Rightsizing Pumping Systems

Lowering life-cycle costs by closely matching pump- and piping-system design performance to actual operating requirements

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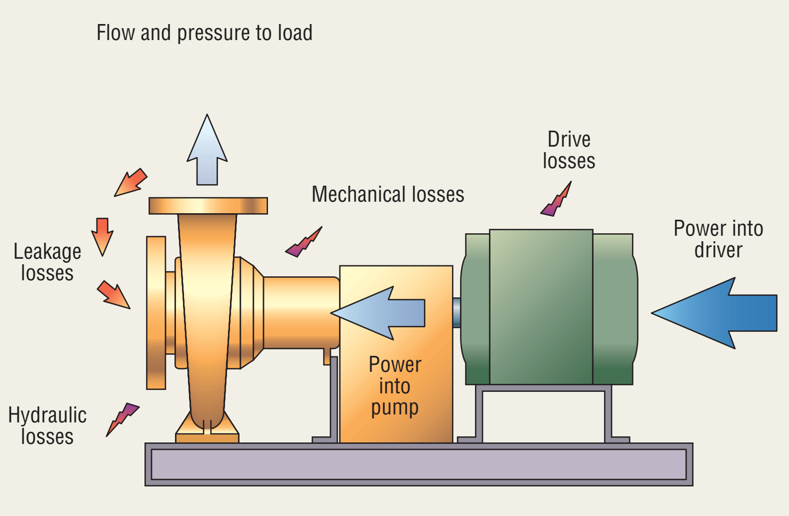
I will never forget the words of Les, a veteran contractor who took me under his wing early in my career. One day, as we were driving to a job site and I was talking endlessly about the importance of closely matching design performance to actual operating requirements, or "rightsizing," Les turned to me and said, "David, I never was sued for putting in something that was too big."

Les made an excellent point: Not big enough is not good in this business, as undersizing equipment can incur financial penalties that dwarf any first-cost savings. But while there is something to be said for "playing it safe," equipment that is oversized is prone to inefficiency and premature failure via throttling, short cycling, and other phenomena.



Source: Bell & Gossett

FIGURE 1. Pump-efficiency losses.

There is good news, however: You can have your cake and eat it, too, by following these guidelines as you develop and install a system:

- Understand the load requirements—including those beyond design—as well as the immediate and long-term needs of the owner, the building, and the HVAC process. Communicate with the owner.
- Tailor the design of the system to the actual requirements of the loads the system served in terms of peak capacity, redundancy, and turndown capability. Keep the owner "in the loop."
- Select and configure the pumps so they operate at peak efficiency most of the time. Bear in mind that many systems spend much time operating at non-design conditions.
- Include a safety factor to ensure construction-related "surprises" do not compromise the prime mover's ability to meet the design intent. Typically, I include a safety factor of 10 percent. It is your seal that will be going on the documents, however, so choose a number

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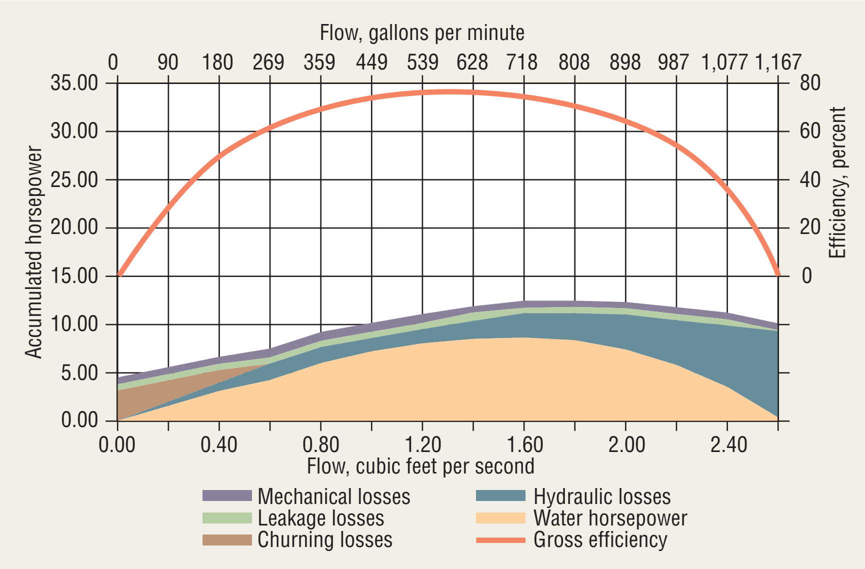


Source: "Centrifugal Pumps" by R.L. Daugherty, McGraw-Hill (1915)

FIGURE 2. Efficiency-loss variations of a typical centrifugal pump operating at a constant speed.

with which you are comfortable, based on your experience and assessment of what can go wrong between design and construction and as the system ages.

