energydesignresources

design brief

PUMPING SYSTEM TROUBLESHOOTING

Summary

This brief explores practical and proven troubleshooting and assessment techniques for identifying and solving pump system problems commonly encountered in existing buildings during retro-commissioning processes. It focuses on pump and system interaction, with a primary focus on the parallel pump arrangements that are common in heating, ventilation, and air-conditioning (HVAC) applications.

A case study format is presented for readers. Each example is presented in the following context:

- What were the indicators of the problem and how can the problem be corrected?
- Are there any "ripple" effects associated with the problem?
- How can the costs and benefits be assessed and can persistence be ensured?
- How can the problem be prevented in future projects?

Introduction

Based on a manufacture's survey and as reported by the Pump Matter Initiative, 60 percent of all pumps are improperly applied. Of those, 90 percent are not specified for the proper operating point. Troubleshooting, assessing, and solving pump system problems will improve performance and energy efficiency. Reliability and redundancy issues also may be addressed.

Design professionals, commissioning agents, and facilities engineering professionals will benefit from the techniques and examples presented in this brief. The information provides a fundamental understanding of

Identifying and solving pump system problems commonly encountered in existing buildings will improve performance and translate into energy savings.

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how pump theory and design practice apply to day-to-day operating environments. Each example describes a real-life scenario and provides viable options for resolution.

This brief is intended to supplement the Energy Design Resources (EDR) Design Brief titled *Centrifugal Pump Application and Optimization*. The companion brief focuses on pump theory, selection, specification, and optimization from a field perspective that includes gaining an understanding of how the pump will interact within the applied system. Assimilating the information in this brief and its companion document helps those who are responsible for the performance of various pumping systems. Identifying improvements in performance will translate to greater reliability and energy savings throughout the life of a pumping system. In addition, the knowledge gained will benefit future projects ensuring that pumps are correctly sized from the start, and are operated and tuned to the ever-changing requirements of the facility.

Pump Tests

Pump tests are integral to the pumping system troubleshooting process and frequently are used in assessing a problem. The premise behind a pump test is that for a given pump with a given impeller size operating at a given speed, a specific relationship exists between the pump head and the flow it will produce. This relationship is documented by the pump performance curve and is the direct result of the characteristics of the pump design. Similarly, for a fixed piping system, a specific relationship exists between the flow in the system and the head required to produce the flow. This relationship is described by the system curve.

If the pump curve and its associated system curve are plotted on the same grid, the point where the two curves intersect is the operating point for the pump/system combination. Pump tests leverage this relationship by measuring the differential pressure across the pump, then using the measured pressure and the pump's performance curve to determine the flow rate in the system and the associated system curve.

Pump Tests to Troubleshoot Pumping Systems

Pump tests can be used for fundamental troubleshooting purposes.

To Assess the Performance of a Pump or Set of Pumps: If the impeller size is known and a pump performance curve is available, then measuring the difference in pressure across the pump can be used to determine the flow that is being produced by the pump as well as its efficiency and the brake horsepower required to produce the flow. In addition, a shut-off test (forcing the pump briefly to a no-flow condition, also called dead-heading the pump) can be used to assess the condition of the pump's wear rings.

To Determine Impeller Size: If the pump is new or it can be assumed that the wear rings have been subjected to little or no wear, then a shut-off test can be used to determine impeller size. Specifically, the differential pressure produced with no flow is the point where the impeller curve crosses the 0-flow line on the flow axis.

To Assess System Performance: An informal survey of facilities engineers revealed that only a minority of pumping systems are equipped with flow meters. In the same survey, it also was determined that virtually every pumping system is equipped with one or more pumps. Therein lies the purpose of a pump test. If the pump performance curve is available and the pump impeller size is known, then a pump test can be used to identify a point on the system curve. Then, the system curve can be used to project what will happen in the system as the flow is varied.

Pump Test Limitations

A pump test is a fairly simple test to perform and can be accomplished with a minimum amount of equipment. However, there are limitations that should be kept in mind when using this technique.

- *The test is only as good as the pump curve*: The test depends on having an accurate pump curve for the pump that is under test. Without it, the data can not be interpreted.
- *The impeller size must be known*: If the impeller size is unknown, the results may be inconclusive. Frequently, a shut-off test can be used to verify impeller size, though this procedure must be performed with caution to prevent over-pressurization of the piping between the pump and discharge valve, which can cause damage to the pump and the system.

Pump testing procedures and general testing guidance may be found in a number of sources. One resource is the Functional Testing Guide, developed by the U. S. Department of Energy in collaboration with a number of states. It is available online at www.peci.org/ftguide.

- *Results on the "flat" part of the curve are difficult to interpret*: If the test results are on the "flat" or near horizontal part of the performance curve, then they can be difficult to interpret because a minor change in head can be associated with a major change in flow.

Pump testing concepts are illustrated in **Figure 1**. *The Centrifugal Pump Application and Optimization* Design Brief, the companion brief to this document, provides more detail about the concepts.

Pump Testing Procedures

Functional tests, including those used to test pump, typically contain the following elements. Sample testing procedures may be available online from various resources.

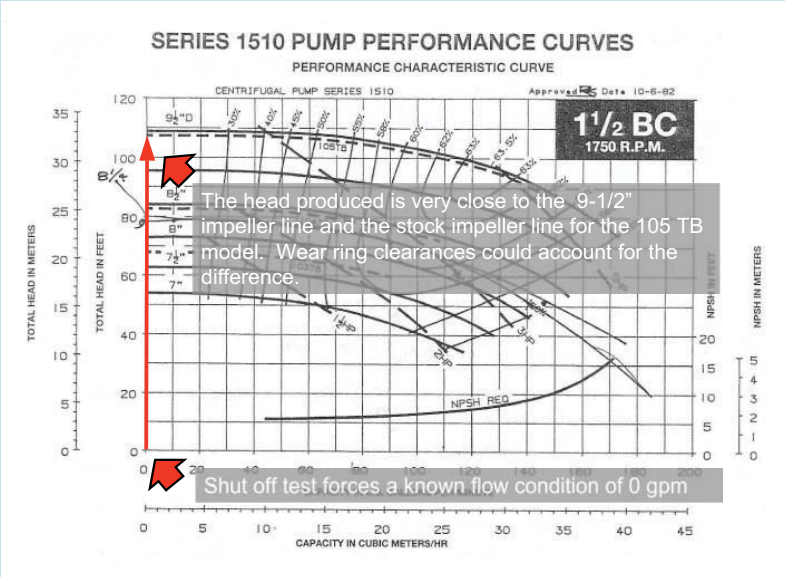
- *Statement of Purpose*: This section of the procedure provides a brief statement of why the test is being performed, which often ties back to the system's design intent.
- *General Instructions*: Information regarding how test results should be documented and other general information is provided in this section.
- *Equipment Required*: Tools and special equipment required for the test are listed including any special calibration or certification requirements.
- *Acceptance Criteria*: The criteria used to determine if the test is passed or failed are documented and often tie directly to the test goals and design intent. Some tests are run only for information gathering purposes and may not have acceptance criteria.
- *Precautions*: Most tests place the system under test at some level of risk. Any special precautions that need to be taken to control or minimize the risk are typically listed in this section.
- *References*: Information such as pump curves or system diagrams that may be useful to have while performing the pump test or to prepare for the test is listed in this section and may be attached to the test.
- *Test Procedure*: This section is the heart of the test and documents the specific steps to be taken, including provisions for recording the test results.

Figure 1: Steps in a Typical Pump Test

Pump testing concepts are illustrated below for a pump test where the differential pressure at shut-off (0 gpm) was 106.4 ft. w.c. and the differential pressure wide open was 93.9 ft. w.c.

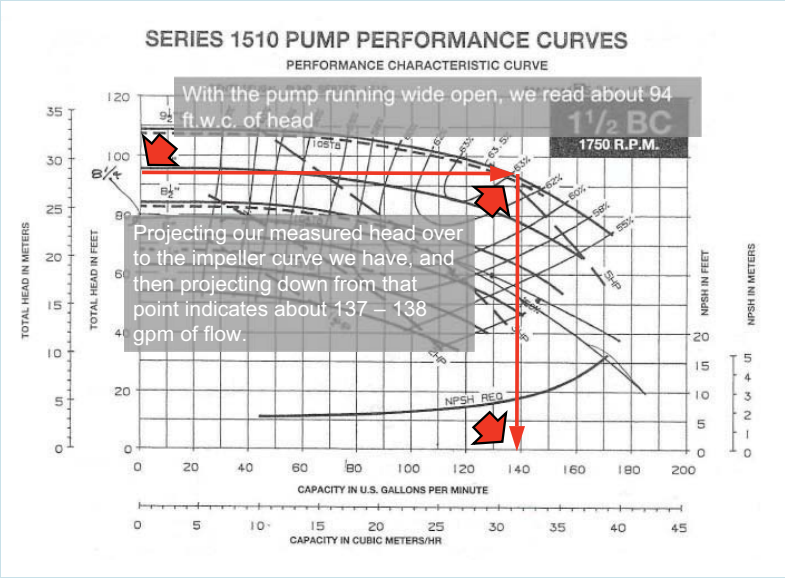
Step 1 – Verify Impeller Size with a Shut-Off Test

Forcing the pump to 0 gpm verifies the impeller size.



Step 2 – Determine Wide Open Operating Point

With the impeller size known, the wide open head reveals the operating point.



Pump Terms

For a complete list of pump terms, visit www.pump-manufacturers.com and look for A-Z Glossary.

B.H.P. - brake horse power.
The actual amount of horsepower being consumed by the pump as measured on a pony brake or dynamometer.

Cavitation - bubbles form in the fluid low pressure area and collapse in a higher pressure area of the pump, causing noise, damage, and a loss of capacity.

Ft. W.C. - feet of water column. Pumping head is usually given in feet of water column or equivalent metric units.

Head - the equivalent height of the liquid. The term head is used instead of pressure in the centrifugal pump industry.

HP - horse power. The term is used to describe pump sizes.

kW - kilo-watt-hour. It describes energy use.

Net Positive Suction Head (NPSH) - the positive pressure above the pump fluid's vapor pressure that needs to exist at the inlet flange to prevent cavitation.

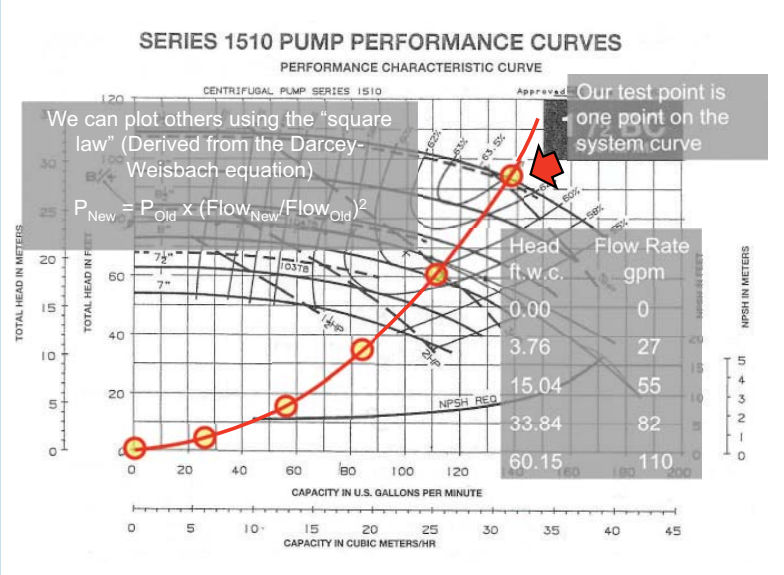
RPM - revolutions per minute and is a typical unit of measurement for pumps.

VFD - variable frequency drives and is a system for controlling the rotational speed of a pump motor.

Figure 1: Steps in a Typical Pump Test (continued)

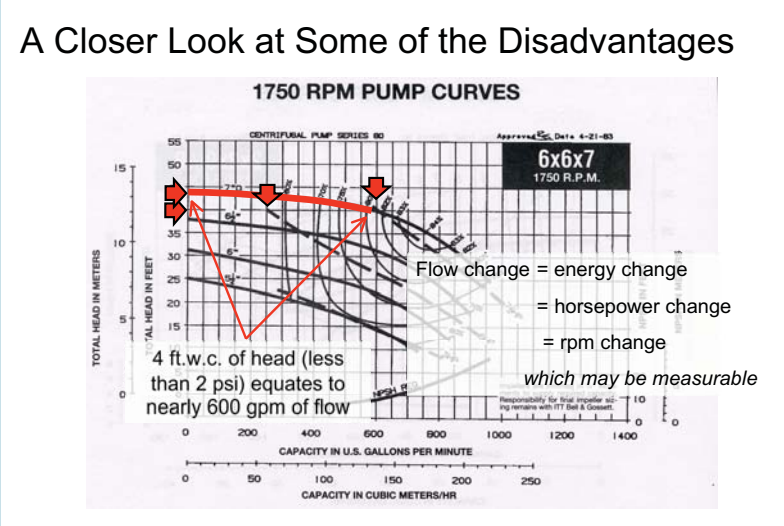
Step 3 – Plot the System Curve

The operating point becomes the first point on the system curve.



Pump Test Limitations

If the operating point is on the flat part of the curve, a test based solely on pump head may be inconclusive. Taking additional data may allow a conclusion to be drawn.



