real current flowing through them is dissipated as a heat loss. Motors with high power factors draw less reactive current. Thus, the losses due to the system resistance are minimized when high power factor motors are employed and less reactive current flows through the distribution system.

This has some implications that will be explored in the section that discusses non-overloading motor selections for pumps.

All motors are not created equal. Recent research shows a potential variation of 7-17% in efficiency for motors with identical horsepower ratings (UPPCO, 2005). It is important to recognize the difference between a manufacturer who supplies a motor that *meets* the minimum standard specified (for example, the minimum motor efficiency requirements of California's Title 24) compared with a manufacturer who *exceeds* the minimum standard efficiency requirements. The cost difference between the two selections can be insignificant; the owner stands to lose a significant opportunity for energy savings by choosing a motor with the lowest first cost.

Despite the obvious advantages, tight budgets and timelines can make it difficult for designers and facilities personnel to do much more than specify compliance with a minimum efficiency standard. Fortunately, software-based tools that allow designers to develop tighter specifications, and facilities engineers to make better purchases are becoming available. One such software tool is the U.S. DOE MotorMaster. This free program lets you quickly compare different motor options and pricing from a database of motors from 18 manufacturers.

Variable Speed Drives (VSDs)

Variable-frequency drives (VFDs) are the most common form of variable-speed drive (VSD) currently found in commercial pumping applications. However, VSD technologies like eddy-current clutches, hydraulic clutches, and variable-pitch-diameter-belt systems may be found in retrofit applications.

Regardless of the technology, properly applied VSDs can improve the overall efficiency of the *system* served by the pump. However, VSDs do introduce some cost. A typical VFD has semiconductors along with cooling fans which reduce the efficiency of a pump's drive system. Also, not all motors are rated for use with a VSD. The implications of these issues are discussed in the section on pump optimization.

Motors that are controlled by VFDs are subject to stresses that are created by the non-sinusoidal waveform of a VFD. Specifics vary with the technology used by the drive, but in general, the harmonic content associated with the modified wave form can affect the motor in several ways:

- Heat generated by the harmonic losses associated with the motor's iron laminations and copper windings raises the motor temperature, shortening its life (Houdek, n.d).
- Motor efficiency is reduced by the losses generated by the harmonics (Houdek, n.d.).
- Audible noise generated by the high-frequency harmonic content can annoy occupants and tenants situated near the drive (Houdek, n.d.).
- Voltage spikes stress the insulation in the motor, leading to premature failures in motors not designed for use with VFDs (Stone, et al. 1999).
- Eddy currents induced in the laminations and shaft of the motor cause pitting and premature bearing failures as they flow through the bearings to ground (Boyanton, n.d.).

Fortunately, VFD technology is constantly moving forward to address these problems. Where concerns exist, non-drive technologies can be applied to deal with the issues. Most of these features are best accommodated during the design phase of a project.

- Load reactors can be installed on the load side of the drive to mitigate the harmonic problems.
- Pulse-width-modulated drives allow the carrier frequency to be adjusted as a way of dealing with audible noise.
- Motors can be equipped with shaft-grounding systems to allow eddy currents to reach ground without going through the bearings.

Specifying that the motors be rated for use with variable-speed drives is recommended for any project where drives will be applied. Requiring coordination with motor and drive suppliers will also ensure compatibility and minimize problems.

Mechanical Losses

Mechanical efficiency losses similar to those encountered in the prime mover occur within the pump itself. Most of the losses result from friction in the bearings and the drag induced on the impeller as it rotates in the fluid in the pump volute.

Leakage Losses

Because the pump is composed of a rotating impeller inside the fixed volute, clearances must be maintained between the two to accommodate:

- Manufacturing tolerances for the impeller and volute;
- Expansion and contraction of materials associated with operating temperatures that differ from those at the time of assembly;
- Eccentricities in the shaft;
- Flexing and lateral motion of the shaft related to the operating loads experienced by the impeller.

Since the pressure at the discharge of the impeller is higher than the pressure at the inlet, water circulates from the discharge through the clearances to the inlet. This represents an efficiency loss since the water is moved by the impeller but is never circulated to the system served by the pump.

Minimizing and controlling these clearances is a key efficiency factor of pump design. Ensuring that they remain within the design tolerance helps ensure the pump's efficiency, so most manufacturers incorporate wear rings at one or more locations in the volute. These components provide an easily replaced surface between the impeller and volute at key locations in the leakage path, allowing tight tolerances to be established, maintained, and eventually restored when contact with the spinning impeller increases the clearance and reduces efficiency.

Wear rings are an optional rather than a standard feature with some manufacturers. This allows a competitive advantage to be gained in a bidding environment where minor differences in cost can determine who gets the job. Thus, the project specifications should require wear rings to ensure the persistence of the pump's efficiency and minimize the cost of maintaining that efficiency over time. Over the course of the pump's operating life, wear rings should be inspected and replaced if necessary any time the pump is opened for service or if the performance level has decreased.

Hydraulic and Churning Losses

The complex flow patterns that occur as the pumped fluid changes direction, accelerates, and decelerates while passing through a pump's impeller cause hydraulic and churning energy losses.

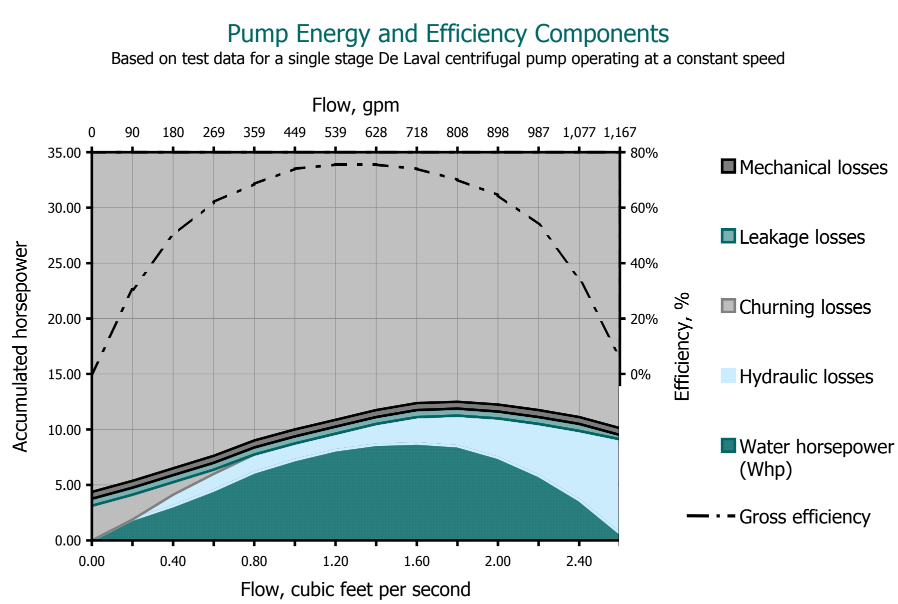
- Interaction between the fluid and the walls of the chambers in the impeller and volute causes frictional losses that vary with flow.
- The velocity change that occurs as the fluid moves from the impeller to the volute causes a shock loss that varies with flow.

Water exiting the impeller into the volute undergoes several similar changes, complicated by the fact that the motion of the water in the volute creates a spinning velocity field that accelerates the flow toward the pump discharge.

The Cumulative Effect of Pump Efficiency Losses

The test results presented in **Figure 3** paint an informative picture of the relative relationships and cumulative effect of the energy input, energy output, and efficiency

Figure 3: The Cumulative Effect of Pump Efficiency Losses



losses for a centrifugal pump (Daugherty, 1915). Note that items like leakage and mechanical losses are fairly constant regardless of flow while hydraulic losses and power vary with flow. For a given pump, there will be only one specific operating point associated with the pump's peak efficiency.

A Pump Optimization Example

Throughout this design brief, an example will be used to illustrate the optimization principles that are described.

This building, located in San Francisco, was originally intended to be a relatively efficient building. However, subsequent retrocommissioning revealed a host of lost opportunities for efficiency, some of which were addressed through retrocommissioning, and others of which can never be recaptured.

Content related to the example will be highlighted in light blue, just like this sidebar, to make it easy to distinguish from the more general content in the guideline.

Pump Performance

To optimize the performance of pumps and pumping systems, one must first understand how performance is characterized. The performance of a pump is described and communicated graphically by a *pump curve*. This pump curve is matched to a *system curve,* which describes the performance of the piping system, to determine the point at which the pump and system will operate. Both the pump and system curves can be manipulated to ensure that the operating point will be optimal.

Pump Curves

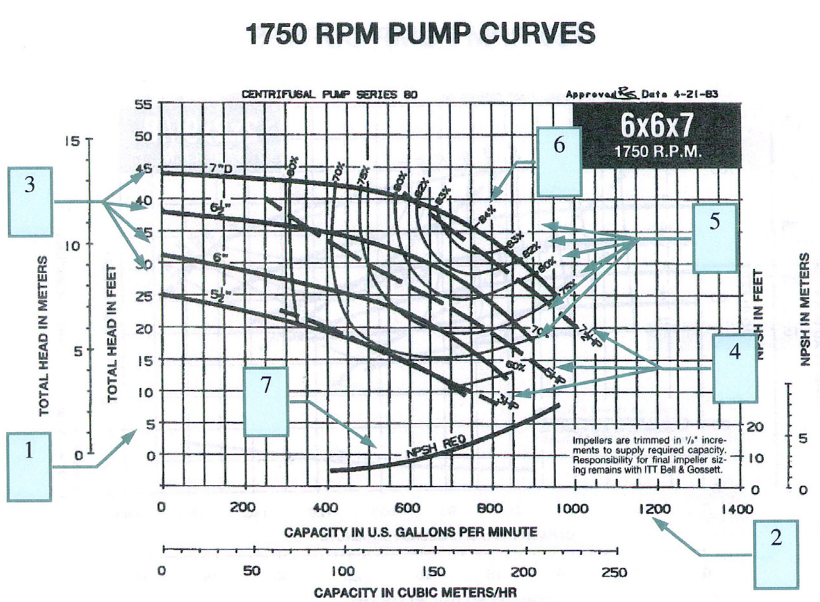
Pump performance is typically portrayed with a pump curve plotting head (the differential pressure developed across the pump) on the vertical axis against flow on the horizontal axis. A specific relationship will exist between head and flow for a fixed pump speed and a given impeller size. **Figure 4** illustrates a typical pump curve. A family of curves for different impeller sizes are shown together for a given size pump operating at a given speed. Some manufacturers also plot constant efficiency lines, constant brake horsepower lines, and net positive suction head required on the same curve. Other manufacturers use a separate curve to represent this data, usually with the parameter of interest on the vertical axis and flow on the horizontal axis.

Pump curves are developed under controlled conditions in the factory through testing. Usually, the pump is installed with a straight inlet and discharge connection and a means to measure flow and control head. The head is then varied while flow, speed, power, and other parameters of interest are documented.

The test curves form the basis of the cataloged curves used by the manufacturer to market the product. Quality control in the manufacturing process is designed to ensure that a pump will match the cataloged performance within the manufacturer's tolerances. When purchasing a pump, designers and specifiers can request its certified pump curves to ensure compliance with the published data.

With a basic understanding of the pump curve, one can consider the way that the curves would change if the pump characteristics were changed, and how the curves would change for more complex combinations of pumps.