- During design, consider physical constraints and other factors associated with installation to minimize "surprises."
- Monitor and participate in the construction process to address "surprises" proactively.
- Commission and tune the equipment to the "as-installed" conditions so the system operates as close to peak efficiency as possible.
- Train the owner's operating staff to ensure it understands how to maintain optimal performance as the system ages and the loads change.

Stating your intentions clearly in contract documents is vital. Even the most skilled tradespeople need guidance. More than a statement of intent that generally is diagrammatic in nature is needed.

PUMP SELECTION

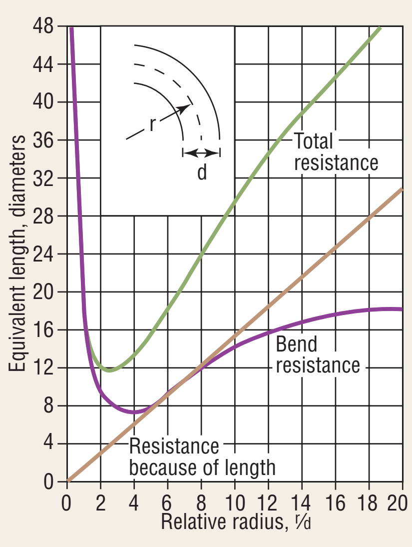
Pump efficiency is affected by several types of losses (Figure 1). While some of these losses are relatively constant, many vary with flow (Figure 2), resulting in a point where gross efficiency peaks, a point specific to the geometry and physical arrangement of individual pumps. In a perfect world, we would pick pumps that always operate at their "sweet spot." In the real world, however, we consider ourselves lucky if we find a pump from a standard product line with a peak efficiency point near our requirements. We consider ourselves even luckier if we figure out a way to keep a pump operating at or near its peak efficiency point.

Still, applications engineers have ways to optimize pump selection. One is to find the most suitable pump and write a specification so tight that it virtually eliminates all other pumps from consideration. In an industry in which competitive bidding is the norm, however, this method usually is deemed unacceptable. A more viable approach involves the specification of pump performance in terms of fundamental parameters, including flow, head, maximum brake horsepower, minimum pumping efficiency, minimum motor efficiency, minimum motor power factor, and maximum motor speed. Supplementing the specification of these parameters with requirements concerning maintenance and materials of construction creates a well-defined, level playing field.[1]

DISTRIBUTION-SYSTEM OPTIMIZATION

Pump selection is only part of the rightsizing equation. The distribution system can play an equally important role, as fitting design and application come into play.

Figure 3 depicts pressure drop through an elbow. The resistance associated with fluid moving past the wall of the pipe increases directly with the length of the turn.[2] But as the relative radius increases, the dynamic losses associated with the change in direction drop radically before rising and nearly leveling out. As a result,



Source: Crane Co. Technical Paper 410 (1957)

FIGURE 3. Elbow pressure drop vs. radius.

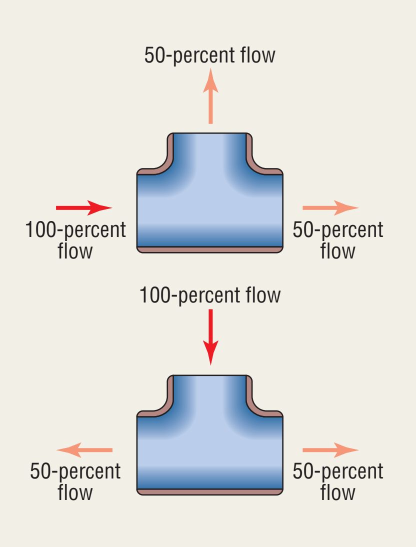


FIGURE 4. Splitting flow with a T.

total resistance is optimized when the elbow's relative radius (the radius of the elbow in terms of the pipe's diameter) is approximately 1.5 to 5. In the real world, this means a long-radius elbow with a relative radius of 1.5 will have a significantly lower pressure drop than a standard elbow, which has a relative radius of 1. Frequently, this savings in pressure drop (which translates directly to a savings in energy) can be achieved with little, if any, additional first cost.