Source: Pump Curves Courtesy of Bell & Gossett

- *Return to Normal*: This section outlines the procedures that need to be used to return the system to a normal operating state. It also may include a space for the test team to sign off on the test.

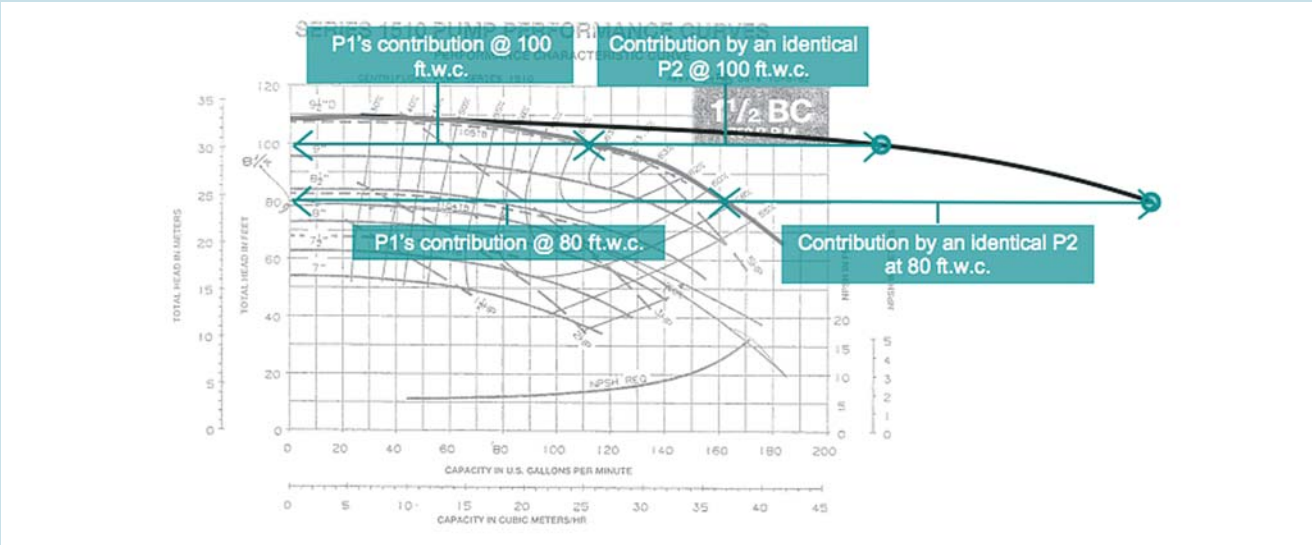
Parallel Pumps

Parallel pumps are common in HVAC systems for a number of reasons including reliability and redundancy, and the need to provide incremental steps of capacity to match incremental, constant volume loads. Also, parallel pumps can provide incremental capacity to match the phases of a project or allow for dimensional constraints that prevent fitting one large pump capable of producing the required flow into the available space.

When two pumps are placed in parallel, their suction and discharge connections are referenced to the same pressure via the header system interconnecting them. As a result, the flow they produce together is equal to the sum of the flow produced by each pump at the same differential pressure or head. For identical pumps with identical impellers, this means that the flow for the two pumps will be twice the flow produced by one pump at all operating conditions as illustrated in **Figure 2**.

Figure 2: Pump Curve for Two Identical Pumps Piped in Parallel

The heavy lines superimposed over a manufacturer's pump curve illustrate the performance curve (heavy black line) that is generated when two identical 1-1/2 BC pumps with 9-1/2" impellers (the heavy gray impeller curve) are piped in parallel. Specifically, each pump contributes an equal amount to the total flow rate at the same head. The horsepower used by each pump is the horsepower associated with its contribution. For instance, at 220 gpm total flow (the first or left-most highlighted point on the combined curve), each pump is providing 110 gpm at 100 ft. w.c. using about 4.75 bhp.

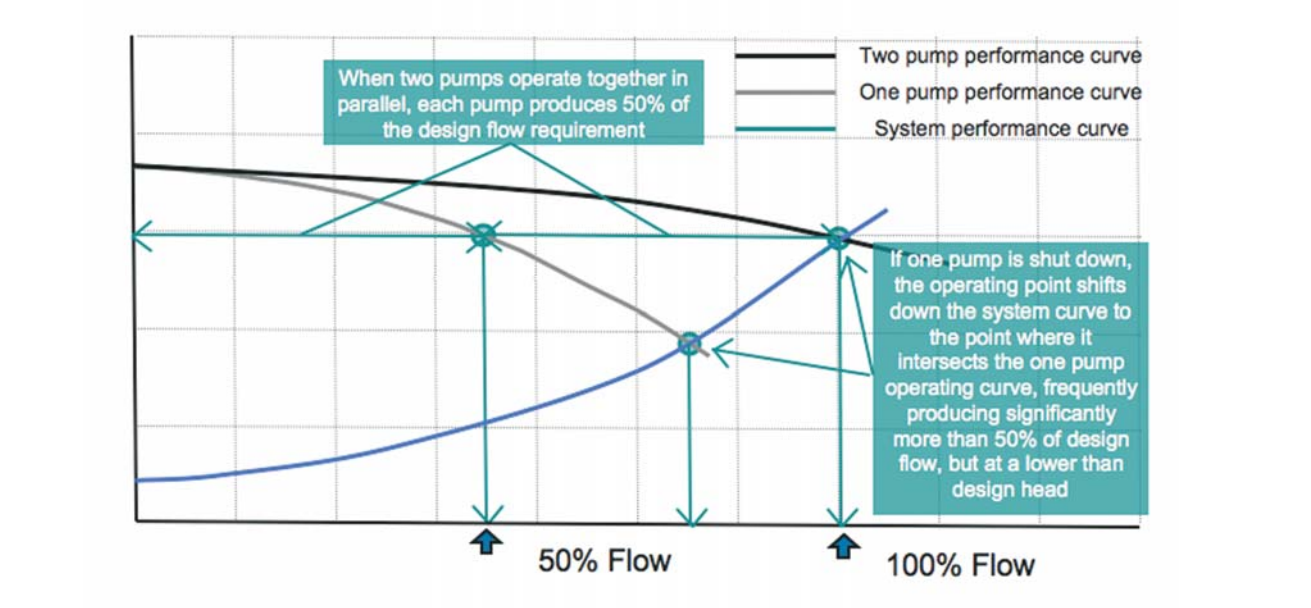


Source: Pump Curve Courtesy of Bell & Gossett

Figure 3 illustrates how a system curve might interact with the pump curves for two parallel pumps piped in parallel.[1] When operating with one pump, the amount of flow produced, with all other things being equal, can be significantly greater than 50 percent of design. However, the one pump operates at a lower head than was available with two pumps operating. The magnitude of the difference between the flow contribution of two pumps operating in parallel versus one pump operating alone will be a function of how "steep" the system curve is (i.e. how quickly the head required rises with flow). For fairly flat system curves, one pump can frequently provide 60 to 75 percent of the flow produced by two pumps operating together. In contrast, for a steep system curve, the second pump may only produce a three to five percent increase in system flow when it is started.

Figure 3: System Curve Interaction with Two Identical Parallel Pumps

This figure superimposes a system curve (the flow versus head operating curve for the piping system served by the pump) on a generic case of the parallel pump situation depicted in Figure 2. The pumps will operate at the points where their impeller performance curves (heavy gray and black lines) intersect the system curve (the heavy light blue line). As a result, one pump operating alone will deliver more than 50 percent of the design flow produced by two pumps operating in parallel.



Source: David Sellers, Facility Dynamics

Dissimilar Pumps in Parallel

If dissimilar pumps—pumps with different performance characteristics, especially those with radically different pump curves—are piped in parallel, things can become more complicated. Dissimilar parallel pump arrangements are often obvious. For example, two completely different pumps in terms of make or model can be located right beside each other and piped to the same header. However, it is important to remember that just because two parallel pumps appear to be identical does not necessarily mean that they are. The impeller in one pump could be trimmed from the size indicated on its nameplate while the impeller in the other is still the nameplate size as shipped from the factory. **Figure 4** illustrates what happened when the impeller in one of the pumps depicted in **Figure 2** was trimmed to optimize its performance in the system while the impeller in the other pump was not trimmed.[2]

To be in parallel from a hydraulic standpoint, two pumps only need to share the same point of connection on the suction and discharge. Physically, they may be in different locations. This physical separation often can make it difficult to recognize a parallel pump situation in the field. Frequently, the pumps in this type of situation are not the same, and mysterious problems ensue, as illustrated in the following case study.

During the first weeks of operation, the designers of a flywheel tank-type domestic hot water system serving a campus received complaints that recirculation flow intermittently was lost. Generally, the recirculation flow seemed to exist during periods of heavy use, but did not exist during low-load periods. As a result, when the load did increase, there was a significant and unacceptable time delay between when a user opened a tap and when they received hot water.

The heat exchanger serving the system and its associated storage tank were located in a central plant. The loads served were scattered across a number of campus buildings located several hundred feet apart. **Figure 5** illustrates the system in question. The initial reaction of the design team was to focus their troubleshooting efforts on the recirculation pump. However, a field investigation supplemented by data logging revealed that the pump was operating continuously during the periods

Typical domestic hot water system (DHW) design issues include providing the capacity to handle intermittent, high-flow, short-duration peak loads with little or no capacity or flow required between peaks. A second issue is minimizing the amount of time that a user must wait to receive hot water after opening a faucet.

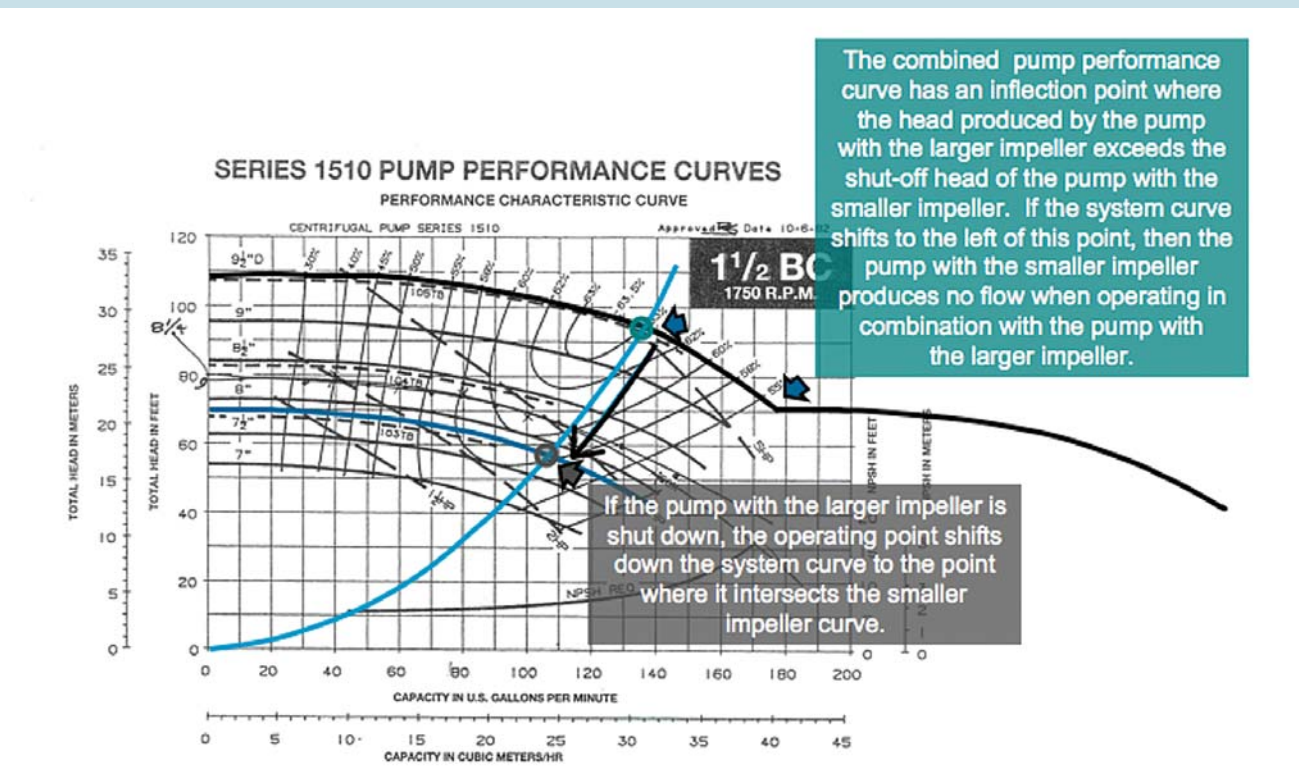
Flywheel-type systems use a storage tank that is charged by a circulating pump during low load conditions to store capacity for peak loads.

A recirculation pump maintains flow through out the distribution network, making hot water immediately available to the branch connections and minimizing the wait time for users.

when recirculation seemed to be lost. The team then focused their effort on the system balance, believing there must be some sort of elusive balancing issue that led to the disruptions in service. Unfortunately, the results of their efforts were fruitless. All of the system balancing valves were located appropriately and properly adjusted. The problem continued to be elusive, until the project engineer realized that the system diagram illustrated in **Figure 5** also could be drawn as illustrated in **Figure 6**. The revised system diagram made it obvious that the flywheel tank pump and recirculation pump were essentially piped in parallel.