Figure 4: A Typical Pump Performance Curve



Item	Description
1	Pumping head in feet of water column or equivalent metric units.
2	Flow rate in gallons per minute or equivalent metric units
3	Pump performance for different impeller sizes (flow vs. head)
4	Brake horsepower curves
5	Pump efficiency curves
6	Peak efficiency point
7	Net positive suction head curve

Pump Affinity Laws

Curves for different impeller sizes can be developed from the test data for a single impeller size because there is a mathematical relationship between all the properties of a pump, referred to as the "Pump Affinity Laws." The pump affinity laws can be used to predict the performance of pump at a different speed or with a different impeller diameter if the performance is known at a specific speed and with a specific impeller diameter. The laws state that:

- Pump flow will vary directly with speed or impeller diameter:

$$\left(\frac{Q_2}{Q_1}\right)=\left(\frac{N_2}{N_1}\right)=\left(\frac{D_2}{D_1}\right)$$

- Pump head varies directly with the square of the flow, speed, or impeller diameter:

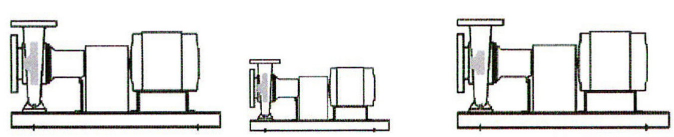
$$\left(\frac{H_2}{H_1}\right)=\left(\frac{Q_2}{Q_1}\right)^2=\left(\frac{N_2}{N_1}\right)^2=\left(\frac{D_2}{D_1}\right)^2$$

- Pump brake horsepower varies directly with the cube of the flow, speed, or impeller diameter:

$$\left(\frac{BHP_2}{BHP_1}\right)=\left(\frac{Q_2}{Q_1}\right)^3=\left(\frac{N_2}{N_1}\right)^3=\left(\frac{D_2}{D_1}\right)^3$$

However, one must recognize that the pump affinity laws are an approximation, because they assume a geometric similarity between the modified pump and the original pump, a condition that does not actually exist in the field when an impeller is trimmed. In other words, the laws assume that a reduction in impeller size is accompanied by proportionate reduction in all other pump dimensions. In the field this is not the case, as is illustrated in **Figure 5**. Note that while the impeller *diameter* is reduced, the casing size, inlet size, and impeller eye diameter are unaffected. As a result, the performance predicted by the affinity laws will not exactly match the performance achieved in the

Figure 5: Geometric Similarity and the Affinity Laws



These pumps are geometrically similar

This pump is not geometrically similar, because the impeller is trimmed.

field with an impeller trim. This implies that the affinity laws should be used with caution only to make modest changes to the pump from a published pump curve, and that the revised operating point should be verified by testing. Some manufacturers test their pumps with all available impeller sizes to address this issue.

Density, Temperature, and Viscosity Impacts

When pump curves are plotted as foot-pounds of energy per pound of fluid pumped (or simply head in feet), the impeller curves are independent of fluid temperature and density. However, density will affect the brake horsepower requirement in direct proportion to the ratio of the specific gravity of the fluid that will be pumped and the specific gravity of the fluid used to test the pump (typically water at standard conditions with a specific gravity of 1.0).

Viscosity will also affect pump performance. For most HVAC applications, the effect of viscosity will only have to be considered if the viscosity of the fluid is significantly different from the viscosity of water between 40 and 400°F. Viscosity effects should be considered for systems with fluids whose viscosities are considerably different than water, such as glycol solutions often used for freeze protection. Increased viscosity tends to shift the high flow end of the pump curve down and to the left due to the increased shear forces associated with the more viscous fluid. Higher viscosity will also increase the pumping horsepower requirement due to the increased drag of the rotating parts of the pump, although the effect is not linear. If viscosity is to be considered, correction factors can be used (using nomographs published by the Hydraulic Institute, as described, for example, in Bell and Gossett 1992, Buffalo Pumps 1959, durion Company 1976, and Karassick 1976).

In addition to these direct effects on performance, consideration of density, temperature, and viscosity should include:

- pump characteristics—motor insulation class, lubricant, and material choices—should all reflect expected operating temperatures,
- Net Positive Suction Head requirements: higher temperatures affect the vapor pressure of the fluid, resulting in the potential for cavitation at lower suction pressures,

- flow-rate requirements: differences in density and specific heat relative to water will often require a higher flow rate to convey the same amount of energy, and
- head requirements: piping friction varies between fluids, so the pumping head requirement will vary if a fluid other than water is used.

Low Flow Limitations

Pumps encounter limiting conditions when operating at both ends of the pump curve. At the low-flow end, the limiting condition is the head associated with no flow, commonly called the shut-off head of the pump. This operating point can be measured by fully closing the pump's discharge valve and documenting the differential pressure generated by the pump while moving no water. This can be accomplished safely for a centrifugal pump because it relies on centrifugal force to move water through the impeller rather than a positive displacement mechanism like a piston. However, the following cautions should be observed:

- Before testing, verify that the peak pressure that could be generated by the pump with its largest impeller is lower than the rated pressure of the pump casing and all components between the casing and the throttling valve.
- Minimize the duration of the test because even though the pump is not moving any water (beyond what is recirculated through clearances) it is doing work on the water confined in the pump casing. Eventually, this work will appear as heat, and prolonged operation with the discharge valve closed could produce temperatures high enough to boil the liquid, and damage the pump.

The test works because, for a given impeller size at a given speed, there is a very specific shut-off head. If the impeller size is unknown but the tester is reasonably confident that the wear rings and impeller are in good shape, then the differential pressure generated by the test identifies the 0 gpm flow rate produced by the impeller and thus its size. If the tester knows the impeller size, then the test can be used to assess the condition of the wear rings and impeller; if the pump fails to deliver the pressure associated with 0 gpm, then excessive clearance at the wear rings or pitting of the impeller might exist, indicating a loss of pump efficiency.

High Flow Limitations

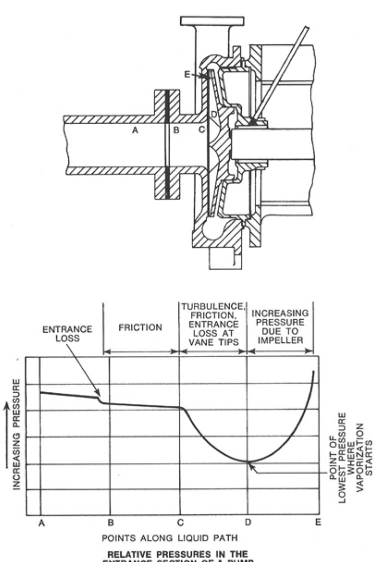
The limiting conditions at the high-flow end of the pump curve are related to noise, cavitation, and the potential for overloading the motor serving the pump. The noise and cavitation issues are related to the net positive suction head issue discussed in **Sidebar 4.** As the flow rate increases, the fluid velocity increases and its pressure decreases, creating the potential for cavitation at the eye of the impeller.

Addressing Limiting Conditions

Pump design and operation should consider the limits of the pump curve on system operation. Systems that operate for sustained periods at or near shut-off head may require recirculation lines with regulating valves to limit maximum pressures. Pumps serving variable-flow systems are examples of this situation. Another option for limiting the detrimental effect of operating at or near shut-off head is to install thermal relief valves on the pump casing. These valves open and allow a small quantity of water to flow through the pump when their setpoint temperature is reached. They are often found on booster pumps serving domestic-water systems.

Net Positive Suction Head

The figure below illustrates the cross section through a pump along with the relative pressure drop experienced by the fluid as it flows from the pump inlet line through the eye of the impeller to the pump outlet.



Notice the low pressure that is created at the eye of the impeller due to the high velocities in that area. If this pressure is below the vapor pressure of the pumped liquid, then it will boil and the pump will fail to perform. This phenomenon is termed caviation and it can be severe enough to damage or even destroy the impeller. Net Positive Suction Head (NPSH) is the positive pressure above the pumped fluid's vapor pressure that needs to exist at the inlet flange to prevent cavitation. Because the pressure drop is a function of flow, NPSH requirements increase with flow.

(Image courtesy Durion Company manual)

High flow rates can occur if the head in the system is lower than anticipated, for example when pumps are piped in parallel and only one pump is operating. In this case, the pump will "run out the curve," that is, the operating point will shift down and to the right on the impeller line. When this occurs, the required horsepower can exceed the motor's capacity, causing the motor to trip its overloads. To avoid this problem, a larger motor can be installed that will sustain the operation of the pump at any point on its curve: a "non-overloading" motor. Because this higher-capacity motor will run not fully loaded even under design conditions, there will be a minor efficiency penalty. But the part-load efficiency curve for most motors is fairly flat until about 50% load, so the penalty may not be significant. In addition, the efficiency of larger motors is typically a few points higher than that of smaller motors. Essentially, selecting a non-overloading motor prevents operational problems and only slightly reduces efficiency.

System Curves

For a fixed-piping network, there will be a specific relationship between the flow through the network and the head or pressure required to produce the flow, regardless of the pump used. Plotting the system head requirement against system flow generates a system curve that defines how the pump will interact with the system.

If the system curve is plotted on the same axes as the pump curve, the point where the system curve and the pump curve intersect will be the resulting system operating point. One of the primary goals of a pumping-system design is to select a pump so that the system curve and the pump curve intersect at or very near the peak-efficiency point of the pump.

Figure 6 shows the pump curves from **Figure 4** with a system curve superimposed. The "Design" point shows a pump that was selected to deliver 550 gpm at about 40ft.w.c. Note the following about the conditions at the design point, all of which can be read from the pump curve by interpolating between the data lines:

- The required impeller size is 7 inches
- The pump efficiency is about 78%.
- The required brake horsepower is just under 7.5.

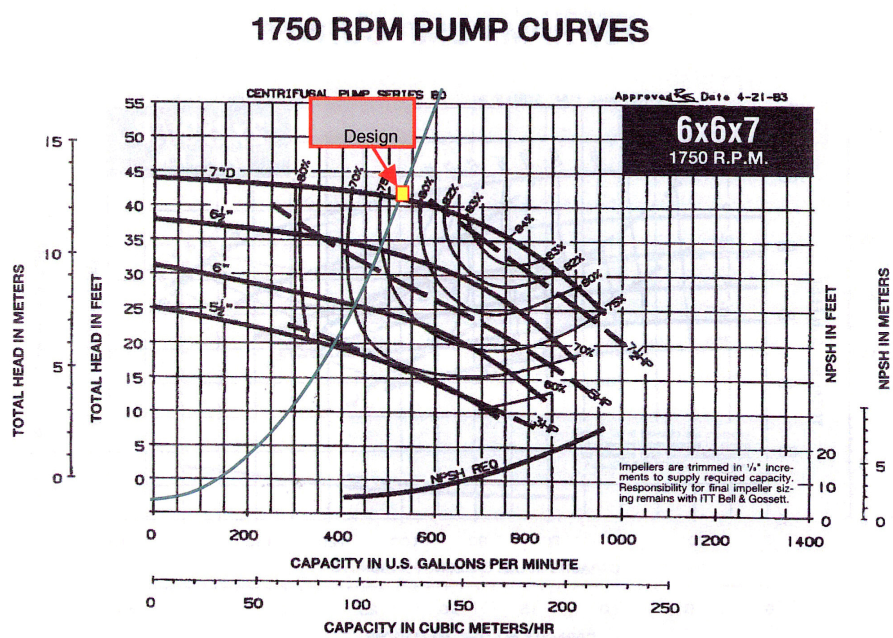
After one operating point is established, the entire system curve can obtained by plotting points using the following equation, often referred to as the "square law."