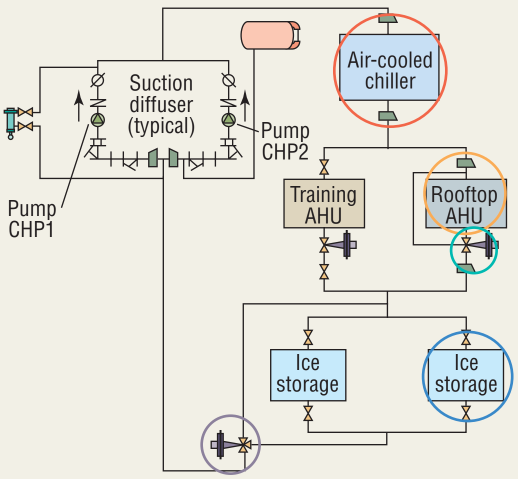
Subtle differences in the way a fitting is applied can have a surprising effect on pressure drop. For instance, both fittings in Figure 4 split flow equally in two directions. The top fitting brings in flow through the run, while the bottom fitting brings it in through the branch. The effect? The bottom fitting can have six times the pressure drop of the top one at a given flow rate.[3]

MATCHING DESIGN PERFORMANCE TO THE ACTUAL REQUIREMENTS OF A LOAD

When I first attempted to rightsize in a real-time design environment, I found myself in a quandary: To optimize my pump selection, I needed to optimize the loads and the distribution system, neither of which would be finalized until much later in the design process. Further, I was being pressured by other disciplines to provide information critical to their designs, even though that information was based on data that was not yet firm.

Eventually, I learned a technique that can be used to estimate pump head in a matter of minutes. In my experience, the estimates usually are within 10 to 15 percent of final requirements. In addition to design, the technique lends itself to troubleshooting and existing-system assessment. Figures 5 and 6 show how the approach is being used to assess energy-savings potential in an existing ice-storage system. The approach involves:

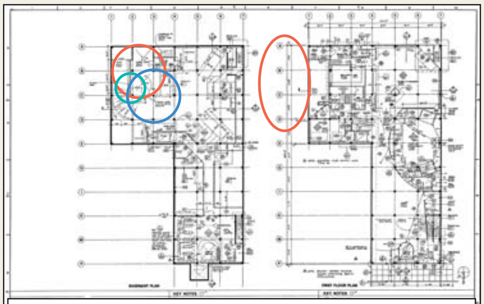
- Development of a system diagram illustrating all components and their hydraulic arrangement. This often is the first step in a design or troubleshooting process, regardless of the approach used to estimate pump head.
- Estimating head loss at design flow for each major component (tube bundles, coils, control valves, etc.). These estimates can be based on past experience, shop-drawing or catalog information, or simply a guess. Frequently, I estimate a range, rather than a single value, especially when using the technique for assessment or troubleshooting.
- Assessing losses in the piping circuit using an estimate—based on the physical arrangement of the components in the building—of the equivalent feet of



Item	Maximum ft wc	Minimum ft wc
Evaporator	10	15
Coil	5	10
Control valve	5	10
Ice tank	10	15
Control valve	10	15
Pipe network*	7	9
Total	**42**	**69**

*See Figure 6.

FIGURE 5. Estimating pump-head requirements from preliminary design information.



Dimensions on architectural drawings show about 36 lineal feet of run in tank area.

Piping location	Minimum equivalent feet	Maximum equivalent feet
Ice-tank area	54	72
Riser	54	90
AHU area	54	72
Total	**162**	**214**
Loss (ft wc)	**7**	**9**
Assumed friction rate: 4 ft wc per 100 ft of pipe (top end of normal design window).		