Figure 4: Two Dissimilar Pumps in Parallel and Their Interactions with a Steep System Curve

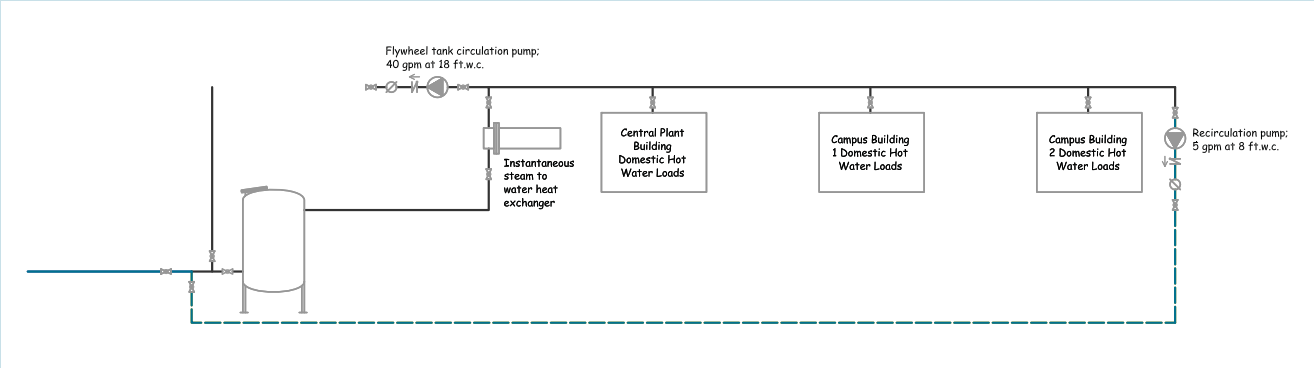
When two pumps with different performance characteristics are piped and operated in parallel, the combined performance curve will have an unusual, non-contiguous shape; it does not matter if the differences between pumps were the result of the pumps being a different make or model, or the result of an impeller trim in only one pump. The curve below illustrates what happened when the impeller on one of the **Figure 2** pumps was trimmed while the other was not. If the pump with the untrimmed impeller is operated, the operating point on the system curve is such that the pump with the smaller impeller produces no flow, if it is started. This is because the flow produced by the pump with the larger impeller generates more head in the system than the shut-off head of the pump with the smaller impeller. Power readings taken on the pump with the smaller impeller will verify that it is in fact under no load when operated concurrently with the larger pump. If the pump with the larger impeller is shut down, then the system operating point shifts down the system curve to the point where it crosses the smaller impeller curve. The flow in the system drops to 106 gpm, and the smaller pump loads up and draws approximately 2 kW (2.5 bhp at a motor efficiency of approximately 90 percent).



Source: Pump Curve Courtesy of Bell & Gossett

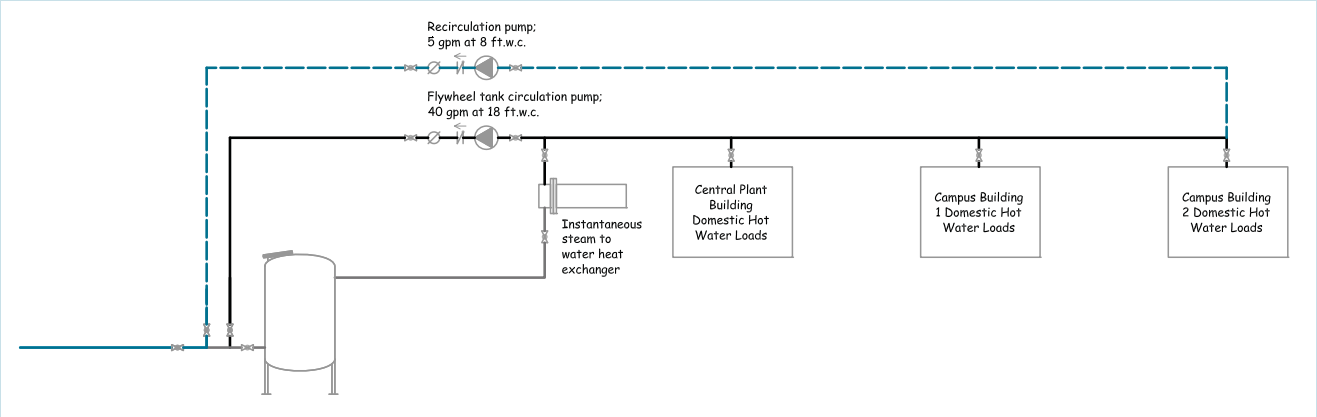
Figure 5: A Flywheel Tank-Type Domestic Hot Water System

The system diagram below illustrates the flywheel tank-type domestic hot water system that was installed on a recent project. The recirculation pump that charges the storage tank is located at the central plant while the recirculation pump that ensures a rapid response to a demand for hot water is located in a remote building hundreds of feet away.



Source: Dave Sellers, Facility Dynamics

Figure 6: A Different Perspective on the System Illustrated in Figure 5



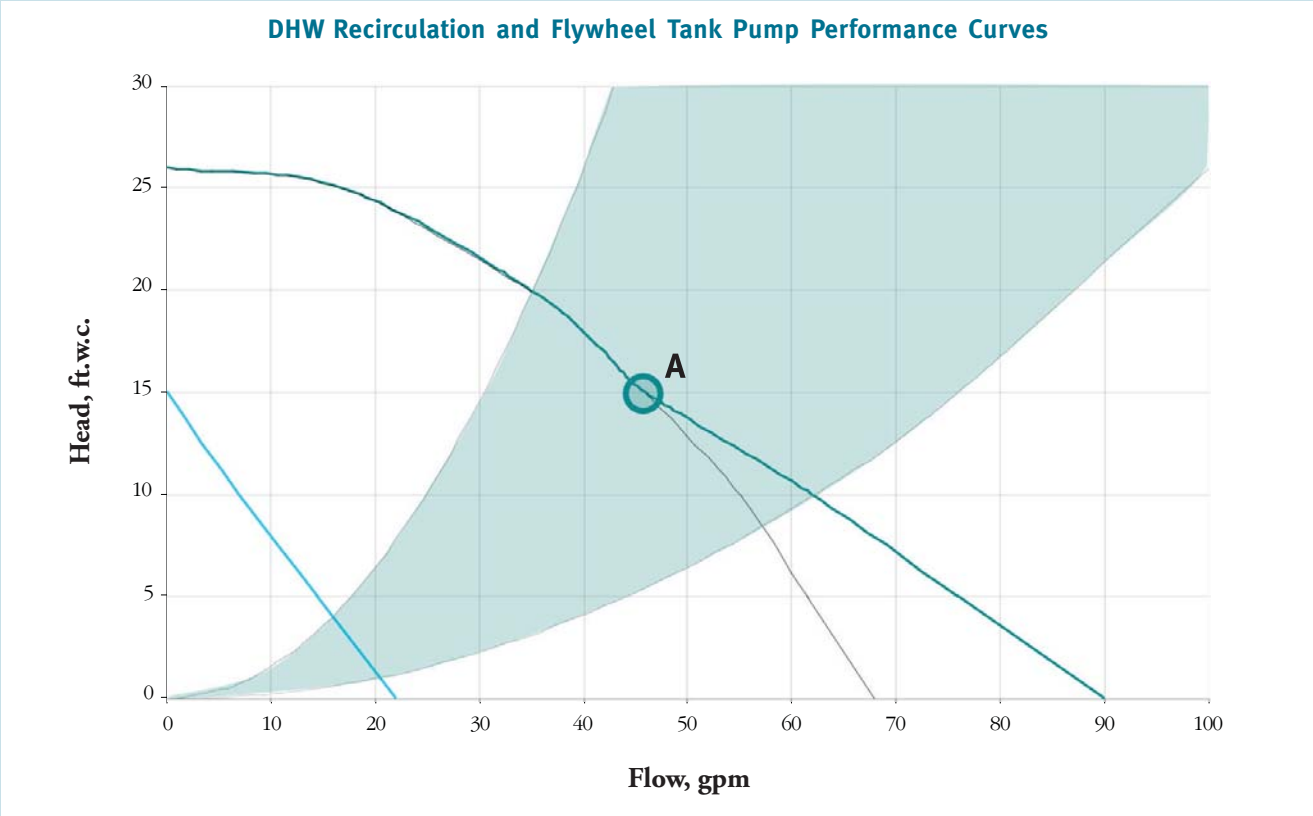
Source: Dave Sellers, Facility Dynamics

Recognizing the potential for problems associated with dissimilar pumps piped in parallel, the project engineer plotted the combined pump curve for the arrangement. The result is depicted in **Figure 7**. The combined pump performance curves made it clear that the potential existed for the differential pressure or head produced by the flywheel tank pump to "dead head" the recirculation tank pump under the right operating conditions. Field testing revealed that the system curve for the piping network varied over a range that included conditions where both pumps would produce flow, and conditions where the head from the flywheel tank pump would exceed the shut-off head for the recirculation pump and eliminate recirculation flow. The system curve shifted depending on

the demand placed on the system by the users. Though, somewhat erratic in its magnitude, the shift was predictable in terms of its occurrence due to the fairly repetitive nature of domestic hot water loads.

Figure 7: The Combined Pump Curves for the Parallel Pumps in Figure 5 and Figure 6

This diagram illustrates both the individual and combined pump curves for the pumps in the domestic water system with the mysterious loss of flow. The light blue line is the curve for the recirculation pump, the black line is the curve for the flywheel tank pump, and the teal line is the combined pump curve. The inflection point in the combined pump curve is point A. The shaded teal area represents the range of system curves that were generated in the system as users opened and closed faucets and other domestic hot water loads. When the system curve shifts to the left of point A, the head produced by the flywheel tank pump will deadhead the recirculation pump and there will be no recirculation flow. In contrast, when it shifts to the right of point A, the recirculation pump can begin to circulate water.



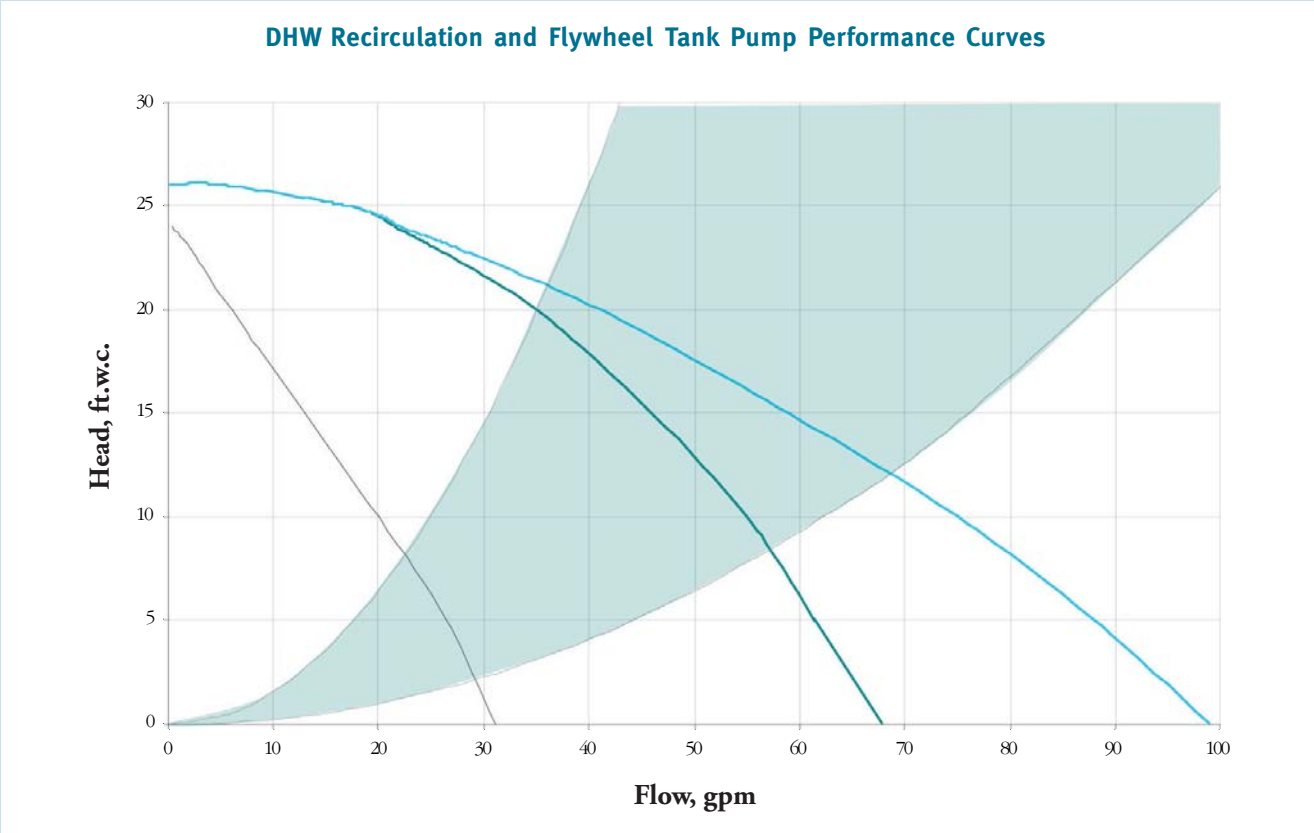
Source: Dave Sellers, Facility Dynamics

Once the cause of the problem was identified, the solution was obvious. The pump selections were modified to produce the required flow at similar heads. The result was the replacement of the original recirculation pump with a pump that had a curve that was more compatible with the flywheel tank pump's curve as illustrated in **Figure 8**. This modification solved the problem and had the added advantage of being a very persistent solution since undoing it would require that somebody

inadvertently replace one of the existing pumps with a pump that did not have the correct characteristics. This is more likely than it may sound, especially with a small in-line pump where the perception can be that one little pump is just as good as any other. By taking the time to modify the design documents to reflect the revised pump selection and by involving the staff in the modification process, the design team took important steps in ensuring the persistence of the solution.