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

Although by coincidence the equation is the same as one of the pump affinity laws, it actually has its roots in the Darcy-Weisbach equation for flow in pipes. The exponent of 2 assumes fully developed turbulent flow. This condition does not exist at all times in HVAC systems and recent research suggests that an exponent of 1.85 is more appropriate. However, using an exponent of 2 is often close enough for field analysis, and also allows the curve to be quickly plotted from the design point.

The system curve only applies to a fixed-piping network. If a valve is throttled or a component in the system is changed, then a new system curve is established. Variable-flow systems actually operate over a range of system curves. For these systems, considering the movement of the operating point along the pump curve as operating conditions change is important during design.

Figure 6: Typical Pump Performance Curve and System Curve



Developing and analyzing system curves can provide insight into how a pumping system will perform (or why it is failing to perform as intended). The *Bell and Gossett Pump Engineering Manual* discusses system curves in depth and is an excellent resource for understanding pump curves, system curves, and their applications in engineering design and analysis.

The next sections describe how system curves differ for a couple of special situations frequently seen in HVAC systems.

Open Systems

The system curve discussed above is based on a closed system; that is, a system that is not open to atmosphere. Chilled and hot-water systems for commercial buildings are typically (but not always) closed systems and their pumps only need to provide the energy necessary to overcome the resistance due to flow. Changes in elevation have no impact on the pump energy requirement because the weight of the water in the supply riser to the top of the building is balanced by the weight of the water in the return riser coming back down.

Open systems have a break in the piping circuit that opens them to atmosphere. Pumps serving these systems must also provide the energy to lift the water through any unbalanced changes in elevation in addition to the energy to overcome resistance due to flow. The static lift in a cooling-tower circuit is a good example. The system curve for this application will not pass through 0 ft.w.c. at 0 flow.; rather, the curve is created by adding the static lift to the friction-related head at all operating points. A fixed resistance like a pressure regulator or flow-control valve will also shift a system curve from the 0-0 point by an amount equal to the constant resistance it generates in the system.

Example of Pumps in Parallel

The pump and system curve shown here are for the example system, designed to deliver 1,100 gpm at 40.5 ft.w.c. at design. Note the following:

- At design, each pump will be delivering 550 gpm at 40.5 ft.w.c. at 78% efficiency using approximately 6.5 bhp each (Points A and B).
- The design flow rate is delivered for approximately 13 bph (2 times 6.5).
- If one pump fails, the system performance shifts to point C, providing 82% of design flow using only one pump which will draw something in excess of 7.5 bhp.

If the required performance with one pump running is only 50% of the design requirement, then this condition could be met using less than 3 bhp (point D). A situation like this is common in HVAC applications where parallel chillers or boilers are piped to a common header to serve a common distribution system. In another example, the pump was sized to serve one chiller delivering a nominal 2,800 gpm at 120 ft.w.c. Field testing revealed that it could in fact deliver nearly 4,300 gpm at 80 ft.w.c. when operated against the wide open system. This pump may be able serve both chillers on its own, saving significant pump energy over the original design intent that involved running two pumps.

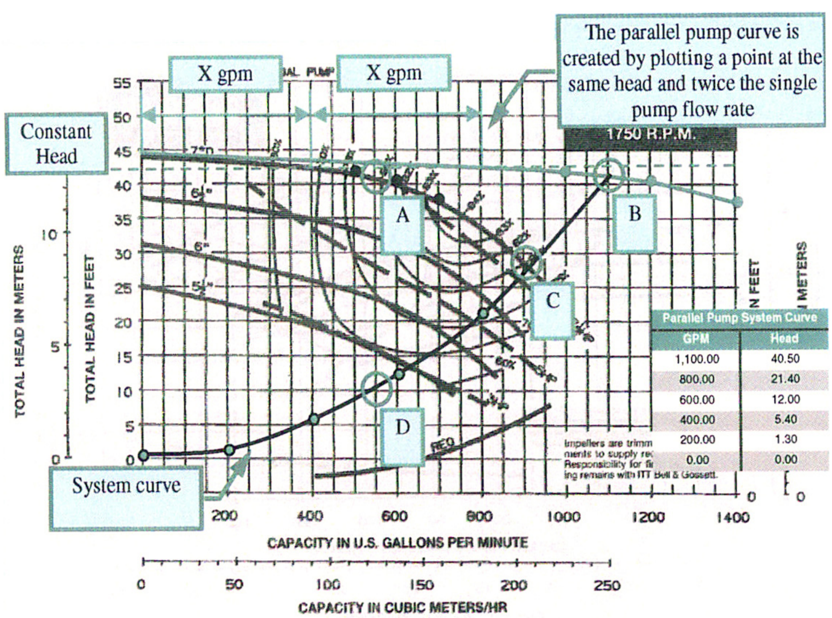
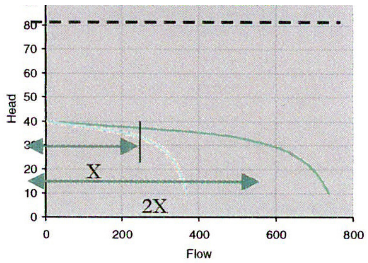


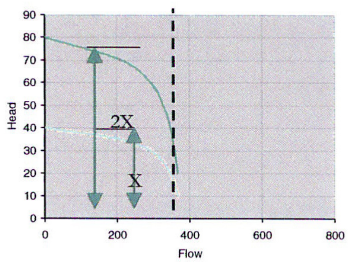
Figure 7: Creating a Curve for Series and Parallel Pumps

Constant Head



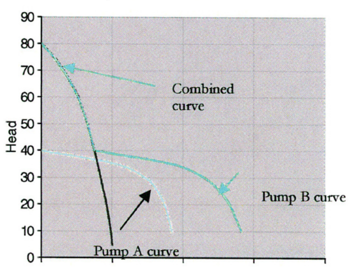
The faded teal line is the curve for one pump. The dark teal line is the curve for two of those pumps piped in parallel.

Constant Flow



The faded teal line is the curve for one pump. The dark teal line is the curve for two of those pumps piped in series.

A Curve for Dissimilar Pumps in Parallel



Series and Parallel Pumping Systems

When two pumps are piped in parallel, the suction and discharge connections for the two pumps reference the same pressure. Thus, the points on the pump curve for two parallel pumps are created by adding the flow of the individual pumps at a constant head. **Figure** 7 illustrates pump curves for two identical pumps in parallel.

Arranging pumps in series is less common in HVAC applications. When two pumps are piped in series, all of the water that flows through one must flow through the other. Thus, the points on the pump curve for two series pumps are created by adding the head of the individual pumps at constant flow (see **Figure** 7).

The interaction of parallel pumps with the system creates challenges but also provides unique opportunities to the design and operating team. Due diligence must be exercised when applying pumps with dissimilar characteristic curves in parallel. Under some operating conditions, the pump with the higher head rating can generate sufficient pressure to shut off flow from the other pump. When the system geometry hides the parallel arrangement, mysterious, intermittent operating problems can occur. Similar considerations apply to pumps in series.

Selecting Pumps

Designers identify the needs of their system, and try to select a matching pump from a manufacturer's product line. The key to specifying an efficient pump is to do so in terms of fundamentals including the required flow and head, the minimum acceptable pump and motor efficiency and motor power factor, and the maximum acceptable brake horsepower and speed, as well as configuring other factors such as inlet diffusers and balancing valves. Specifications prepared this way ensure that the efficiency envisioned by the design is realized at bid time while leaving manufacturers free to exploit their strong points.

Establishing the Flow Requirement

Pumps should be selected based on the amount of head necessary to produce the flow required by the load when it is served by the piping network. The piping network is defined by the physical arrangement of the pump and the system.

The flow rate can be mandated by an equipment manufacturer to ensure proper performance of their machinery, or it can be determined by the designer. In HVAC, the flow is typically provided to handle some sort of heat-transfer process and will generally be a function of:

- the energy released or absorbed by the heat-transfer process,
- the physical properties of the fluid that is circulated by the pump, and
- the heat-transfer characteristics of the equipment served by the pump used to handle the load.

If water is being circulated through one side of a heat exchanger in an HVAC process, then the following equation can be used to solve for gpm—the required flow rate:

$$Q = 500 * gpm * \Delta T$$

where:

Q = *the thermal energy transferred to or from the water stream in Btu/hr,*
500 = *a units conversion constant to account for the properties of water at the conditions normally encountered in HVAC systems,*
gpm = *the water flow rate in gallons per minute, and*
ΔT = *the temperature change created by the energy transfer to or from the water stream within the heat exchanger.*

Once the load is known, designers will manipulate the characteristics of the heat-transfer equipment to establish an optimum performance for both the pumping system and the system it serves. For example, the circuiting and/or fin spacing associated with a chilled-water coil might be manipulated while air- and water-side flow rates and entering temperature are held constant. This is done to achieve a water-side temperature rise that matches the design criteria associated with the chiller plant while providing a leaving-air condition that meets the cooling and dehumidification loads on the air handling unit.

Similar considerations apply to the tube bundles associated with the condensers and evaporators in chillers, the heat exchangers used to transfer heat from steam to water in heating hot water systems, and the heat exchangers used to transfer heat from boilers to the air or other fluid streams they serve.

Establishing the Head Requirement

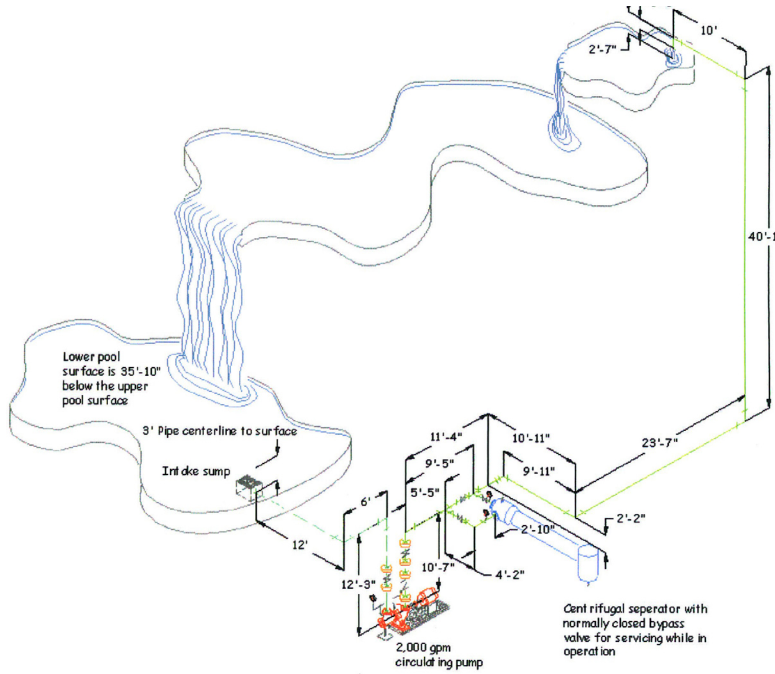
The head required from a pump is determined by assessing the amount of energy needed to move the flow circulating through the piping network from the pump to the load and back again. For most systems, establishing the load and the physical arrangement of the system occurs before establishing the head.