FIGURE 6. Estimating equivalent feet of pipe from preliminary design information.

straight piping that will be required.[4] The American Society of Heating, Refrigerating and Air-Conditioning Engineers and experience indicate that for most commercial projects, equivalent feet of straight piping will be one-and-a-half to two times lineal feet of piping. I used this rule of thumb for the existing ice-storage system (Figure 6). Specifically, the equivalent feet for 36 lineal feet of pipe in the ice-tank area was 54 (1.5 x 36) at the low end and 72 (2 x 36) at the high end. Once equivalent feet of straight pipe is calculated, pressure drop can be determined by using either friction rate associated with design flow in anticipated line size or a typical design limit, such as 4 ft wc per 100 ft of pipe, which would apply regardless of line size.

- Balancing assessments of requirements with the system's anticipated long-term needs. For instance, a properly maintained closed system will corrode less than an open one, in which air is entrained continuously (photos A and B). If a system is large and could be in place for years, having a little leeway in terms of capacity and motor size could be good. There is no sense in being penny-wise and dollar-foolish.

I concluded that the pump head required for the ice-storage system as installed should be 42 to 69 ft wc (the bottom line in Figure 5). For assessment purposes, this was good enough to compare to the pump nameplate and building design data and make a decision regarding energy-savings potential. In a design scenario, the pump head used to make a first pass at pump selection and motor size could be the largest value, the smallest value, or the average value, depending on how confident you are and the repercussions of being wrong. Or, the estimates could be refined to narrow the range.

THE REPERCUSSIONS OF MISSING THE TARGET

For the ice-storage system, the design intent as reflected in the pump-nameplate data and contract documents was 110 ft wc. Specifically, the system was designed so that both pumps ran, each providing 53 gpm (106 gpm total) at 110 ft wc. Thus, my assessment indicated significant energy-savings potential,[5] in addition to a considerable difference of opinion regarding the system's pumping-head requirements. I had the benefit

PHOTOS A and B. Both of these pipes are the same age and served the same chiller, seeing nearly identical operating hours. The one on the left is the condenser-water line (an open system), while the one on the right is the chilled-water line (a closed system). Both systems were provided with a water-treatment program by a competent contractor, with oversight by a knowledgeable owner.

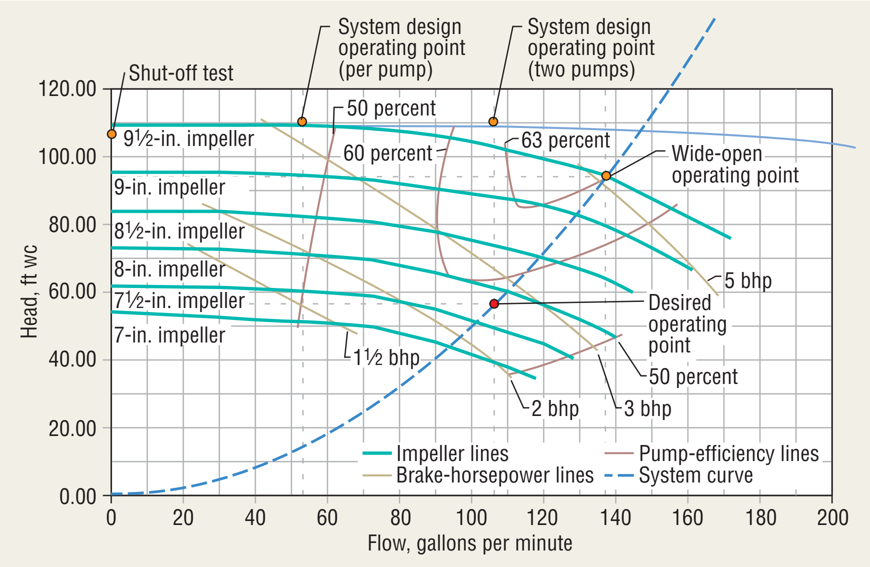


FIGURE 7. Pump-test results for the ice-storage system.

of hindsight in that I was assessing an installed system after the fact, while the designer had the benefit of more time and information (maybe). To find out who was right, we ran a pump test, the results of which are presented in Figure 7.