Figure 8: The Modified System

Selecting the recirculation pump to provide its design flow at the same head as was required for the flywheel tank pump resulted in a selection that had a shut-off head that was more closely aligned with the shut-off head of the flywheel tank pump. As a result, both pumps were capable of maintaining flow under the range of operating conditions seen by the system (shaded area) as the users opened and closed faucets and other domestic water loads.



Source: Dave Sellers, Facility Dynamics

Other Parallel Pump Examples

Parallel Pumps Serving Incrementally Varying Loads

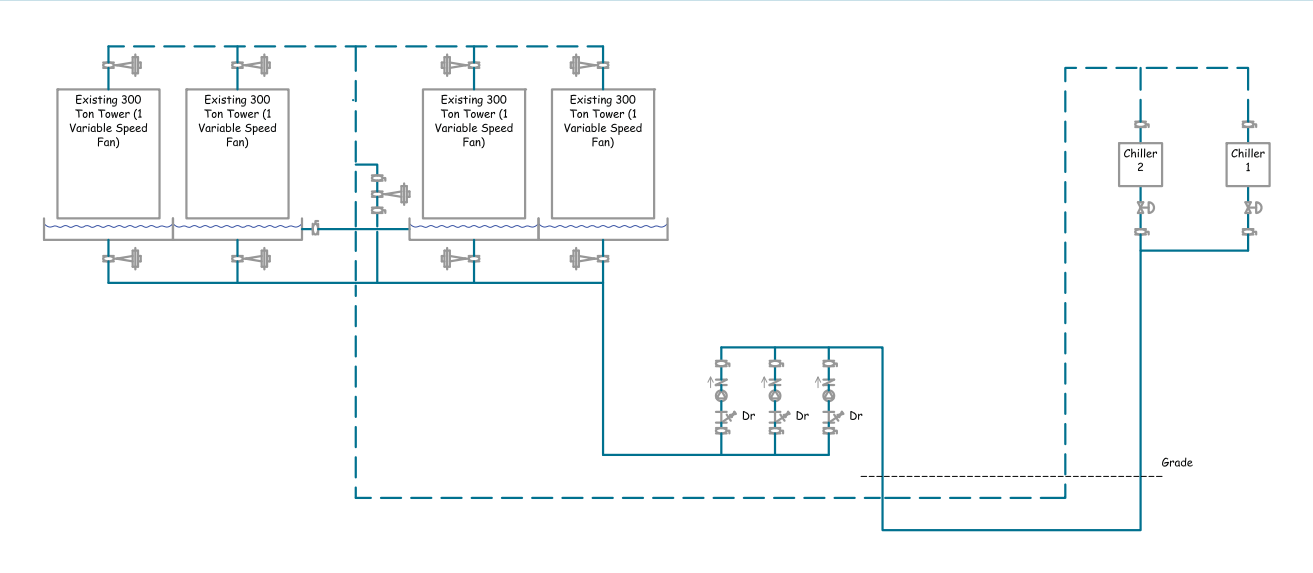
It is very common in HVAC systems to have parallel pumps arranged to serve loads that vary incrementally. The condenser water systems serving chiller plants are good examples of such a condition, as illustrated in **Figure 9**. There are several characteristics of this system type that make it an ideal optimization target during design or in a retro-commissioning process.

There are long piping runs that are common to the operation of one or both pumps. For the system illustrated in **Figure 9**, more than 200 lineal feet of pipe exists each way between the pump/cooling tower location and the chillers. There also is nearly 100 lineal feet of common piping between the suction side of the pumps and towers.

There are parallel equipment paths that are common to the operation of one or both pumps. For the system illustrated in **Figure 9**, the four cooling tower cells represent such a situation since all cells are open for flow no matter how many chillers are operating.

Figure 9: A Condenser Water System with Parallel Pumps Serving Incrementally Varying Loads

In this system, three pumps are piped in parallel to serve two chillers, which also are piped in parallel. Heat is rejected through four cooling tower cells in parallel arrangement. One pump is associated with the operation of each chiller, and the third pump serves as a standby, coming on line if one of the other pumps fails. All tower cells have flow over them in all operating modes. The isolation valve at each chiller opens when the chiller operates and closes when it is off line.



Source: Dave Sellers, Facility Dynamics

The loads served have a specific flow rate associated with their operation. In the **Figure 9** system, the chillers represent this type of load. To operate and achieve design performance, the chillers must receive their design flow rate. Flow rates in excess of design also will allow the chiller to operate and may even provide a modest improvement in kW per ton due to lower condensing temperatures. However, the improvement at the chiller could be easily overwhelmed by the pumping energy required to produce the extra flow. Flow rates below design will work up to a point and will have a reduced pumping energy requirement associated with them, all other things being equal. However, the operating penalty imposed on the chiller due to elevated condensing pressures may eradicate any savings achieved at the pump. At some point, the chiller will trip off due to a high head pressure safety if the flow becomes too low.

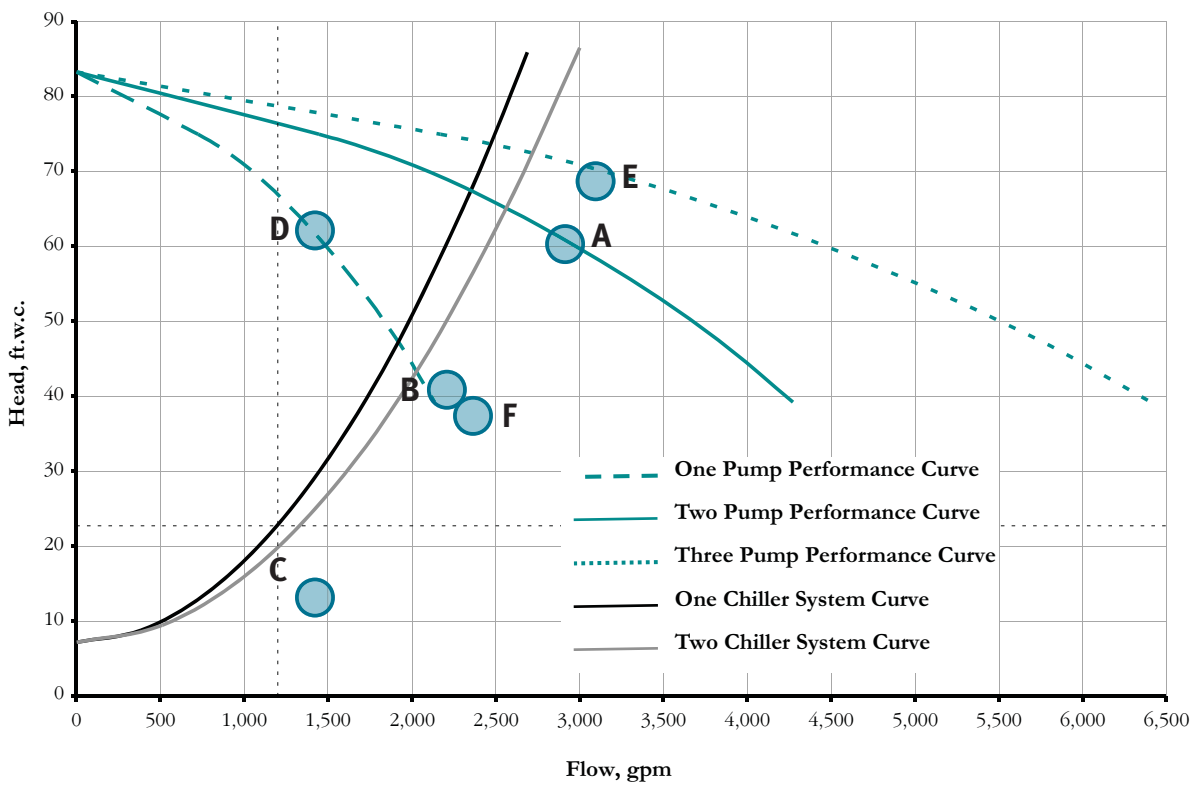
Figure 10 illustrates the tested performance of the condenser pumping system depicted in **Figure 9**. While the system successfully achieves the design flow required with two chillers in operation, it also, by its nature, will deliver more flow than is required when one pump is operated with one chiller. This over-delivery represents an optimization opportunity that can be realized during design if it is recognized at that time. However, it also can be realized retroactively in the field. For systems that spend a significant time at part load, the savings potential can be quite significant. In the example system, over 6,000 hours a year are spent with one chiller in operation and the following optimization options were considered:[3]

Throttle to the required flow for one chiller when operating one pump with one chiller: Throttling to design flow is the typical solution implemented in the field when a pump is over-delivering. This may or may not reduce the pumping energy required depending on where the operating points are on the pump's performance curve. For the pump in the example, the bhp required will be nearly constant at 30 bhp for flows above approximately 1,400 gpm. Below 1,400 gpm, the bhp begins to drop as flow is reduced and head is increased. In the example system, throttling to 1,200 gpm reduced the pump power from 30 bhp to 26 bhp. However, this solution is complicated on the example system by the fact that throttling is not desired when the pump runs concurrently with the second pump. This could be addressed in a couple of ways:

Figure 10: Performance Curves for the Figure 9 Condenser Water System

The design flow for this system with two chillers in operation was 2,400 gpm at 65 ft. w.c. (1,200 gpm per pump using about 26 bhp). The installed system achieved near design results with two chillers in operation (gray system curve; 2,550 gpm or 106 percent of design; point A). But, when the installed system is operated with one chiller and one pump (black system curve), it produces 1,900 gpm of flow at approximately 48 ft. w.c. of head (point B), fully loading the 30-hp pump motor. Moving down the one chiller system curve to the 1,200 gpm flow rate required by one chiller reveals that it could be achieved with approximately 23 ft. w.c. of head. The manufacturer's test data for the pump in question indicates that the pump would only require approximately 7 bhp at this condition. Thus, adjusting the pump's performance with one chiller running to provide only the head and flow required versus allowing the pump to "run out its curve" has the potential to save approximately 23 bhp in pump energy. The obvious and common solution is to throttle the pump to the design flow requirement (point D) which would reduce the bhp requirement to approximately 26 bhp.

It also is interesting to note that operating the third redundant pump concurrently with the other two pumps will provide very little additional benefit in terms of flow. Specifically, the three pumps would deliver a total of 2,700 gpm at 72 ft. w.c. using approximately 26 bhp each (point E). Or stated another way, operating the third pump provides an additional 250 gpm at the cost of 26 bhp when compared with the performance achieved with two pumps in operation.



Source: Dave Sellers, Facility Dynamics

- A manual procedure could be used where under-operators would manually throttle the pump to a predetermined flow rate any time one chiller was in operation. Such an approach would have a relatively low first cost but would be unacceptable for the facility owner served by the plant due to the automated sequencing of the chillers that is currently in place.

- The automatic chiller sequencing program and control hardware could be modified to use the automatic condenser isolation valves to throttle flow to the active chiller if one chiller is operating, open fully if both chillers are operating, and close fully if the chiller served is off line. Since the control valves are already equipped with pneumatic actuators, it would be fairly simple to modify the control hardware to provide an analog signal to them rather than a digital (open/close) signal. Field testing could be used to determine the analog command necessary to position the valve to throttle appropriately if only one chiller was on and the system could be programmed accordingly.

Trim the impeller of the back-up pump to provide the required performance. This approach involves modifying the existing back-up pump to provide the required level of performance for one-chiller operation. It has the advantage of significantly increasing the savings relative to throttling and a lower first cost than other options with higher savings potential. However, it also has a significant disadvantage. The modified pump will become a dissimilar pump relative to the unmodified pumps. As such, it will have insufficient head to operate in conjunction with one of them should the other fail. Thus, the redundancy of the system is compromised by the modification.

Install a separate, smaller pump sized for the performance required with one-chiller operation. This approach has the advantage of tailoring the pump selection to the specific requirements associated with one-chiller operation while maintaining the redundancy offered by the third existing pump sized to provide the necessary flow and head with two chillers in operation. The specific requirements will be reliable numbers since they are based on a field test of the existing system rather than a design estimate of the head required at the desired flow rate. On the down side, this approach requires piping modifications, electrical power work, control work, and the purchase of a pump, all of which increase costs.