There are several techniques for assessing the required pumping head once the required flow rate has been determined, including:

- approximation (for early design consideration only),

Comparison of Pump Head Assessment Techniques

A resort hotel water feature serves as a good example of a pumping system with elevation changes in addition to other head requirements. The table below illustrates the similarities and differences between the different pump head estimating techniques described in the text when applied to the water feature illustrated here. Note how irrespective of technique, the pressure loss through the centrifugal strainer and the head required to lift the water to the upper pools predominate. Generally, this is true for most HVAC systems; i.e. the equipment and, for open systems, elevation changes, will dominate the pressure drop requirements unless the piping network is extensive and operating at a high flow rate relative to the line size.



	Field Estimate		Manual Calculation		PSIM Model	
Elevation change (30-40 ft)	30-40	Estimate based on observation	36	Based on drawing elevations between pools	37	Based on drawing elevations between pools plus distance above upper pool to top outlet
Pipe to pump suction (20-30 ft)	1-1	Distances paced off in the field and assessed at the 4 ft.w.c. per 100 lineal feet of pipe design rule	1	Scaled from the drawings	1	Scaled from the drawings
Fittings to pump suction (10-30 equivalent ft)	0-1	Added 50-100% more equivalent feet to the linear feet of pipe based on industry rule of thumb	1	Based on the piping isometric and manufacturer's data for valve losses	With above	Based on the piping isometric and program parameters for valve and fitting losses
Pipe to centrifugal seperator (10-15 ft)	0-1	Distances paced off in the field and assessed at the 4 ft.w.c. per 100 lineal feet of pipe design rule	1	Scaled from the drawings	5	Scaled from the drawings
Fittings to centifugal seperator (5-15 equivalent ft)	0-1	Added 50-100% more equivalent feet to the linear feet of pipe based on industry rule of thumb	3	Based on the piping isometric, tables in the ASHRAE pocket handbook and manufacturer's data for valve losses when available	With above	Based on the piping isometric and program parameters for valve and fitting losses
Centrifugal seperator	15-20	Estimate based on past experience	12	Based on manufacturer's data	12	Based on manufacturer's data
Pipe to upper pool (100-120 ft)	4-5	Estimate based on observation	1	Scaled from the drawings	6	Scaled from the drawings
Fittings to upper pool (10-15 equivalent ft)	0-1	Added 50-100% more equivalent feet to the linear feet of pipe based on industry rule of thumb	4	Based on the piping isometric and manufacturer's data for valve losses	With above	Based on the piping isometric and program parameters for valve and fitting losses
TOTAL	51-69	Safety factor implied by low-high range approach	63	Includes 10% safety factor	61	Includes no safety factor

- take-off and manual calculation,
- computer modeling, and
- re-evaluation of head assessments in the field.

Approximation

Approximating pump head is useful in the early phases of design to establish approximate pump sizes and electrical loads, as well as in the field to assess the suitability of an existing pump given the existing conditions. It is often possible to estimate the required head in a matter of minutes based only on the physical distances between the components in the system. Refer to the *HPAC* article "Right-sizing Pumping Systems" for an example of this technique (Sellers, 2005). When using this approach, it is common to anticipate a range in which the pumping head requirement might lie. Of course, more detailed methods of establishing head requirements should always be used in later design phases.

Take-off and Manual Calculation

Before computers, a common approach to establishing the pumping head required by an HVAC pumping system was to tabulate:

- the lineal feet of pipe,
- the number and type of fittings, and
- the number, type and characteristics of the heat transfer and other equipment elements.

Each element in the tabulation was assigned a pressure drop based on manufactures data, published charts, and tables of flow coefficients. Adding up the pressure drops associated with the various elements established the pump requirements. Chapter 36 in the *2005 ASHRAE Handbook of Fundamentals* contains the tables and charts needed for such a calculation.

Computer Modeling

There are several computer programs that let you model and assess simple piping networks. One such program is the Pumping System Improvement Modeling Tool, available at no cost from the *Pump Systems Matter Initiative* website, (www.PumpSystemsMatter.org). There are also commercially available programs like Pipe-Pro™ which let you assess more complex piping networks in great detail.

Re-evaluation of Head Assessments in the Field

During construction, the facility staff and commissioning provider can use a copy of the pump head calculation to assess and verify design intent. Construction observation can verify that the installed piping configuration matches the design, so that any discrepancy can be addressed before the installation has progressed too far.

If construction-phase testing reveals that performance does not match expectations, the pressures in the piping circuit can be measured and compared to those used during design. Significant deviations often indicate the source of the problem, which can include a fabricated fitting instead of a factory fitting, debris caught inside of the pipe, or a field arrangement that differs from the design intent.

Selecting the Pump

Knowing the head and flow rate required at the design operating point, you can select a pump. Review the pump curves for pumps with the desired characteristics, and a select a pump with a design operating point that achieves maximum efficiency. This point will not necessarily lie neatly on one of the shown impeller curves, so you may need to select the next largest impeller size (and throttle the pump discharge slightly to achieve the required flow rate).

Selecting the Motor

You can use the pump power equation to select the required motor size. The brake horsepower at the maximum expected flow is the power that the motor must be able to supply. In most instances, the result of this calculation will not exactly match one of the commercially available incremental motor sizes (1 hp, 1.5 hp, 2 hp, 3 hp, 5 hp, etc.) and the pump will be equipped with the next larger standard motor size. In some situations, the motor may be two or even three sizes larger than the calculations suggest, which results in non-overloading performance (described earlier, under "Addressing Limiting Conditions").

Example of Motor Option Comparison

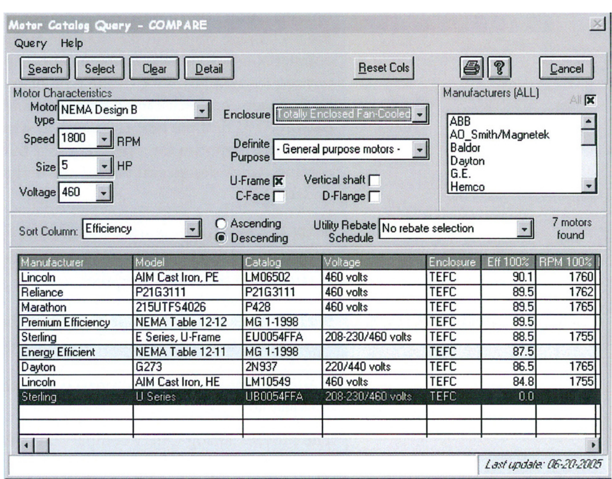
The following example uses MotorMaster to compare motor options under several different scenarios for the pumps serving the ice storage system in the example facility. While the comparisons are specific to the motors serving the pumps, the techniques are illustrative of the type of analysis that can be used by designers, commissioning providers and facilities engineering personnel to find the best right answer when making equipment selections. Suppose for a moment that the example system is just coming out of design and you, as the designer, wanted to understand:

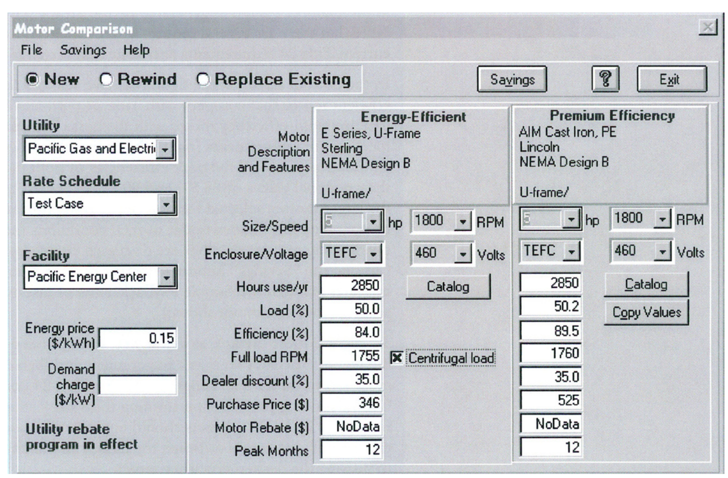
- What is the range of efficiency available for the nominal 1,800 rpm, totally enclosed fan cooled 5 hp motors that the circulating pumps will require?
- How much room for improvement is there over the minimum threshold of 87.5% required by Title 24?
- What is the cost of the improved efficiency over Title 24, if available?

The Motor Catalog Query comparison in MotorMaster allows the user to view a list of all motors meeting their criteria that are in the current database. The top screen in the accompanying figure shows the list of motors that meet the requirements of the pump in the example system as generated by the MotorMaster database. Note that there are several motors with efficiencies lower than the current Title 24 requirements and none that match it exactly

In the bottom screen in the figure we have selected an energy-efficient motor that complies with Title 24 (left) and compared it to a premium-efficiency motor with the best efficiency available meeting our requirements (right). Note that the program has automatically derated the peak efficiencies of both motors (from their nominal values of 88.5% and 90.1% respectively), because the larger motors selected for non-overloading operation will not be operating fully loaded under normal conditions (non-overloading motor selections are discussed elsewhere in the brief). Clicking the "Savings" button generates an economic analysis that indicates an attractive Simple Payback Time of just over three years (output screen not shown).

To capture these savings, as a designer, you will need to specify the higher efficiency level as a requirement and then enforce that requirement rather than simply specifying Title 24 compliance. Commissioning providers performing third party shop drawing review for LEED™ projects should target verifications of this type. Small differences between the design and reality can add up to missing the desired efficiency target.





Pump Options

Well thought-out pump sizing is only part of pump selection. Equally crucial are the configuration and features of the connections between the pump and the piping network, equipment, and system it serves. For the trim and fittings, “the devil is in the details” where cost/benefit and functionality are concerned. Knowledge of the operating criteria associated with the system is also key. Most manufacturers have lists of optional features for their stock offerings that allow the application engineer to influence pump design—features like seals, coupling, and wear rings. By selecting the features and options that are appropriate to their project, application engineers can influence some parameters of the pump design in a way that will ensure its long-term efficiency and operability, without having direct control over the details of the pump design.

Valves

Virtually every pump installation will require some sort of isolation and throttling device on the pump discharge. For parallel pumps, preventing backflow is also necessary. One approach to providing these functions is to install a service valve, a balancing valve, and a check valve. Such an approach uses off-the-shelf items but requires that each device be installed separately. In the larger pipe sizes, this means that each device will require a set of flanges and the associated welds, bolts, and gaskets that can be costly and labor intensive. To improve economy, several manufacturers have developed combined valves that provide all three functions. Some manufacturers use one set for all functions while others provide a separate valve seat for the check-valve function. The “one seat does it all” approach (illustrated in **Figure 8**) has advantages and disadvantages. On one hand, it minimizes the size and number of components required to fabricate the valve and thus, its cost. On the other hand, if the seat fails, then all functionality is lost, including the isolation function. As a result, repairing a seat failure that causes backflow through the standby pump in a parallel pump application will require a system outage, in contrast to a system that uses a valve with a separate seat and disc to provided the isolation function. The use of independent components also addresses this problem.