A detailed discussion of the test results is beyond the scope of this article. The key points are:

- One pump, operating wide open (unthrottled), provided 130 percent of the required capacity, with consumption of a little more than 5 bhp.
- The design flow could be provided by a pump with 57 to 58 ft wc of head (approximately the midpoint of my projected range).
- Throttling allowed either pump to deliver the design flow while operating near its peak-efficiency point, with consumption of a little less than 5 bhp.
- Trimming impellers allowed either pump to meet the design flow requirement with consumption of a little more than 2.5 bhp—a significant energy savings despite the fact the operating point was off peak efficiency.
- A variable-frequency drive (VFD) would have allowed pump speed to be

reduced to provide the required capacity at a lower horsepower and would have tended to preserve the operating point at or near the pumps' peak efficiency point. However, it would have added complexity and cost and introduced an efficiency loss of its own, probably in the range of 8 to 12 percent, given the speed reduction that would have been required.

CONCLUSION

The pump test revealed the ice-storage system could be made more efficient (and more redundant) via an impeller trim—a simple, cost-effective technique. However, it also revealed that the existing pump was the wrong pump. Given the benefit of hindsight, we can see that a pump selected for peak efficiency while moving 106 gpm at 58 ft wc would have been a better solution from the start, saving first cost and optimizing operating cost for the life of the system. Fortunately, the system's relatively small size minimized the first-cost penalty associated with the mismatch, and the retrocommissioning process mitigated the long-term impacts.

Sadly, retrocommissioning indicates there are many systems with a significant mismatch between installed pumping capacity and actual operating requirements. On a recent project, approximately 50 hp could have been saved had each of four pumps been rightsized from the start. While significant savings still could be captured by modifying the existing pumps, pumps with characteristics more closely matching the actual operating requirements probably would be 8- to 10-percent more efficient.[6] In fairness, two of the pumps are evaporator pumps serving a closed system, while the other two are condenser pumps serving an open system. In the case of the condenser pumps, the excess installed head could be a benefit as corrosion and time take their toll; in the near term, however, tuning them to the existing requirements could have significant energy implications, with a short payback justifying the effort. Training and documentation would help the operating staff make good decisions if the effects of aging result in an increase in pumping-head requirements. For the evaporator pumps, the excess head may represent more of a lost opportunity to save energy and first cost. Table 1 summarizes the ripple effects of the mismatch in terms of first cost and annual operating cost for one of the pumps.

Hindsight always is 20/20. It is relatively easy for me, as a commissioning provider, to say where a pump should have been selected, given the benefit of an installed system and no pressure from

	Original	Revised	Savings
First-cost savings			
Motor size	125	75	50
Nominal amperage	156	96	60
Motor efficiency	90 percent	90 percent	N/A
Nominal kw	104	62	41
Motor cost	$4,025	$3,050	$975
Wiring cost	$5,050	$4,600	$450
Total	**$9,075**	**$7,650**	**$1,425**
Annual operating-cost savings			
Hours of operation	3,000	3,000	N/A
Annual kwh	310,833	186,500	124,333
Electric rate, $ per kwh	$0.1000	$0.1000	$0.1000
Annual operating cost	$31,083	$18,650	$12,433

TABLE 1. Opportunity lost when a rightsizing target was missed.

a compressed design timeline and tight design budget. However, concern for our children compels me to say we must do better. My experience with the assessment technique described in this article tells me we can do better, while my experience optimizing machinery and systems tells me we can have a good time doing it.

NOTES

1) See "Specifying Pumps" in the November 2003 issue of *HPAC Engineering* for more on developing this type of specification.

2) The larger the radius of the elbow, the larger the circumference of the circle it represents and, thus, the longer the flow path.

3) To learn more about how fitting arrangement can impact system efficiency and first cost, go to *www.energydesignresources.com/resource/25*.

4) This technique converts fittings to the equivalent length of straight pipe that would generate the same pressure drop. This number is added to the actual length of straight pipe to allow the entire piping circuit's head loss to be assessed.

5) The throttled valve on the pump discharge confirmed this, revealing that the balancer had to add pressure drop to the system to force the pump up its curve to the design operating point.

6) At their design operating point, the existing pumps have a respectable efficiency of 81 to 82 percent. An impeller trim would allow the design flow to be delivered with 50 bhp, instead of 108 bhp, but at an efficiency of 70 to 72 percent.

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