Install a VFD and operate the back-up pump at a reduced speed to serve one chiller. This approach has the advantage of allowing the existing back-up pump to provide the desired performance with one chiller in operation without compromising the redundancy it provides. Normally, the

variable-speed pump would be operated at reduced speed to serve a one-chiller operating state. If two chillers are required, the two fixed-speed pumps would be operated. This approach delivers peak efficiency by eliminating the losses associated with a VFD from the operating equation. However, if one of the fixed-speed pumps failed with two chillers required, the variable-speed pump can be ramped-up to full speed to provide back-up capability. On the down side, the variable-speed drive introduces some efficiency losses into the system when operating on one chiller that would not be present in the case of a fixed-speed pump selected so its peak efficiency point matched the required operating parameters.

Table 1 contrasts the savings and costs associated with the various options. As can be seen, all of the options are attractive from a simple payback basis. However, if cash flow were an issue, either of the throttling options would generate viable savings at an acceptable cost. In the example given, the owner was willing to invest in measures that delivered greater savings, though he was concerned about maintaining redundancy. Thus, the impeller trim option was eliminated from consideration. The speed reduction approach was selected because it delivered the best simple payback.

Table 1: Optimization Options for the System Depicted in Figure 9 and Figure 10

Optimization Option	Energy Savings		Implementation Cost, $	Simple Payback, years
	kWh	$		
Manual throttling	17,880	$2,146	$1,000	0.47
Automatic throttling	17,880	$2,146	$4,000	1.86
Impeller trim	87,544	$10,505	$5,170	0.49
Speed reduction	129,436	$16,309	$14,828	0.91
Smaller pump	109,915	$13,190	$18,600	1.41

Source: Dave Sellers, Facility Dynamics

Oversized Parallel Pumps Serving Incrementally Varying Loads

Oversized parallel pumps that serve incrementally varying loads are a special case of the situation discussed in the preceding section. As previously mentioned, oversized pumps provide a common opportunity for energy savings in the field. It is not uncommon for a pair of parallel pumps to be oversized to the point that field testing reveals that one pump can serve loads originally intended to be served by two pumps.

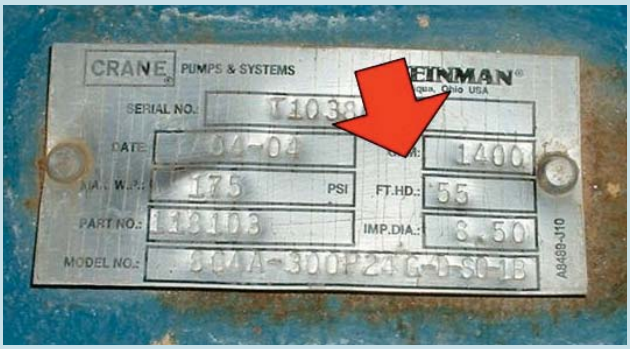
Two field indicators help identify opportunities of this type. The obvious one is a heavily throttled valve on the discharge of a pump. The valve is most likely throttled to force the pump back to design performance. Though, the pressure drop it creates is dissipating energy from the fluid stream that was just placed there by the motor only moments before.

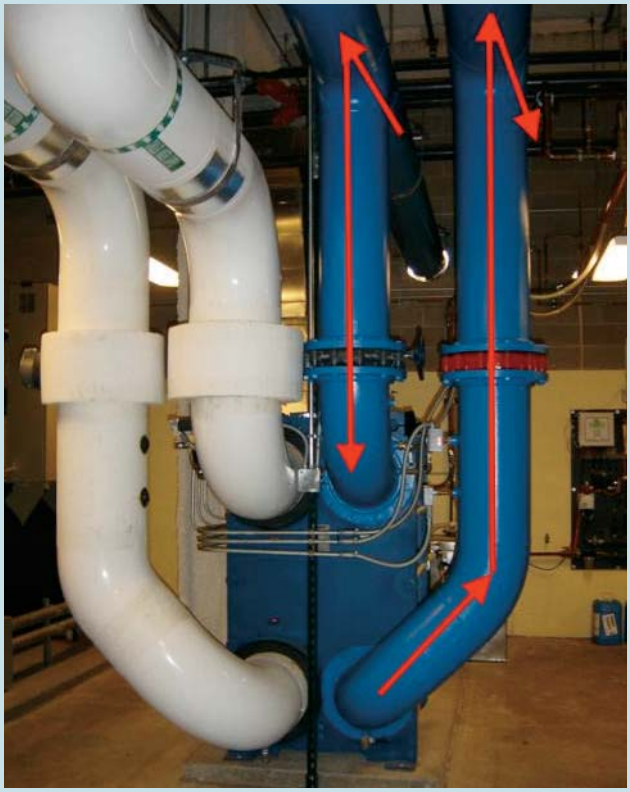
A less obvious indicator is a pump nameplate rating that does not make sense in the context of the physical installation in the field, as illustrated in **Figure 11**. The system shown is a condenser water system serving two chillers and a plate and frame heat exchanger. There are two operating modes.

- *Wet Economizer Mode*. In this mode, the cooling towers are controlled to drive the condenser supply temperature down into the upper 40°s/low 50°s F (degrees in Fahrenheit) to provide a "wet economizer" cycle via the plate and frame heat exchanger.
- *Conventional Condenser Water System Mode*. In this mode, the cooling towers supply condenser water to the chillers at 75°F to 85°F in a typical condenser water application.

Automatic isolation valves on each chiller and the heat exchanger isolate them from the system when they are not in service. Because of the difference between the estimated head and the pump nameplate on the project, the system was targeted for testing as part of a retro-commissioning effort that was undertaken at the facility. The results of the test are shown in **Figure 12**.

Figure 11: A Field Indicator of a Parallel Pump Optimization Opportunity





The picture at the top is the nameplate from one of the pumps illustrated in the middle picture on the left. The middle and lower pictures illustrate the piping circuit served by this pump, which includes two cooling tower cells (middle picture) serving a plate and frame heat exchanger (bottom picture) and two chillers (located immediately to the left of the heat exchanger).

Given the fairly straight piping run, the nameplate pump head of 55 ft. w.c. seems surprisingly high. Assessing the pump head required in the field and contrasting it with the nameplate can be surprisingly easy as is illustrated in the tabulation below.

Item	Loss, ft.w.c.	
	Low End	High End
Heat Exchanger	15	20
Tower Lift	8	10
Piping	4	6
Fittings	2	6
TOTAL	29	42

The projected losses are simply field estimates based on past experience or physical distances. Contrasting the estimated pump head requirement of 29 to 42 ft. w.c. with the pump nameplate of 55 ft. w.c. indicates there is at least a modest if not a significant opportunity for optimization. As a result, the system was targeted for further testing.

Source: Dave Sellers, Facility Dynamics

The test results revealed two viable optimization strategies for this system.

- *Option 1*. It may be possible to operate the system using one pump to serve both chillers and provide a VFD on the second pump to slow it down to match the requirement for one chiller or the plate and frame heat exchange. Using one pump to serve two chillers causes the chillers to operate at a slightly higher head pressure than they would with design flows and, as a result, their kW/ton is slightly reduced. However, the 25 hp (20kW) savings achieved by eliminating the operation of one condenser pump for a significant portion of the year (approximately 3,000 hours) more than offset the increased chiller energy burden. The dry environment, which allows the cooling towers to deliver condenser water at below standard temperatures without excessive tower fan energy, also helps to offset the modest increase in chiller kW associated with the reduced condenser flow rates.

- *Option 2*. The test depicted in **Figure 12** also revealed that, by pure coincidence, trimming the impellers on the pumps to the smallest available size will provide the design flow requirement when both chillers are in operation, and provide flow in excess of the design requirement when only one chiller or the plate and frame heat exchanger are in operation with no power penalty.

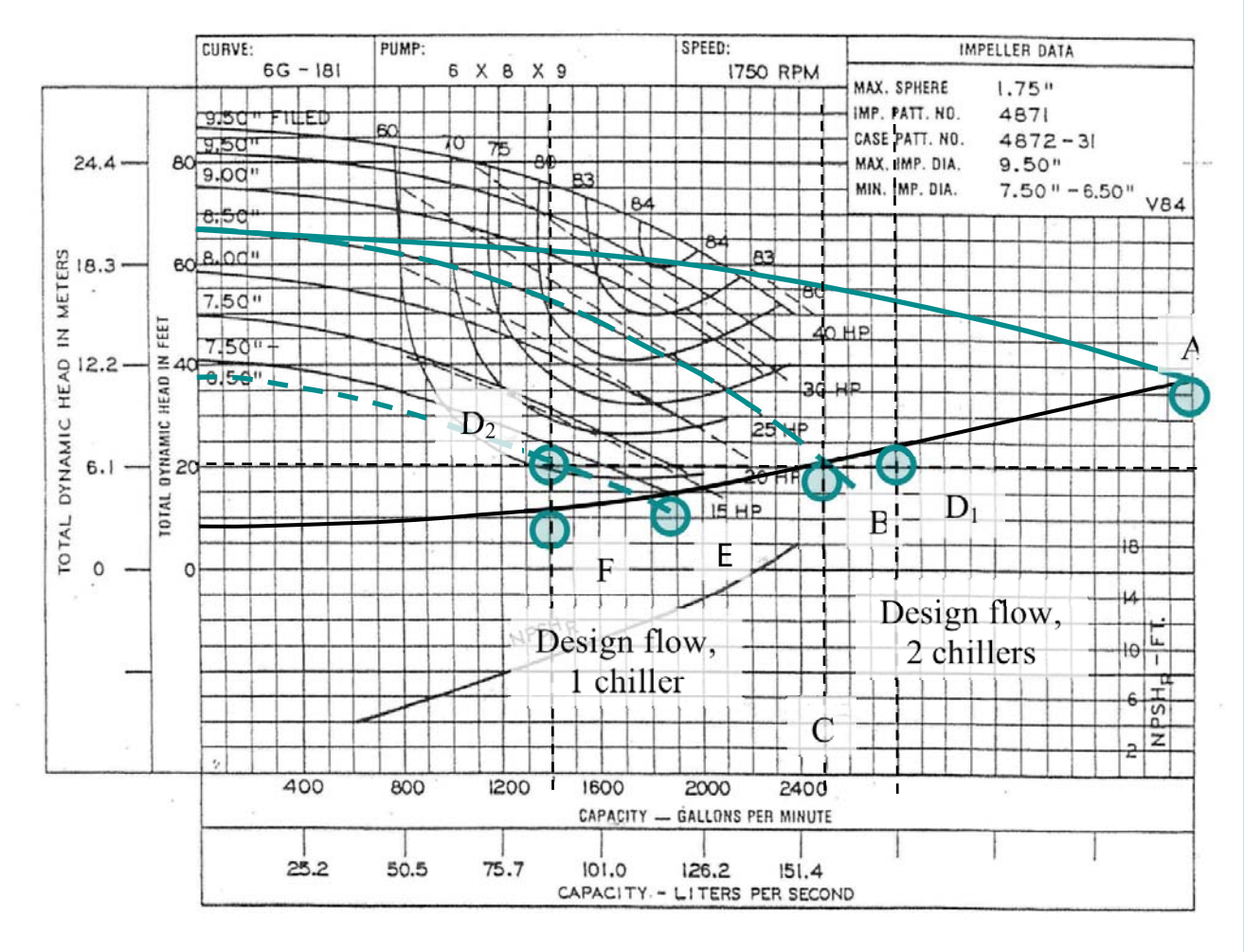
The savings and implementation costs associated with these two options are summarized in **Table 2**. Both options have attractive simple paybacks. For the owner, the VFD option is the most attractive because it delivers the most savings and the owner is able to leverage the overspeed capability of current technology VFDs. The extra five hp provided by the non-overloading motor selections used for the condenser pumps and the 1.15 service factor associated with the condenser pump motors allow one pump to deliver design flow to both chillers by operating at a higher-than-nameplate speed. Specifically, the pumps are equipped with 30-hp motors to ensure that the overloads will not trip when they run out their curve.

Figure 12: Pump Test Results for the System Illustrated in Figure 11

The pump curve and test results for the system illustrated in **Figure 10** are depicted below. The single pump performance curve is the long dashed teal colored line while the curve for the two pumps in parallel is the solid teal colored line. With both pumps running and both chiller condensers open to flow while the plate and frame heat exchanger was isolated from the system to simulate operation in the chilled water production mode (versus the wet economizer mode), two pumps were able to deliver approximately 4,000 gpm (point A; 140 percent of design) using 50 bhp total. Shutting down one pump resulted in a flow of 2,500 gpm or 1,250 gpm per chiller (point B and line C; 90 percent of design flow) using 25 bhp. While imposing a slight penalty at the chiller in the form of an increased kW per ton, the reduced flow saved 25 hp of condenser water pumping energy for a net gain.

It also would be possible to serve both chillers at the design requirement (and thus, no penalty at the compressor due to the higher head pressures associated with less than design condenser flow) by trimming the impellers of both pumps to the smallest available size (the short dashed teal colored curve). By pure coincidence, this impeller size provides the design flow with both chillers in operation (points D_1 and D_2; 2,800 gpm total, 1,400 gpm per chiller). If one pump is shut down, the remaining pump will run out its curve and deliver approximately 1,850 gpm (point E).[4] For this particular pump, the power required at E is not much different than what is required at D_2. So, there is no energy penalty associated with the extra flow from the pumps standpoint, and it probably provides a modest improvement in chiller kW per ton due to the lower head pressures associated with the higher flow rates.