Figure 8: Options for Valves

Combined Function Valve

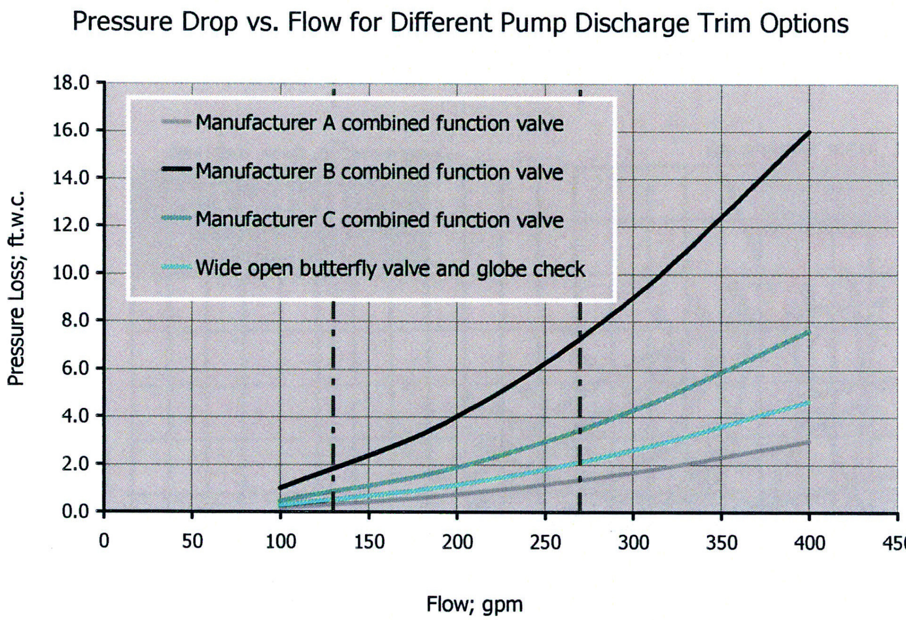
This is a picture of a combined function valve prior to installation.

Discharge Connection

This is a close-up of the discharge showing the check valve spring and valve actuating handle.

Inlet Connection

This is a close-up of the inlet showing the valve disc situated in the valve seat.



Item	Low End of the Application Range		High End of the Application Range	
	4 inch	10 inch	4 inch	10 inch
Flow, gpm	130	1,650	270	3,000
Difference in wide open valve pressure drop, ft.w.c., lowest vs. highest pressure drop	1.50	1.50	6.00	6.00
Brake horse power associated with the difference in pressure drop	0.07	0.78	0.60	5.68
kW associated with the difference in pressure drop	0.05	0.58	0.45	4.24
Annual kWh and $ associated with the difference in pressure drop, 2,600 operating hours per year	140 $21	1,515 $227	1,167 $175	11,020 $1,653
Annual kWh and $ associated with the difference in pressure drop for 24/7 operation	472 $71	5,091 $764	3,921 $588	37,029 $5,554

Figure 8 compares pressure loss vs. flow for 4-inch combined-function valves from different manufacturers and compares a wide-open butterfly valve (which can serve as an isolation valve and a throttling valve) with a globe-type check valve. The dashed lines show the normal operating range in terms of flow for a 4-inch line. Note that there are significant differences between the manufacturers. Also note that a similar analysis for another line size would result in a different ranking for the manufacturers in terms of lowest to highest pressure drop.

The table in **Figure 8** illustrates the difference in operating costs. Data for 10-inch valves with similar pressure drop characteristics at the low and high end of their application range area also shown. Note that the difference in pressure drop can result in a significant difference in operating cost for two equivalent valves if the only requirements are function (isolation/throttling/check valve) and size. The pressure difference could also mean the pump selection misses the peak-efficiency point on its curve and, in an extreme case, may even mean the pump can not deliver design flow.

Suction Diffusers

Pumps require a uniform velocity profile entering the impeller to ensure factory-rated

Example of The Impact of Pump Components

Field inspection of the pump at the example facility revealed that the pump was furnished with both a suction diffuser and a strainer. Since the suction diffuser accomplishes the strainer function, the strainer represents a first and operating cost that could have been avoided. Specifically, a more detailed design review or construction observation process could have saved the owner approximately $500 in first cost and over $1,000 in operating costs by eliminating the strainers from both pumps in the system. These savings alone would have paid for the time necessary for a commissioning provider to perform design review.

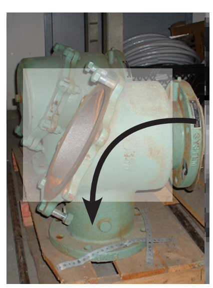
When the strainer and suction diffuser were opened up to inspect their screens and eliminate the screen from the strainer (and thus the energy burden associated with its pressure drop) the commissioning team discovered that the fine mesh start-up screens were still in place. While desirable during initial flushing, fine meshed screens in a closed system that has been thoroughly flushed and cleaned typically represent an unnecessary pressure drop. Eliminating the fine screen while retaining the coarse screen in the strainer or suction diffuser subsequent to initial system flushing will ensure that things like valve plugs and other large items that might come loose during the operating life of the system are intercepted before they can damage the pump. Even in an open system, some facilities engineering teams find it desirable to run with only the coarse screens once the initial clean-up process has been completed since the fine screens can become plugged with sediment and other fine debris that would otherwise circulate freely and could be dealt with by some other means.

performance. Generally, this can be achieved by installing the pump with 5-10 equivalent diameters of straight pipe leading up to the inlet connection. Unfortunately, this requirement can consume a considerable amount of space in the equipment room. Suction diffusers allow the space to be reclaimed while ensuring design performance. Most manufacturers will guarantee their cataloged performance if their pump is used with the appropriate suction diffuser from their product line. This assurance is generally achieved by three suction-diffuser design characteristics.

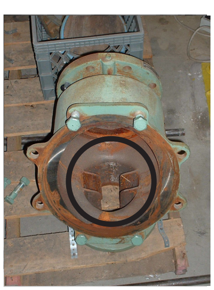
- The portion of the device where the water makes the turn is generally enlarged relative to the line size and incorporates a strainer. The enlargement reduces the water velocity and thus the pressure drop while the strainer element acts as a diffuser.
- The outlet of the chamber with the strainer includes straightening vanes, de signed to ensure that a uniform velocity profile enters the eye of the pump's impeller.
- The strainer included with the device eliminates the need for a separate strainer, providing a modest cost savings by eliminating a part and a pair of flan ges on larger pipe sizes. **Figure 9** illustrates a typical suction diffuser installed in a piping system as well as its internal construction.

Figure 9: Suction Diffusers

Installed Suction Diffuser
This is a picture of a suction diffuser installed in the field.



Suction Diffuser Strainer
The screen occupies the area indicated by the shading. Flow is as indicated by the arrow.

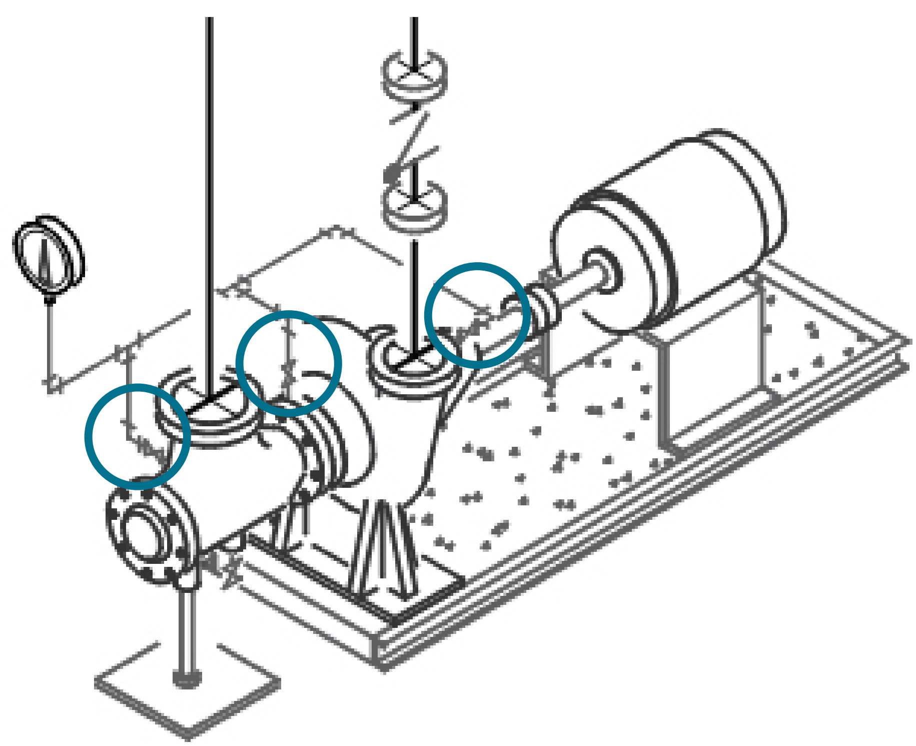


Suction Diffuser Guide Vanes
Straightening vanes (circled) ensure a uniform velocity profile into the impeller eye.

Flange Taps

Pump performance should be measured in a manner that matches the technique used at the factory to develop the pump curves. Flange taps ensure that the measurement point duplicates the point used for the factory test, so installing flange taps is crucial in meeting this goal. Locating the taps in the piping ahead of and after the pump will not provide identical results, especially if there are fittings between the tap and the pump flange. Since pump performance is based on differential pressure, one gauge is actually better for assessing the operating point because the gauge error will be canceled out when the readings are subtracted. This provides an opportunity for value engineering that also adds rather than detracts from its operational value. **Figure 10** illustrates how one gauge can be piped to pick up the necessary pressures, including the pressure ahead of the strainer. Note how three ¼-inch ball valves (circled) and a piping network allow one gauge to read multiple pressures.