The test also revealed that one pump operating at around 11 ft. w.c. could provide the design flow required by one chiller or the plate and frame heat exchanger (point F; 1,400 gpm).[4] To get to this operating point, it would be necessary to install a VFD on the pump and slow it down when it ran alone to serve one chiller.



Source: Pump Curve Courtesy of Bell & Gossett

Table 2: Optimization Options for the System Depicted in Figure 11 and Figure 12

Optimization Option	Energy Savings		Implementation Cost, $	Simple Payback, years
	kWh	$		
One pump at full speed serves two chillers, One pump at reduced speed serves one chiller	121,507	$10,632	$20,300	1.91
Trim pump impellers	108,873	$9,526	$9,340	0.98

Source: Dave Sellers, Facility Dynamics

As can be seen from **Figure 3**, when the pump runs out its curve, the impeller curve is virtually parallel with the constant 25-bhp curve. Thus, there are at least five "unused" horsepower available in any existing operating mode. Based on the available test data, operating one existing pump at 1,988 rpm would provide design flow through both chillers and require 35 bhp. If the existing 30-hp motors are allowed to run into their service factor, they can deliver 35 hp.

Purchasing a drive that allows speeds to be increased as well as decreased from the nameplate rating of the motor served would allow one pump to serve both chillers under all operating modes. It also provides design flow when necessary at a significant reduction in pump energy compared to the current operating approach with no penalty at the chiller on a design day.

Parallel Pumps and Redundancy

The generic pump curve presented in **Figure 3** illustrates the interactions that occur between a piping network's system curve and the performance curves for two parallel pumps. Of key importance for the following discussion is to recognize that if one pump in a set of parallel pumps is shut down, the residual flow per pump produced by the pump or pumps that remain on line can be significantly higher than the total flow produced by all of the pumps divided by the number of pumps in parallel. For instance, in **Figure 3**, if one of the two pumps is shut down, the flow produced by the remaining pump is approximately 78 percent of the total produced by both pumps rather than 50 percent (100 percent total flow divided by two pumps).

The relationship between total flow produced by all pumps and residual flow produced when one or more of the parallel pumps are shut down is a function of the relative shapes of the system curve and the pump performance curves, as illustrated in **Figure 10**.[5] If all three pumps are operated with both condensers open to flow, approximately 2,700 gpm will be delivered. If one pump is shut down and nothing else is changed, the flow will drop to 2,550 gpm. Ninety-four percent of the flow associated with three operating pumps is provided by 67 percent of the total pumps. If two pumps are shut down, 2,050 gpm is delivered or 76 percent of the total three-pump flow with 33 percent of the total pumps in operation. This non-linear relationship between the number of pumps and the flow produced with different combinations in operation can be exploited in some situations to provide a significant level of redundancy without a full standby pump.

The example illustrated in **Figure 13** involves two parallel pumps that serve chilled-water loads on a distribution system in a southern California coastal facility. Due to the relatively mild climate, the system spends a significant number of its operating hours at part load. An examination of the load profile and pump curves reveals the following:

- There are 5,927 hours annually at or below 600 tons (68 percent of the hours in a year). For these hours, only one pump is necessary and the second pump provides 100 percent redundancy if the first pump failed. In other words, for a significant portion of the year, the second parallel pump provides full redundancy without the need of a separate standby pump.

- One pump operating on its own can deliver approximately 1,750 gpm of flow, though at a lower than design head. In general terms, this equates to the flow required by approximately 900 tons or 75 percent of design capacity.

- It is likely that less than design head is required at less than design load since a significant portion of the pump head is associated with the flow of water through the distribution network. As the load and flow drop, the head required to move the water drops off, generally

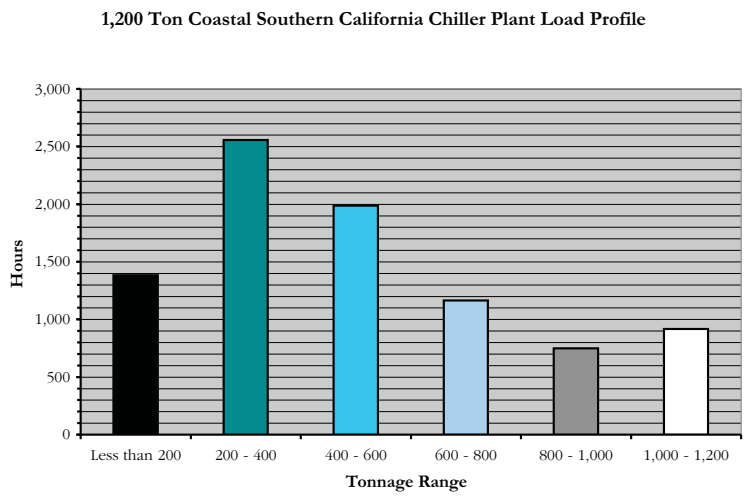
following a square relationship. In other words, if it takes four ft. w.c. of head to move a given flow rate through a given length of pipe and fittings, then if the flow is reduce by 50 percent, the head required will be reduced by 75 percent if nothing in the piping run is changed.

- There are 1,539 hours between 600 and 900 tons (18 percent of the hours in a year). Based on the preceding bullets, one can conclude that for these hours, the capacity provided by a single pump provides some measure of redundancy. Although, it is not at the 100 percent level provided by a separate stand-by pump.
- There are 1,293 hours annually between 900 and 1,200 tons (15 percent of the hours in a year). For these hours, the failure of one pump would noticeably and significantly compromise the ability of the system to meet the load.

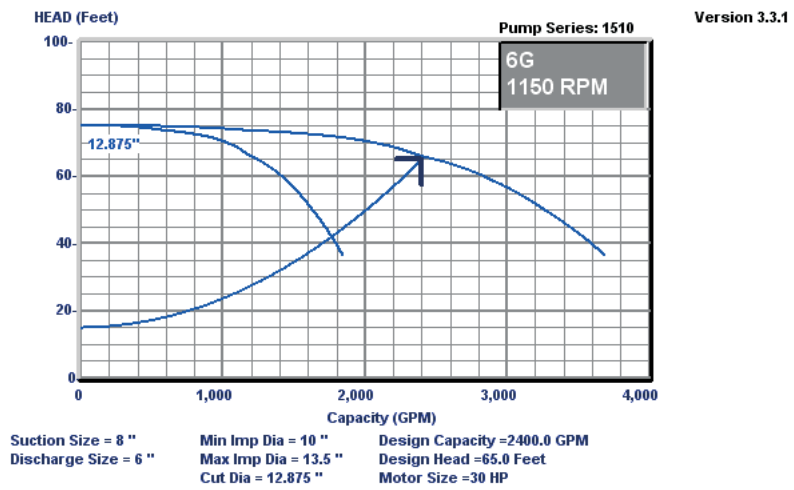
These facts provide some interesting points as a design is developed or moves into construction, especially if budgets are tight or if a value-engineering process is being invoked to control costs. In essence, for 68 percent of the hours in a year, the failure of a single chilled-water distribution pump in this hypothetical system will not compromise the ability of the system to meet the load. For another 18 percent of the hours in a year, the failure of a single chilled-water distribution pump would only incur a minor compromise on the ability of the system to meet the load.

Stated another way, a chilled-water distribution pump would need to fail concurrently with a peak load condition that exists for only 14 percent of the time before it would result in a significant compromise to the system's ability to meet the load. In mission critical facilities like those associated with laboratories, manufacturing processes, or health care, such a risk may still be deemed unacceptable and the addition of a third completely redundant pump would be essential. For a less strenuous application with a tight budget, the level of redundancy provided by two pumps may be deemed adequate, especially if eliminating the stand-by pump releases budget dollars to address other issues.

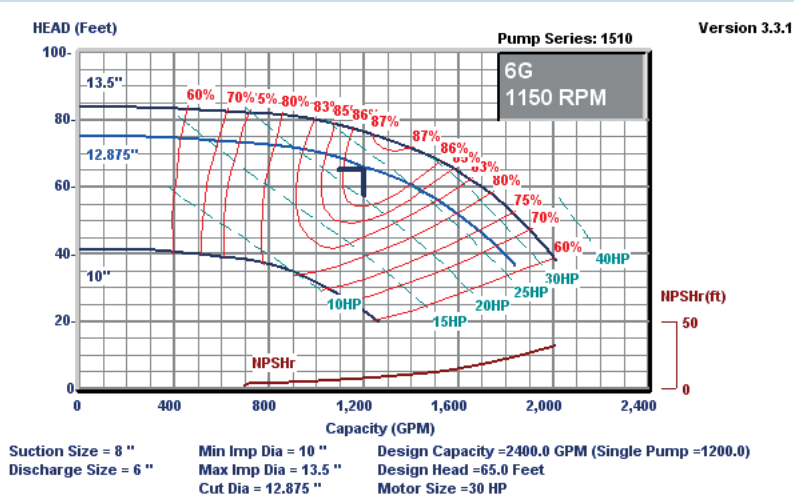
Figure 13: Redundancy Without a Redundant Pump



The diagrams to the left illustrate the cooling load profile and pump performance curves for a hypothetical building located in a coastal climate in southern California. The top diagram represents the hours the plant spends at different load conditions. It is served by a variable-flow chilled water distribution network (loads with two-way valves) that is in turn served by two 1,200 gpm pumps piped in parallel. The nature of the loads requires year-round, 24/7 operation.



The middle diagram illustrates the performance curves for the two parallel pumps serving the facility and their interaction with the chilled water distribution network's system curve. The design condition is 2,400 gpm at 65 ft. w.c. (point A). If one pump is shut down, the remaining pump can deliver approximately 1,740 gpm to the wide-open system at approximately 42 ft. w.c. This implies a certain level of redundancy without a redundant pump.



The bottom diagram illustrates the performance details of each pump, including the efficiency curves and horsepower curves. The pump efficiency is quite high at the design point. Using a VFD to vary the capacity of the pumps as the load changes and the two-way valves at the loads throttle will tend to preserve this peak efficiency. In contrast, a system designed to simply allow the pump to ride up and down its system curve will see variations in pump efficiency as the two-way valves modulate the flow in response to changes in the load.

Source: Dave Sellers, Facility Dynamics
Pump Curves Courtesy of Bell & Gossett

It is important to remember that the cost of providing a redundant pump includes more than the pump price. The cost of the piping connection, pump trim, and electrical service can equal or exceed the cost of the pump itself as illustrated in **Table 3**.

Figure 14 illustrates how pump physics might be exploited to provide redundancy at a lower first cost than might be associated with the addition of a third pump. The solution involves providing the capability to operate either of the parallel pumps at a speed above the one associated with achieving design conditions with both pumps available for service. Typically, this solution involves providing a larger motor, drive, and electrical service for both pumps since either pump operating at the higher speed will be performing the work normally provided by two pumps.

This approach offers a couple of potential advantages. In a new construction project, the larger electrical service requirements are incremental cost additions, because those components would be provided anyway. This solution simply requires more capacity. In some situations (both new and existing), there may be redundancy available within the capacity of the equipment that would be normally installed. For instance, if 30-hp motors are installed on two parallel pumps to deal with a 22-bhp design load, then seven horsepower is available for providing additional capacity in a single-pump operating mode. Capturing this capacity will involve programming the VFD to operate over synchronous speed, which is a capability that is standard in most drives. It also will involve making sure that the rotating parts of the pump (motor and pump bearings, seals, impellers, couplings, etc.) are capable of operating at the higher speed. Typically, this is not an issue for five to 10 percent increases above synchronous speed.

It is important to remember that for centrifugal pumps, horsepower varies as the cube of the speed. Thus, it will not take much of a speed increase to consume the seven "extra" horsepower, and it may not be possible to achieve design conditions using one pump unless a larger motor and electrical service is provided. However, taking advantage of the "extra" horsepower will further reduce the number of hours that a facility is at risk of not being able to achieve design flow, making the lack of a full standby pump more palatable.

The synchronous speed of an induction motor is based on the supply frequency and the number of poles in the motor winding and can be expressed as:

$$\omega = 2\,60\,f / n \quad (1)$$

where

ω = *pump shaft rotational speed (rev/min, rpm)*

f = *frequency (Hz, cycles/sec)*

n = *number of poles*

With a variable frequency drive, its possible to modulate the speed of the motor by changing the frequency supplied to the synchronous motor.

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Table 3: The Cost of Adding One Pump to the System Illustrated in Figure 13

The tabulation represents the cost to add a third redundant pump to the system described in **Figure 13**. The pump represents only a small fraction of the total installed cost. Given these costs, a facility owner faced with a tight budget or a design team faced with a value-engineering process may conclude that the redundancy provided in a parallel pumping arrangement by the nature of the interaction of the pump and system curves is adequate for a project that is not mission critical.

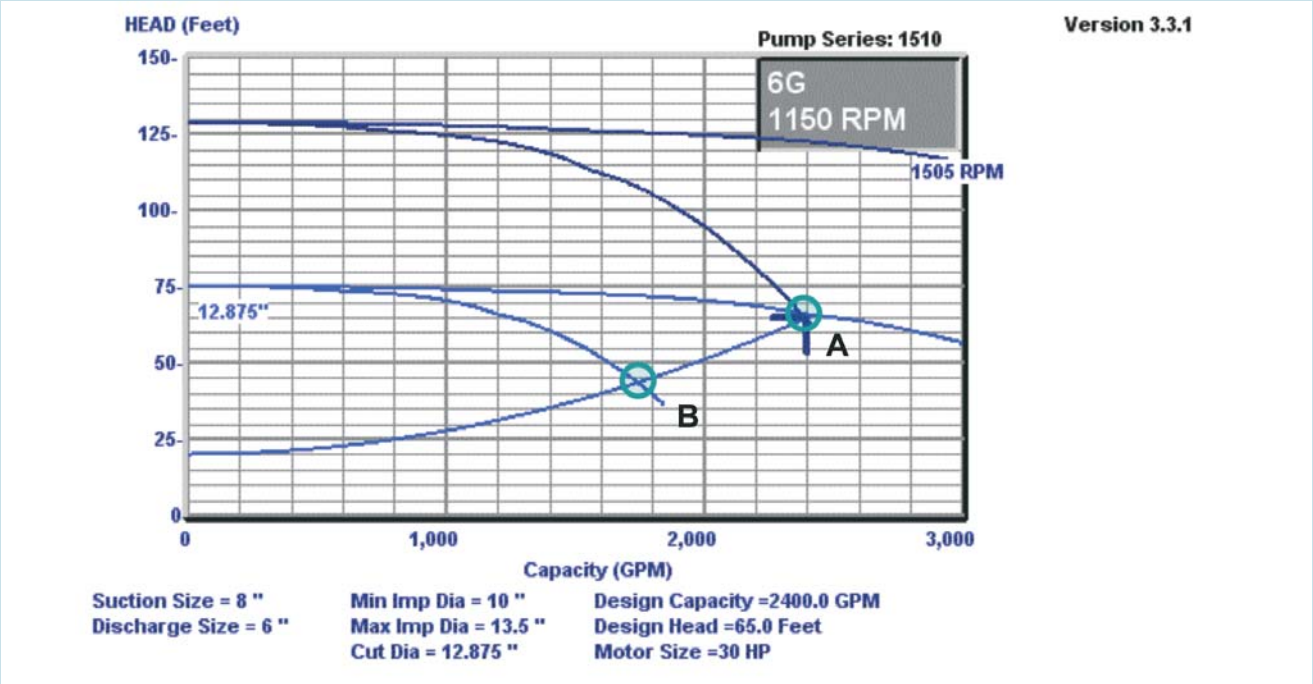
No.	Description	Quant.	Units	Material		Outside Contractor Labor			
				Unit Cost, $	Total Cost, $	Rate, $ per hour	Unit Hours	Total Hours	Total Cost, $
	Pump	1	ea	$10,600.00	$10,600	$43.75	48.0000	48.0	$2,100
	8" suction diffuser	1	ea	$1,775.00	$1,775	$43.75	9.6000	9.6	$420
	8" butterfly valve	2	ea	$320.00	$640	$43.75	5.3330	10.7	$467
	Butterfly valve operator	2	ea	With valve		With valve		0.0	$0
	8" check valve	1	ea	$950.00	$950	$43.75	5.3330	5.3	$233
	8" flex connector	2	ea	$310.00	$620	$43.75	2.0000	4.0	$175
	8" pipe flanges	12	ea	$92.50	$1,110	$43.75	3.4290	41.1	$1,800
	Flange gaskets and bolt sets	12	ea	$16.20	$194	$43.75	1.6000	19.2	$840
	8" pipe	20	ft	$48.94	$979	$43.75	0.8280	16.6	$725
	10" x 8" reducing tee	2	ea	$558.00	$1,116	$43.75	12.0000	24.0	$1,050
	8" 45 degree elbow	2	ea	$115.00	$230	$43.75	6.4000	12.8	$560
	Gauge valves (1/4" ball valves)	3	ea	$10.30	$31	$43.75	0.3330	1.0	$44
	Vent and drain (3/4" ball valves)	2	ea	$16.95	$34	$43.75	0.4000	0.8	$35
	Pressure gauge	1	ea	$167.00	$167	$43.75	0.2500	0.3	$11
	Pipe insulation	20	ft	$4.78	$96	$38.76	0.1780	3.6	$138
	Pump/suction diffuser insulation	20	sq. ft	$4.24	$85	$38.76	0.1680	3.4	$130
	Fitting insulation - flanges	12	ea	$19.12	$229	$38.76	0.7120	8.5	$331
	Fitting insulation - special fittings	3	ea	$19.12	$57	$38.76	0.7120	2.1	$83
	Fitting insulation - 8” elbow	2	ea	$14.34	$29	$38.76	0.5340	1.1	$41
	Fitting insulation - 10” tee	2	ea	$15.45	$31	$38.76	0.6000	1.2	$47
	1” conduit	50	ft	$3.66	$183	$45.53	0.1230	6.2	$280
	#8 gauge wire	200	ft	$0.41	$81	$45.53	0.0100	2.0	$91
	Pull box	2	ea	$26.00	$52	$45.53	1.4450	2.9	$132
	Variable speed drive	1	ea	$13,930.00	$13,930	$45.53	51.2820	51.3	$2,335
	Panel board switch	1	ea	$1,100.00	$1,100	$45.54	2.0000	2.0	$91
	VFD factory start-up	1	ea	$0.00	$0	$91.06	4.0000	4.0	$364
	Pad	1	ea	$150.00	$150	$37.81	1.6000	1.6	$61
	Grouting	1	ea	$75.00	$75	$37.81	1.6000	1.6	$61
	Alignment	1	ea	$25.00	$25	$91.06	2.0000	2.0	$182
	Start stop point	1	ea	$330.00	$330	With material		0.0	$0
	Proof of operation point	1	ea	$850.00	$850	With material		0.0	$0
	Speed command point	1	ea	$330.00	$330	With material		0.0	$0
	Network interface for diagnostics	1	ea	$550.00	$550	With material		0.0	$0
	Verification checks and start-up	1	lot	$0.00	$0	$125.00	2.0000	2.0	$250
					$0			0.0	$0
TOTALS					**$36,629**			**289**	**$13,075**

TOTAL - All Cost Components			**$49,704**
TAX		**5.00%**	**$1,831**
TOTAL DIRECT COST			**$51,536**
CONTINGENCIES			
	Design	**0.00%**	**$0**
	Construction	**3.00%**	**$1,546**
CONTRACTOR'S MARK-UPS			
	Overhead	**10.00%**	**$5,308**
	Profit	**5.00%**	**$2,919**
Net Mark-up with contingencies		**123.35%**	
GRAND TOTAL			**$61,309**

Source: Dave Sellers, Facility Dynamics

Figure 14: Exploiting Pump Physics to Achieve Redundancy

In some situations, it may be possible to simply exploit physics to achieve a measure of redundancy. Consider the pump curve below, which illustrates the performance of the pumps associated with **Figure 13**, if their speed is increased. The light blue lines represent the performance at 1,150 rpm as depicted in **Figure 13**; two pumps operate to deliver 2,400 gpm at 65 ft. w.c. (point A), and one pump on its own can deliver approximately 1,740 gpm at 44 ft. w.c. (point B). If the single pump is sped up to 1,505 rpm (the lower dark blue curve), it can deliver the design flow at the design head (point A). This solution is not as "clean" as simply adding a third redundant pump, and has costs and potential pitfalls associated with it. However, in some design and retro-commissioning situations, it may represent a viable approach to improving redundancy at a lower first cost than what might be associated with adding a third pump.



Source: Pump Curve Courtesy of Bell & Gossett

It is important to recognize that the approach illustrated in **Figure 14** has some limitations and potential pitfalls. Under normal operating conditions, the motor would run partially loaded. This will cost several efficiency points in terms of motor performance and will degrade the motor power factor. The efficiency of the variable-speed drive also will be reduced. Further reductions in speed associated with load reductions in the system will further degrade these efficiency and power factor compromises.

The cubic relationship between horsepower and speed is a powerful relationship with several implications. To be able to operate the pump at 1,505 rpm and meet the design load, the 30-hp motors and electrical services required by the conventional approach must be upgraded to 75 hp, doubling their cost. The added power is required because in the failure mode, one pump will do the work of two. It will do the work at

lower pump efficiency, approximately 60 percent instead of 86.5 percent, because it operates "out its curve" rather than at the peak efficiency point. Also, 75 hp is the closest available motor size to the nominal 65-bhp operating requirement associated with a single pump providing 2,400 gpm at 65 ft. w.c. However, allowing the pump to operate at speeds above 1,505 rpm will quickly overload the 75-hp motor. Thus, the programming of the speed limits in the VFD and control system becomes critical.

For conventional operating conditions, the piping drops to the pumps would most likely be sized at 8 inches, which would have a friction rate of 2.2 ft. w.c. per 100 feet of pipe at 1,200 gpm (the design flow through one pump under normal conditions) and a friction rate of 4.49 ft. w.c. per 100 feet of pipe at 1,740 gpm (the "failure mode" flow through one pump when the other pump fails at design conditions). Increasing the flow through the 8-inch lines and pump trim to 2,400 gpm will increase the friction rate to 8.36 ft. w.c. per 100 ft. of pipe, which is more than twice the typical design limit. In turn, this will increase the pump head required to move water through the piping connections, increase the potential for erosion, and magnify the impact of a dirty strainer or poor fitting arrangement. This is in contrast with the 10-inch piping connection that would be provided if the piping drops were designed in the first place for 2,400 gpm. Though, operating at these conditions as a temporary measure may be satisfactory especially if the solution is being implemented in hindsight or if full redundancy is not the goal. In a new construction scenario, the piping drops could be resized for this operating condition. However, this would be another incremental cost addition, which, when combined with the incremental electrical cost additions, makes this novel approach less attractive and the conventional stand-by pump more attractive, if full redundancy is the goal.

Net positive suction head (NPSH) requirements may become an issue. Since the pump is normally not applied to deliver the flow rate associated with design conditions, its inlet and outlet connections and the impeller eye will be undersized relative to the design flow rate. As a result, the velocities through these areas will be very high under the

design condition, increasing the potential for cavitation. In the example shown in **Figure 14**, the manufacturer does not rate the pump for use above 2,000 gpm in terms of NPSH. Thus, this would become a limitation preventing full redundancy (2,400 gpm) from being achieved without risk.

The control strategy required to implement the approach illustrated in **Figure 14** can be more complex than what is required by a conventional strategy. Complex control strategies can be more costly to implement and commission, and may have poor persistence if the operators are not properly trained. Novel approaches are subject to misinterpretation by the facility staff. As a result, thorough training and documentation will be of paramount importance.

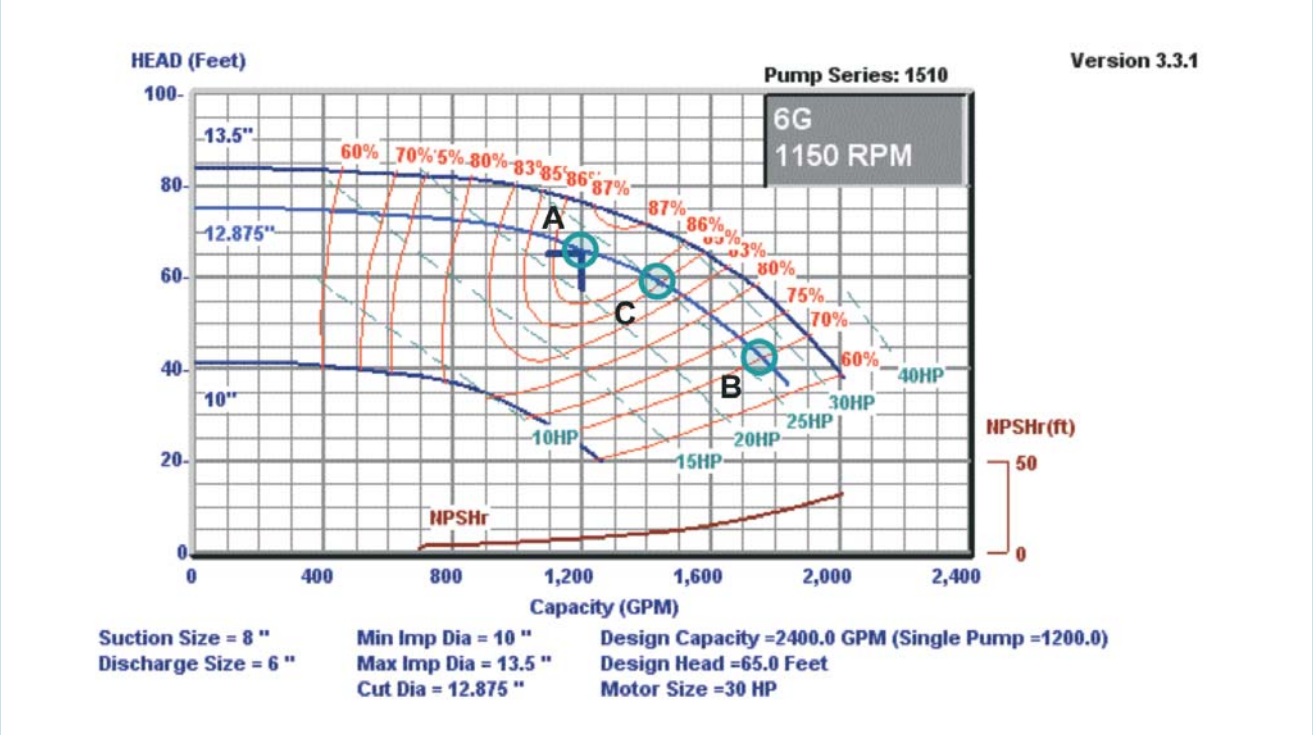
Parallel Pumps and Non-overloading Motor Selections

Figure 15 illustrates the difference between a non-overloading and an overloading motor selection. In first cost driven projects without clear requirements for non-overloading motors, it is not uncommon for a pump to be supplied with a motor that is satisfactory for the intended operating point. However, overloads occur if the head in the system is reduced from the design condition. Head reductions from design can occur for a number of reasons including a pump selection that is oversized,[6] parallel pump applications where the pumps serve loads that are incremental, or where two or more pumps normally operate together and one shuts down due to a failure or a control requirement. Frequently, overloading motor selections result in mysterious field problems.