Figure 10: Flange Taps to Measure Pressure



Design Checklist

- ☑ Consider load profiles in addition to peak conditions
- ☑ Consider the interaction of the pump with the system
- ☑ Consider long-term operating requirements
- ☑ Include commissioning specifications
- ☑ Document design intent
- ☑ Specify efficiency in terms of fundamental parameters
- ☑ Consider premium efficiency motors
- ☑ Require motor performance data submission
- ☑ Specify wear rings
- ☑ Select materials suitable for the fluid to be pumped
- ☑ Select components suitable for the operating locale
- ☑ Include flange taps for performance measurement
- ☑ Provide ball valves on vent, drain and gauge ports
- ☑ Include requirements for efficiency related options
- ☑ Coordinate seals and packings with the fluid pumped
- ☑ Coordinate seals and packings with the application
- ☑ Coordinate seals and packings with the owner
- ☑ Optimize prime mover efficiency
- ☑ Coordinate couplings/guards with the owner
- ☑ Coordinate couplings/guards with code requirements
- ☑ Engineer the mounting arrangement
- ☑ Coordinate VFDs with the motors they serve
- ☑ Use pump/system curves to assess requirements
- ☑ Assess viscosity and density performance impacts
- ☑ Be aware of curve limits and NPSH issues
- ☑ Assess parallel and series pump interactions
- ☑ Select non-overloading motors
- ☑ Provide VFDs for control not balancing purposes
- ☑ Calculate project specific head requirements
- ☑ Use reasonable, but not excessive safety factors
- ☑ Provide a copy of head calculations to the field

Facilities Engineering Checklist

- ☑ Provide design time and budget for rightsizing
- ☑ Budget for commissioning
- ☑ Participate in the design and commissioning process
- ☑ Maintain files for motor and pump data submissions
- ☑ Inspect wear rings when pumps are disassembled
- ☑ Make owner preferences known early in the design
- ☑ Verify owner preferences are reflected in the design
- ☑ Verify owner preferences are reflected in submittals
- ☑ Verify owner preferences are reflected in the field
- ☑ Participate in performance testing
- ☑ Participate in and document training
- ☑ Inspect/replace wear rings when pump is open
- ☑ Periodically verify alignment
- ☑ Minimize cavitation potential by running with coarse screens subsequent to initial flushing
- ☑ Minimize cavitation potential by cleaning strainers
- ☑ Monitor pipe fabrication for deviations from design intent
- ☑ Cross check performance issues against design calcs
- ☑ Use pump/system curves to assess performance
- ☑ Be aware of viscosity and density performance impacts
- ☑ Be aware of curve limits and NPSH issues
- ☑ Be aware of parallel pump interactions
- ☑ Be aware of series pump interactions

Commissioning Checklist

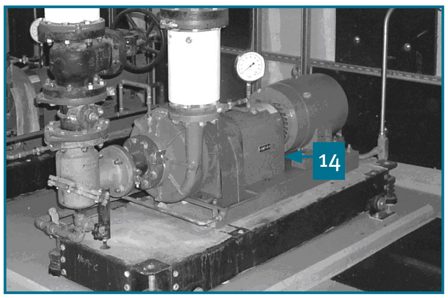
- ☑ Support commissioning spec development
- ☑ Provide O&M focused design review
- ☑ Provide construction observation services
- ☑ Cross-check motor test data with design intent
- ☑ Cross-check construction features with design intent
- ☑ Verify performance via field test
- ☑ Verify shop drawing compliance with design intent
- ☑ Verify installation compliance with design intent
- ☑ Verify installation compliance with supplier specs
- ☑ Verify mount and anchor seismic compliance
- ☑ Verify alignment and pump base grouting
- ☑ Test performance for compliance with design intent
- ☑ Coordinate manufacturer training
- ☑ Support good design with reasonable budgets
- ☑ Periodically verify pump performance
- ☑ Assess renovation impacts in light of original intent
- ☑ Use pump/system curves to assess performance
- ☑ Be aware of viscosity and density performance impacts
- ☑ Be aware of curve limits and NPSH issues
- ☑ Be aware of parallel pump interactions
- ☑ Be aware of series pump interactions

Other Features

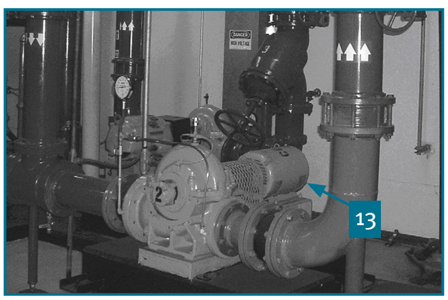
Figure 11 details other pump features and configurations. The table in the figure shows that application considerations can have a significant effect on efficiency, ease of maintenance, and pump life. This implies that system designers and operating personnel can have a significant effect in these areas even though they have no direct control over the

Figure 11: Pump Arrangements and Construction Features

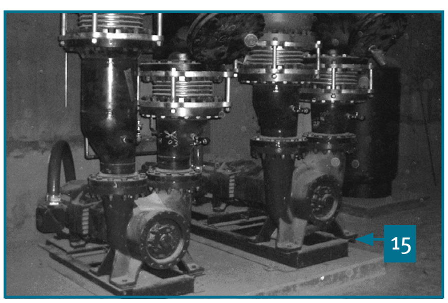
End Suction



Horizontal Split Case

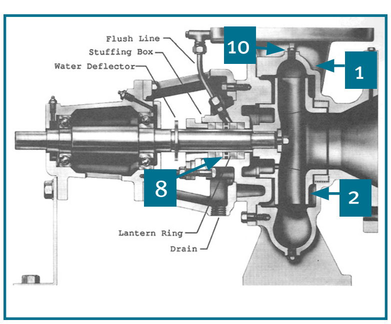


Vertical Split Case

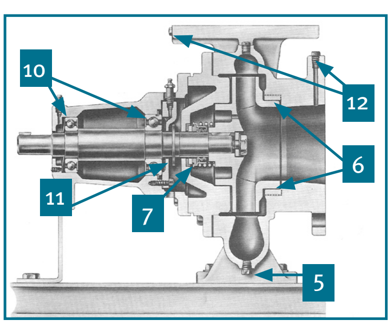


Pump Construction Features				
Component	Application Impacts			Notes
	h	O&M	Life	
1 - Volute/Casing		✓	✓	2, 3
2 - End Suction Impeller	✓	✓	✓	3, 4, 11
3 - Double Suction Impeller	✓	✓	✓	3, 4
4 - Vent	✓	✓		5
5 - Drain		✓		6
6 - Wear Ring	✓	✓		4
7 - Mechanical Seal		✓	✓	3, 7
8 - Packing		✓	✓	3, 7
9 - Flushing Line		✓	✓	3, 8
10 - Bearings	✓	✓	✓	3, 9
11 - Shaft and Slinger		✓	✓	3, 10, 11
12 - Gauge Taps	✓	✓		12
13 - Prime Mover	✓	✓		4
14 - Couplings and Gaurds		✓		13
15 - Mount/Base		✓	✓	14

End Suction with Packing



End Suction with Mechanical Seal



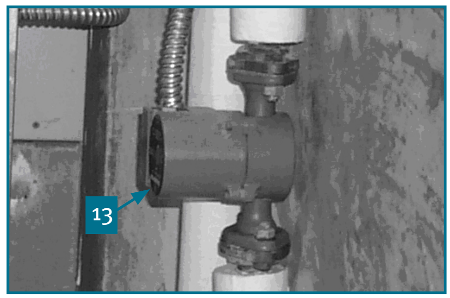
pump design. More detailed information about the implications of pump construction on design and operation can be found in several of the references cited at the end of this brief (Bell and Gossett, Buffalo Forge (1959), Durion (1976), Karassik (1976, 1981), Sellers (2003)). Other considerations to take into account include:

- Make sure the materials of construction are compatible with the application.

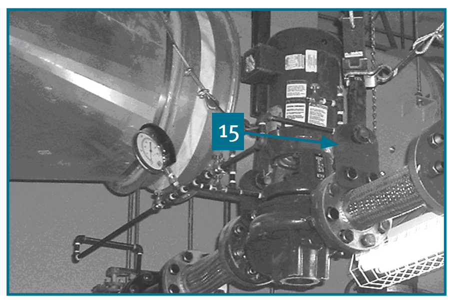
Figure 11 (cont): Pump Arrangements and Construction Features

1. Examples of each feature are only pointed out once in one of the cross-sections although many occur in all of them.
2. Case disassembly provisions like back access or vertical splits can facilitate maintenance by minimizing the disruption to the pump, bearings, and drive system for seal and bearing replacement.
3. Selecting materials compatible with the application promotes long life. Bronze fitted pumps are less subject to corrosion in systems with high makeup rates.
4. Selection and tuning per the recommendations of this guideline and applicable references will ensure efficient operation and minimize NPSH problems.
5. Provide ball valves on casing vents to ensure a fully flooded volute and maximum efficiency.
6. Provide ball valves on casing drains to facilitate drain-down for disassembly.
7. Mechanical seals require less maintenance and minimize makeup requirement but tend to fail catastrophically. Packings require ongoing adjustment and bleed water, but they fail gradually, making them a good choice for fire pumps.
8. Flushing lines are sometimes provided for mechanical seals exposed to water with high levels of water treatment or contamination. Flushing water can be from the circulated flow, with or without filtration, or from an independent source. Similar considerations apply to packings.
9. Bearing selections may be driven by the owner's preferences for maintenance purposes, the requirements of a harsh environment, or the nature of the fluid circulated by the pump. For instance, a pump circulating chilled water in a plant attended around-the-clock would be best served by grease lubricated bearings since ongoing attention from the maintenance staff would ensure proper lubrication and condensation might interfere with an oil lubricated arrangement.
10. Shafts must withstand both radial and axial loads in addition to providing a smooth sealing surface for packings. To minimize overhaul costs, including optional sleeves on pumps that use packings for seals allows the sleeve and not the shaft to be replaced if the packings run dry and scour the surface they ride on.
11. Some manufacturers provide holes in the impeller disk to allow water from the eye of the impeller to access the rear of the impeller where the pressure acting on the area of the disk can offset some of the axial loads associated with water entering the eye of the impeller.
12. Gauge taps on the flanges allow pump performance to be verified using the same reference point as the manufacturer used for testing.
13. Coupling/guard selections are often driven by owner and OSHA requirements.
14. The integrity of a pump is only as good as its base as discussed in Sidebar 10. Spring isolation mounts must be properly snubbed for applications in a seismic zone and require concrete fill to dampen vibration.

In-line

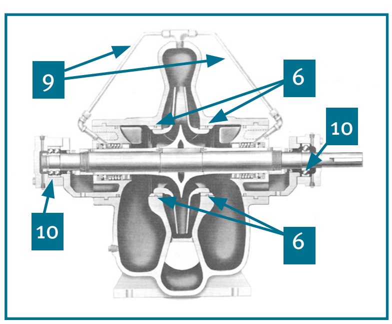


Vertical In-line

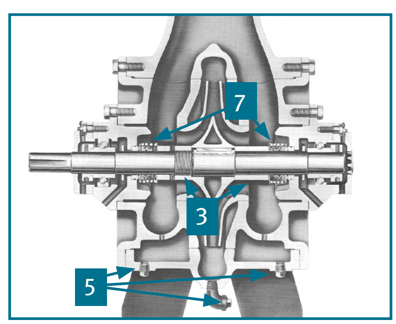


Dual-Arm

Horizontal Split Case with Mechanical Seal



Vertical Split Case with Mechanical Seal



Frequently, this involves assessing life-cycle cost rather than just first cost. For example, the added costs associated with bronze-fitted construction will be money well spent in the long term for systems with elevated oxygen levels and the potential for corrosion. However, an uninformed owner may want to select materials based on lowest first cost. Similarly, unless bronze fittings are clearly specified, competitive bidding will most likely provide lower-cost but less-durable materials.

- Ensure compatibility with the owner's needs and capabilities. For example, grease-lubricated bearings and packings may not be as attractive as permanently lubricated bearings and mechanical seals to an owner with no in-house maintenance capability.

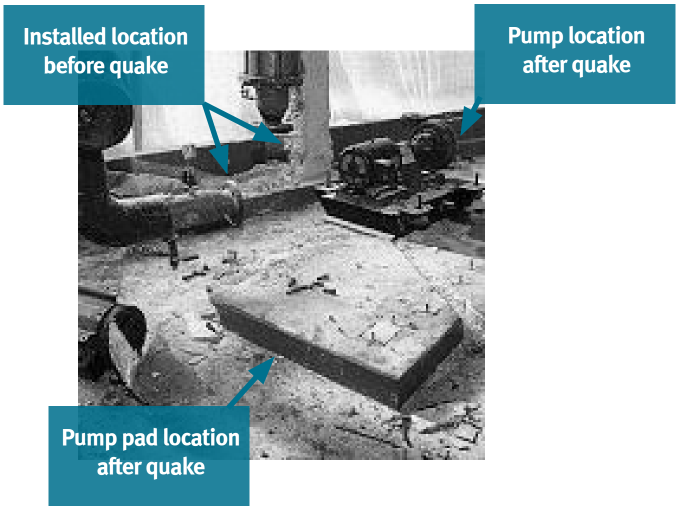
- The operating environment, operating cycle, and system interactions that will be imposed on a pump by the nature of its application are key design considerations. For example, oil-lubricated bearings may be a poor choice for a pump handling chilled water in an environment where condensation will occur. **Figure 11** illustrates different bearing arrangements and other pump-construction features with operational considerations.

- Several manufacturers have designed their pumps so that the internal components can be accessed through the rear of the pump casing, thereby eliminating the need to disassemble the piping when service is required. Close-coupled pumps with a back-access feature should be mounted with capscrews instead of studs to allow the motor to slide out of the way for service.

- A pump's foundation is essential for reliable performance and longevity: Factory base frames should be leveled and anchored to the pad and the pad should be anchored to structure. Failure to do so can compromise the integrity of the installation during a seismic event, as illustrated in **Figure 12**.

- Factory alignment does not guarantee field alignment. The pump should be re-aligned after it is mated to its foundation. Subsequently, the base frame should be filled with grout to improve rigidity. Alignment should be verified after final connections are made for all systems and also after initial operation for systems that operate at high temperatures. Doweling (drilling a hole into the base of the unit and into the supporting frame, and inserting a steel pin) after verifying alignment can help ensure that the pump will remained aligned.

Figure 12: Foundations; the Basis for Reliability and Longevity



Optimizing Performance

Optimizing performance consists of three separate activities: selecting the right pump, achieving the right maximum flow, and adjusting flow to changing conditions over time. It can also be improved later in the life-cycle of the building, although optimization opportunities are more limited then.

Example of Analysis of Various Pump Optimization Alternatives

	Field Estimate		Manual Calculation		PSIM Model	
Elevation change	30-40	Estimate based on observation	36	Based on drawing elevations between pools	37	Based on drawing elevations between pools plus distance above upper pool to top outlet
Pipe to pump suction	1-1	Distances paced off in the field and assessed at the 4 ft.w.c. per 100 lineal feet of pipe design rule	1	Scaled from the drawings	1	Scaled from the drawings
Fittings to pump suction	0-1	Added 50-100% more equivalent feet to the linear feet of pipe based on industry rule of thumb	1	Based on the piping isometric and manufacturer's data for valve losses	with above	Based on the piping isometric and program parameters for valve and fitting losses
Pipe to centrifugal seperator	0-1	Distances paced off in the field and assessed at the 4 ft.w.c. per 100 lineal feet of pipe design rule	1	Scaled from the drawings	5	Scaled from the drawings
Fittings to centifugal seperator	0-1	Added 50-100% more equivalent feet to the linear feet of pipe based on industry rule of thumb	3	Based on piping isometric, tables in the ASHRAE pocket handbook and manufacturer's data for valve losses when available	with above	Based on the piping isometric and program parameters for valve and fitting losses
Centrifugal seperator	15-20	Estimate based on past experience	12	Based on manufacturer's data	12	Based on manufacturer's data
Pipe to upper pool	4-5	Estimate based on observation	1	Scaled from the drawings	6	Scaled from the drawings

Rightsizing to Optimize Pump and System Efficiency

Specifying the correct size pump ("rightsizing") is one of the most fundamental principles in designing pump systems. A pump that is carefully tailored to the needs of the system has been rightsized. **Sidebar 1** shows that an oversized pump can be addressed after the system has been installed, over the course of its operational life. Because of the increased efficiency of rightsized equipment, however, the benefit of lower first and operating costs will be greater if a pump and the system it serves are rightsized from the start.

Optimizing equipment and system efficiency, life cycle cost, and resources (material and labor) are all interrelated. While the engineering required to incorporate rightsizing principles into the design process may cause a modest increase in the engineering time and budget, the resulting installation savings at bid time will more than pay for the effort. As the project moves into its operational life, these first-cost savings combine with the operational savings delivered by rightsizing to greatly reduce life-cycle cost. The lifetime benefits can be further enhanced by commissioning the project to:

- bring operations and maintenance (O&M) into the design and installation process,
- facilitate verifying, testing, adjusting, and tuning the system, and
- train the operators in the design intent and the associated O&M requirements.

The most obvious place to begin optimizing is machinery that operates at peak efficiency under the design conditions. However in most cases, peak-load conditions represent only a small fraction of all operating hours. When considering the efficiency of the pump over the entire range of the load profile, the best overall efficiency may occur when the pump is selected for less-than-peak efficiency at design conditions.

Pumps are seldom sized optimally. In a 2005 survey of pump manufacturers, three of the seven respondents estimated that at least 60% of the pumps they sold were incorrectly specified by the contractor or owner/operator. Of that 60%, most errors (one respondent estimated 90%) were a result of inaccurately specifying an operating point, such as flow rate, required pressure, and net positive suction head.

(Tutterow, 2005). Clearly, the way pumps are applied can be improved. Toward that end, an association of pump manufacturers called the Hydraulic Institute initiated a market-transformation initiative call Pump Systems Matter™ to change and improve the process used to purchase pumps. To find out more and access guidelines, tip sheets, and free pump-system-modeling software, visit their website at www.PumpSystemsMatter.org.

One result of these issues is that pumps are oversized because unreasonably high safety factors are used when assessing pumping head and flow. An experienced contractor once observed, "I never got sued for putting in something that was too big." Properly applied, safety factors can help ensure the under all conceivable conditions the pump will achieve its design intent. In addition, modest safety factors provide the flexibility to adapt the performance of the pump and the system to the constantly changing needs of the facility. However, unreasonably high safety factors can result in an oversized pump. If undetected, an oversized pump would waste energy for its entire operating life. If detected, it would need to be modified to operate efficiently, but would most likely never operate as efficiently as a pump that was rightsized from the start. Oversizing can have a ripple effect on the size of the pump trim and the electrical equipment serving it, creating additional unnecessary first costs.

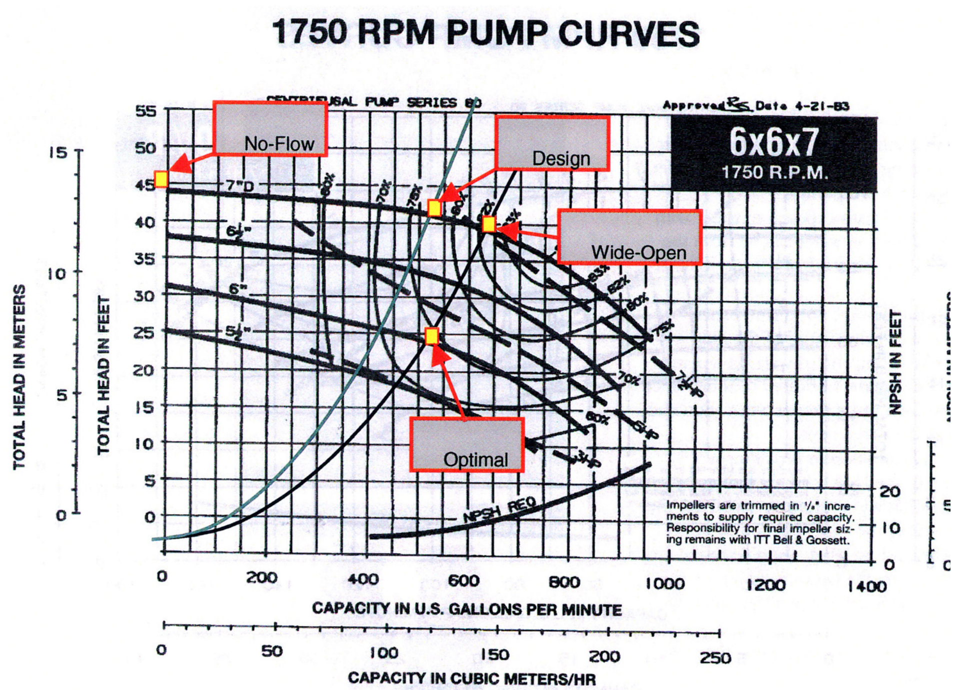
Balancing to Achieve the Right Maximum Flow

Pumps should be sized correctly. Even in the best of circumstances, however, the flow provided by a pump will be somewhat higher than is needed because of safety factors, providing for "room to grow," and the imprecision in identifying the required system head. In addition, if the system changes throughout the life of the building, the pump may no longer optimally match the new conditions. There are several techniques for optimizing the performance of an oversized pump in the field.

Throttling

Throttling is probably the fastest and most cost-effective way to optimize pump performance. This is done by closing down the pump-discharge valve until the required flow rate is achieved. The savings achieved will vary depending on the relationship between the slope of the pump curve and the associated brake horsepower curves. **Figure 13** shows the original design point from **Figure 6**. After installation, however, it was found that the system pressure drops were lower than expected, and the flow rate was 650 gpm, rather than the 550 required. The actual system curve is the one shown with the "Wide-Open" point on the 7" impeller curve. The pump-discharge valve can be closed down to artificially increase the system pressure drop and reduce the flow rate back to the design value. **Figure 13** shows that this reduced flow will slightly reduce the pump's power requirements (it crosses below the 7½ bhp line). (Note, however, that had it been throttled from 900 gpm down to 700 gpm, the pump's power requirements would actually have changed very little, since the pump curve is nearly parallel to the 7½ bhp curve). Throttling is often an immediately achievable, low-cost optimization technique that can be the first step toward reducing pump energy consumption while other more costly or complex techniques with greater savings potential are being assessed.

Figure 13: Typical Pump Performance Curve and System Curve



Example of an Impeller Trim

Before Trimming

After Trimming

Based on the results of the assessment of the various pump optimization alternatives, the owner of the example facility elected to trim the pump impeller. The pictures below illustrate the pump during the reassembly stage of the modification and the pre- and post-trim impeller size. After reassembly, the pump was retested and the new operating point was as projected, based on the original pump test. The end result was that the owner optimized the system so that it was only necessary to run one pump to produce 106 gpm of total flow at approximately 58 ft.w.c. using 2.75 bph. In contrast, the original design intent anticipated that both pumps would run to produce 106 gpm of total flow at 102 ft.w.c. using approximately 7 bhp. Testing demonstrated that one pump was able to produce in excess of 130 gpm of flow using 5.125 bhp prior to modification.

Impeller Trims

Impeller trimming is another way to reduce flow. Impeller trims allow the pump performance to be shifted down the system curve to the required flow rate rather than up the impeller curve. **Figure 13** shows that throttling up the 7" impeller curve from 650 gpm to 550 gpm will only reduce the required horsepower from just over 7.5 to just under 7.5 bhp. Moving down the system curve by reducing the impeller size to approximately 6", however, will result in a power requirement of only about 4 bhp to produce 550 gpm of flow. Note that moving down the system curve due to an impeller trim moves the pump away from its peak efficiency point down to about 75%, primarily due to the geometry changes between the impeller and the pump casing and their effect on the hydraulic phenomenon occurring in the pump. However, although the pump is actually operating less efficiently, it is doing less work and thus there will be significant energy savings.