- Operators may run multiple parallel pumps concurrently even though the load condition does not require it. Because, if they turn off one pump, then the remaining pumps run out their curves and trip off on overload.
- Operating complexity may be increased. This is because when one pump in a parallel pumping arrangement is shut down, the other pumps must be manually throttled to prevent overload trips.

Figure 15: Overloading and Non-overloading Motor Selections

For this example, the pump is the same one used in **Figure 13** and **Figure 14**. The bhp requirement at the design operating point (point A; 1,200 gpm at 65 ft. w.c.) is 23 bhp. If the pump was equipped with a 25-hp motor, it would perform satisfactorily at the intended operating point. However, if the pump that it was piped with in parallel was turned off under design conditions, the remaining on-line pump would run out its curve to the 1,740 gpm at 42 ft. w.c. operating point discussed in the **Figure 13** example (point B). The bhp requirement would equal the motor nameplate rating at approximately 1,400 gpm (point C) and exceed the motor nameplate rating at the final operating condition (point B). Eventually, the motor overload would trip. This problem could be prevented by installing a 30-hp motor on the pump. Such an installation is considered a non-overloading motor selection because there is no operating point on the pump impeller curve that can overload the motor at the design pump speed.



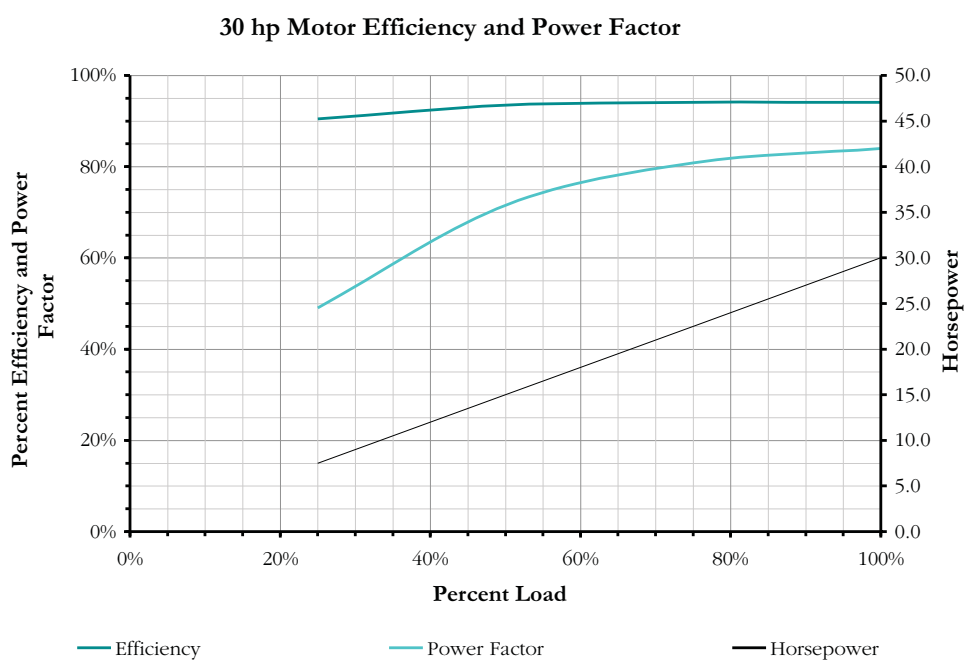
Source: Pump Curve Courtesy of Bell & Gossett

The ideal time to prevent problems of this type is at design. Eliminating this type of problem in an existing system requires that the overloading motor selection be upgraded to a non-overloading motor selection or that some method of automatic throttling or flow control be provided. The cost of implementing either retrofit solution will be significant. Larger motors may require modifications to the electrical system, pump mounting frame, and coupling system to accommodate the new motor. Automatic throttling and flow control options generally require piping and control modifications to add the necessary hardware. In addition, the hardware represents a permanent, fixed pressure drop, which will impose an operating penalty on the system in terms of pump energy.

A common concern voiced with regard to a non-overloading motor selection is the efficiency losses associated with not operating the motor at its rated capacity. While this is true for very low load conditions, motor efficiency is generally constant until approximately 50 percent load, as can be seen from **Figure 16**. In fact, for many motors, the peak efficiency point is actually at less than full load. Of greater concern is the degradation in power factor that may occur as a motor unloads. While this does not directly affect the efficiency of the motor, it can impact the efficiency of the electrical distribution system since the current drawn for the applied load increases, and the losses through the wiring and transformers increase. The utility company also may impose penalties for low power factor conditions, because they are required to have a distribution system capable of supplying the current to generate the magnetic fields in the inductive loads. Energy represented by this current is never actually consumed, but is returned to the system as the fields collapse.

Figure 16: Motor Efficiency and Power Factor as a Function of Motor Load

The curves in the graph were generated from manufacturers published data for a 30-hp premium efficiency motor. The efficiency is fairly flat until about 50 percent load while the power factor tends to quickly degrade. Both efficiency and power factor tend to improve with increasing the motor size. In other words, a 100-hp motor will have better peak efficiency and power factor than a 30-hp motor, and the efficiency and power factor reduction with load will tend to be less severe.



Source: Dave Sellers, Facility Dynamics

Parallel Pumps and Spinning Reserve

As mentioned previously, mission critical facilities often require 100 percent standby capacity for their pumping applications. Frequently, this is accomplished by installing one pump sized to carry the load with a second identical pump piped in parallel with it. The control system is arranged to start the second pump when a failure of the first pump is detected. When applying such an approach, it is important to recognize that a brief time period will occur when loss of flow is experienced. This is because the failure of the first pump must occur and be detected before the second pump can be started and accelerated to operating speed.

In some situations, the momentary loss of service will pose no problem. For instance, a short duration loss of flow in a hot water heating system serving a hospital will be virtually unnoticeable due to the thermal flywheel of the building. In other situations, a momentary loss of service can cause its own problems that are just as troublesome as having no redundancy provided. For instance, loss of cooling water flow to the crystal growers used to form the silicon ingots that are the first step in producing mono-crystalline silicon wafers for the photovoltaic and semiconductor industry can cause the jacket water to instantaneously flash to steam, resulting in an explosion.

In the latter circumstance, a design and operating strategy that keeps two or more pumps running at all times can be the difference between an acceptable and unacceptable response to the failure of any one pump. When one pump fails, the other pump simply runs out its curve and picks up a portion of the load that had previously been carried by the other pump, as illustrated in **Figure 3**. Careful selection and coordination of the pump characteristics with the system characteristics can result in a residual flow rate that is sufficient to prevent disaster and allow the system to respond and re-establish the design flow rate when one pump fails. Typical responses including starting a standby pump, or accelerating the pump that did not fail to a higher speed to pick up the load. Both can be accomplished in a matter of seconds. During this time, the residual flow produced by the pump that did not fail carries the system through the event.

When considering failure modes in mission critical facilities, it is important to understand the response of the piping system and loads subsequent to a pump failure. For example, in a midwestern hospital, a large newly installed condenser water system experienced a power outage that shutdown all of the condenser pumps. The system had been piped with non-metallic piping to prevent long-term degradation from corrosion. Unfortunately, the large diameter, non-metallic piping did not have the same strength with regard to sub-atmospheric pressures as metal piping. As a result, when the pumps were knocked off line by the power outage, and their check valves slammed shut, a combination of water hammer upstream of the valves and a vacuum that was produced by water moving away from the valves and draining to the tower basin caused multiple piping failures. The system was out of service for several days during a heat wave. Resolving the problem required the retrofit of slow-opening/slow-closing check valves, a vent system, and reinforcement or replacement of the piping in certain sections. The event itself did considerable damage to the reputation of the engineer who designed the system as well as the pipe manufacturer.

Conclusion

Optimized pumping designs and systems deliver premium efficiency and performance over their life cycle. The case studies illustrated in this brief demonstrate that a significant part of the optimization of a pump lies in how it is applied and operated in the system. The perfect pump selection made outside the context of the system that it will serve or applied improperly will not deliver the intended performance in terms of efficiency or functionality.

The current pumping stock represents a goldmine of opportunities for saving energy and improving system performance. By taking the time to understand the issues and making the necessary adjustments, practitioners dealing with pumps and pumping systems can realize immediate economic rewards. They also can improve the life cycle of the systems. The insights gained will help make every system they subsequently work on just a little better than it otherwise would have been.

For More Information

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

Energy Design Resources (2007). *Centrifugal Pump Application and Optimization*. www.energydesignresources.com

Sellers, David A. (2003). *Right-sizing Pumping Systems*. Heating, Piping, and Air Conditioning (HPAC) Engineering. March 2003. http://hpac.com

Pump Manufacturers is a general resource for various pumping systems and manufacturers. www.pump-manufacturers.com

Pump System Matters is an initiative created to assist North American pump system users gain a more competitive business advantage through strategic, broad-based energy management and pump system performance optimization. www.pumpsystemsmatter.org

Notes

1. For additional discussion of pump and system curve interactions, see Bell and Gossett (1965 – 1992) and Energy Design Resources (2007) as cited under references.

2. The pumps illustrated in **Figure 2** and **Figure 3** are located in the Pacific Energy Center, a resource and training center operated by Pacific Gas and Electric Company in San Francisco. The original design intent was for both pumps to operate together producing a combined flow of 106 gpm at 110 ft. w.c. In reality, the head requirement of the system was significantly less than originally estimated; so much less that one pump operating on its own could produce nearly 140 gpm of flow. This optimization opportunity was identified during a retro-commissioning project. As a result, the impeller on one pump was trimmed to optimize performance; while the impeller on the other pump was left "as installed" to allow the system to be used for teaching and training exercises on pump optimization.

3. While the optimization options discussed are specific to the example project, they also represent the options that are typically worth consideration for this situation in the general case.

4. The true operating point with only one chiller or the plate and frame heat exchanger on line would be slightly different than the point identified by this test, because the test was conducted with both condensers open to flow. When only one chiller (or the plate and frame heat exchanger) is in operation, the other chiller is isolated, so no flow occurs through it. This will tend to make the system curve steeper and slightly shift up the required head. A test run under the one-chiller condition will reveal the actual requirements, but the two-chiller test captured in **Figure 12** identifies the opportunity and the approximate parameters associated with the one-chiller operating mode.

5. For this system, the third pump is actually a redundant pump and is never intended to operate concurrently with the other two pumps. The discussion regarding **Figure 10** and one-, two-, and three-pump operation is intended only to illustrate the concept of system curve/pump curve interaction for multiple pumps and how the residual flow remaining after one or more pumps are shut down is not directly proportional to the total flow divided by the total number of pumps.

6. If a pump is oversized (i.e. is selected for more head at the design flow condition than is actually required by the system), when it is started, it will run out its curve and move more water than design since the anticipated resistance is not there. Of course, as it runs out its curve and moves more water, the increasing flow of water in the piping network increases the head that the pump experiences. At some point, things will come into balance with the pump moving just enough water through the system to generate a head and flow condition that is on its performance curve. This is the point where the pump performance curve and the wide-open system curve for the installed system intersect. When a balancer throttles a pump to push it back up its curve to the design condition, they are simply putting the extra head anticipated by the design, but not realized in the installation, back into the system as a loss across the balancing valve. For portions of the pump performance curve where the horsepower curve is parallel to the performance curve (as is the case in **Figure 15** for the 12.875 inch impeller curve at flows between 1,600 and 1,850 gpm), throttling reduces flow but does not save energy at the pump. If the balancing operation throttles the pump to a portion of its curve where the performance curve begins to cross the constant horsepower curves (as is the case in **Figure 15** for flow rates below about 1,400 gpm), then pump energy will be saved. The flow reductions also can have an energy impact at the load, but the nature of the impact will vary with the nature of the load. For instance, excess flow through a chiller condenser can actually improve the efficiency of the chiller by lowering its head pressure. However, extra flow through a chilled water coil that is uncontrolled (as may be the case in systems that use face and bypass dampers to control discharge temperature instead of throttling flow) can waste energy by generating lower discharge temperatures than are necessary. Lower discharge temperatures translate into unnecessary dehumidification at the cooling coil and can impose a reheat burden at the load.

Energy Design Resources provides information and design tools to architects, engineers, lighting designers, and building owners and developers. Our goal is to make it easier for designers to create energy efficient new nonresidential buildings in California. Energy Design Resources is funded by California utility customers and administered by Pacific Gas and Electric Company, Sacramento Municipal Utility District, San Diego Gas and Electric, Southern California Edison, and Southern California Gas Company, 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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