Changing Pump Speed With a Motor Change

Modifying performance by reducing motor speed can save a similar amount of energy without reducing efficiency. In the HVAC industry in the United States, motor speeds of 1150 rpm, 1750 rpm, and 3500 rpm are considered standard, and motors that operate at these speeds are readily available. Occasionally the desired speed reduction can be achieved by simply using the next lower incremental motor speed. Because pump horsepower varies with the cube of the pump speed, such a change is usually achieved with a smaller motor, minimizing the cost of the change.

If the required speed is close to, but slightly above, the speed provided by the next incremental motor size, then it may be cost-effective to change to the lower speed motor and fine-tune to the desired flow rate by throttling.

Adjusting Flow to Changing Conditions

One of the most challenging aspects of building HVAC system design is that the systems must respond to a wide range of load conditions created by variations in local climate, occupancy, and use patterns. **Figure 13** illustrates the annual climate patterns for Sacramento, CA. HVAC systems in this area must deal effectively and efficiently with ambient temperature variations that can range from 28°F to 105°F over the course of a year, and from 57°F to 104°F over the course of a day, as occurred in late July, 2006.

Changing Speed with a Variable-Speed Drive

Changing speed reduces pump performance with a minimal effect on pump efficiency since the geometric relationships between the impeller and pump casing are not changed. In fact, because some of the factors associated with pump efficiency are a function of speed (for example, impeller drag and bearing friction), reducing speed can actually improve the pump's peak efficiency by one or two percent.

If the existing flow rate and pump speed are known, the pump affinity laws can be used to predict the approximate speed required to produce the new flow rate.

More often than not, the speed projected to modify the pump performance will vary significantly from one of the readily available motor sizes. In this situation or in situations where the required system flow varies under normal operating conditions, a variable-speed drive (VSD) can effectively optimize pump performance.

Using a Variable Speed Drive for Balancing

It is important to recognize the difference between using a VSD to match the pump performance to the system's continuously varying operating requirements and using a VSD to balance a pump (that is, providing a fixed reduction in operating speed to compensate for oversizing.)

In the former case, the drive is used to continuously vary the performance of the pump to match the continuously varying requirements of the system. Applying VSDs to pumps and their related HVAC systems to optimize performance over the range of actual operating conditions can save a significant amount of energy. Ideally, pumps for such applications will be selected so that they operate at full speed to achieve design flow and at reduced speed as the demand for flow drops for partial loads.

In contrast, a pump that is used for balancing provides a fixed, permanent reduction in pump speed because the performance provided by full-speed operation is never required by the system. This reduction in performance, while generally preserving the pump's efficiency, is not achieved without cost due to the losses that occur in a VSD. In addition, a VSD installation:

- represents a significantly higher first cost compared to other options for permanently reducing pump speed,
- represents increased operating complexity that must be understood by the operation and maintenance team, and;
- introduces an additional point of failure that would not otherwise exist.

Thus, the first cost and efficiency of a VSD used for balancing should be taken into account when considering alternatives. It is possible that the savings achieved at the pump by reducing speed will be offset by the drive. This is not the case when throttling or trimming an impeller.

Specifying Pumps

Ask for What You Need

Ultimately, specifying the performance requirement of a pump will depend on all of the parameters discussed in this design brief when assessed in the dynamic context of the system it serves. A detailed discussion of such an assessment is beyond the scope of this brief, but can be found in a number of resources:

- The Energy Design Resources website includes a number of briefs that cover topics that will effect pump selection either directly or indirectly. Examples include *Design Details, Integrated Design, Drive Power,* and *Chiller Plant Efficiency*.
- The Cool Tools™ project provides a detailed overview of the design and optimization of energy-efficient chiller plants, including the associated pumping applications. Many of the pumping considerations apply to other processes like distributing heating water. Cool Tools is hosted on the Pacific Energy Center website (http://www.pge.com/pec/) under the HVAC link. It is also being revised and will ultimately be moved to the Energy Design Resources website.
- The ASHRAE Handbooks, articles in the *ASHRAE Journal* (www.ASHRAE.org), and articles in industry publications like *Heating, Piping and Air Conditioning* and *Engineered Systems* are continually updating resources that discuss pumping-system applications and calculations. Many of these resources can be accessed from the internet.

Don't Ask for What You Don't Need

Of critical importance is tailoring your pump specification to the specific needs of the project. Boiler-plate specifications that ask for features that are unnecessary or do not make sense for a project can drive up project costs, create confusion, and in some cases, yield a product that is misapplied in the system it will serve. This can also leave the impression that the specifications will not be enforced.

Enforce Your Requirements

Enforcing the requirements of a well-thought-out specification is critical for both short-term and long-term success. In the short term, the success of your project will depend on verifying and ensuring that the engineering requirements you developed are met. Efficiency, performance, and ongoing operating costs are all at risk without enforcement. Misunderstandings, change orders, and scheduling problems could also ensue without engineering oversight of the procurement process. Thoroughly reviewing shop drawings is a key step in verification and enforcement, as discussed in the *HPAC* article titled "Installing Variable Frequency Drives" (Sellers, 2006). Before approval, the pump is an idea expressed in words and lines. After approval, it starts to become a physical reality.

In the long term, a designer's credibility could be compromised if he or she consistently fails to enforce the engineering details called out in the specifications.

Maintaining What Was Asked For

The continuing viability and adaptability of the design intent of the project is highly dependent upon the understanding of the design intent by the operating staff. Thorough training and documentation are essential in this process as is the periodic reassessment of the pump's performance against the original design criteria and commissioning results.

Conclusion

Research and experience indicates that pumps are often improperly applied and oversized, wasting energy and resources. Fortunately, the knowledge and tools exist to improve the situation. By understanding the details of pump construction and its interface with the system, the designer can take the first step in wisely spending the resources that come with installing a new pumping system. By requiring commissioning and training, the designer helps ensure that the system is optimized and that the optimized performance will persist.

The commissioning and operating team can complement the efforts of the designer by understanding and documenting the design intent of the pumping system and enhancing it with an O&M perspective that will ensure it persists for the life of the system. Functional testing at start-up and throughout the life of the system will ensure that the optimized performance envisioned by the design is realized and persists in the field.

Pump Optimization Bottom Line

Analysis of the pump and system curves developed for the example facility, revealed an opportunity to shut off one of the pumps while still providing sufficient flow. The analysis also showed an opportunity to trim the impeller of the operating pump to further reduce the flow without throttling the discharge valve. The net result of these changes resulted in 40% savings in annual pumping energy costs. The bottom line, however, is that these savings could have been even greater, and could have been accompanied with initial first cost savings from eliminating one pump and downsizing the other.

ITEM	Original	Revised	Savings
First Cost Savings			
Motor size	125	75	50
Nominal amperage	156	96	60
Motor efficiency	90%	90%	N/A
Nominal kW	104	62	41
Motor cost	$0	$0	$0
Wiring cost	$0	$0	$0
Total	**$0**	**$0**	**$0**
Annual Operating Cost Savings			
Hours of operation	3,000	3,000	N/A
Annual kWh	310,833	186,500	124,333
Electric rate $/kWh	$0.1000	$0.1000	$0.1000
Annual operating cost	**$31,083**	**$18,650**	**$12,433**

References

Bell and Gossett. (1965 - 1992) *Bell and Gossett Engineering Design Manual.* ITT Bell and Gossett Fluid Handling Training and Education Department, Morton Grove, Illinois.

Boyanton, Hugh. (n.d.) "Electrical Discharge Machining Of Bearings." Shaft Grounding Systems, Inc. Website: www.shaftgroundingsystems.com. Accessed Dec 2006.

Buffalo Pumps. (1959) "Centrifugal Pump Application Manual". Buffalo Pumps, a Division of Buffalo Forge Company, Buffalo, New York.

Daugherty, R.L. (1915) *Centrifugal Pumps*. McGraw-Hill Book Company, Inc. First Edition.

Energy Design Resources. (1998) "Design Briefs: Drive Power." Energy Design Resources website, www.energydesignresources.com. Accessed Dec 2006.

Durion Company. (1976) *Pump Engineering Manual.* The Durion Company Inc. Dayton, Ohio.

Houdek, John A. (n.d.) "Load Reactors Increase VFD System Performance & Reliability." MTE Corporation.

Jackson, Stewart and Liggett, F. (2002) "How to Match Motors to Variable Speed Drives." *Plant Engineering*., July 2002. http://www.reliance.com/pdf/misc/225412.pdf. Accessed Dec 2006.

Joliet Technologies. (n.d.) "Adjustable Speed Drives: What are they? How do they work?" *Application Information*. http://www.joliettech.com/adjustable_speed_drives-application_information.htm. Accessed Dec 2006.

Karassick, Igor, et.al. (1976) *Pump Handbook*. McGraw-Hill, Inc. New York, New York.

Karassik, Igor. (1981) "Centrifugal Pump Clinic." Marcel Dekker, Inc. New York, New York.

Puttgen, Hans. B. (1991) "Adjustable Speed Drives Directory - Third Edition." *CU-7544, Research Projects 2951-9 and 2951-12*. Georgia Institute of Technology, School of Electrical Engineering, Published by Electric Power Research Institute, December 1991.

Sellers, David A. (2006) "Installing Variable-Frequency Drives," *Heating, Piping and Air Conditioning (HPAC)*. November 2006.

Sellers, David A. (2005) "Right-sizing Pumping Systems," *Heating, Piping and Air Conditioning (HPAC)*. March 2005.

Sellers, David A. (2003) "Specifying Pumps," *Heating, Piping and Air Conditioning (HPAC)*. November 2003.

Stewart, Tom. (circa 1995) personal conversation with David Sellers, paraphrased.

Stone, G.C., Campbell, S.R., and Seddirw, H.G. (1999) "Adjustable Speed Drive Surges: How They Affect Motor Stator Windings," *IEEE Transactions, Paper 0-7803-5293-9/99*.

Tutterow, Vestal, Asdal, Robert, and McKane, Aimee T. (2005) "Making Pump Systems Matter," *ACEEE Summer Study*.

Upper Peninsula Power Company (UPPCO). (2005) "Motors: AC Induction Motors." UPPCO website at http://www.uppco.com/business/eba_35.asp. Accessed Dec 2006.

Upper Peninsula Power Company (UPPCO). (2005) "Making Motor Repair/Replacement Decisions." UPPCO website, http://www.uppco.com/business/eba_35.asp. Accessed Dec 2006.

U.S. D.O.E. The free MotorMaster software mentioned in the design brief can be downloaded from the U.S. Department of Energy's Industrial Technologies Program website at: http://www1.eere.energy.gov/industry/bestpractices/software.html#mm

Energy Design Resources provides information and design tools to architects, engineers, lighting designers, and building owners and developers. Energy Design Resources is funded by California utility customers and administered by Pacific Gas and Electric Co., San Diego Gas and Electric, Southern California Edison, and Southern California Gas, under the auspices of the California Public Utilities Commission. To learn more about Energy Design Resources, please visit our web site at www.energydesignresources.com.

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