ECONOMICALLY MATCHING CIRCULATION WATER FLOW TO COOLING TOWER LOAD

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ABSTRACT

Condenser cooling water flow requirements for large turbine-generator sets are controlled both by the thermal input range of the turbine and the variety of climatic conditions affecting the cooling tower. The resulting requirements on the circulating water pumps are considerably exaggerated relative to normal turbine load variations. One Southwestern electric utility is using three partial capacity pumps driven with two speed motors as a practical solution to this pumping requirement. This paper describes the specific idiosyncrasies of this pumping requirement and the optimization of such cooling systems.

INTRODUCTION

In traditional central electric generating stations, the latent heat remaining in the steam as it is exhausted from the turbine is given up in the condenser, the primary heat sink of these Rankine cycle systems. Circulating water flowing through the tubes of the steam condenser absorbs this waste energy as sensible heat. The sensible heat is then displaced by:

• actively cooling the water before it is returned to the condenser as with a cooling tower.

• using water from a large source such as a cooling lake; that allows the heat sink to cool relatively independent of station load.

• completely replacing the cooling water as is done at an open cycle, river cooled station.

Traditionally, cooling towers, particularly the mechanical draft type, have been popular with central station designers, because they allow a station to be located away from large bodies of water or rivers. This closed type of circulating water system is shown in Figure 1. One limit to which a closed system can be optimized is dependent on the flexibility of the pumping system selected. This is the specific subject of discussion herein.

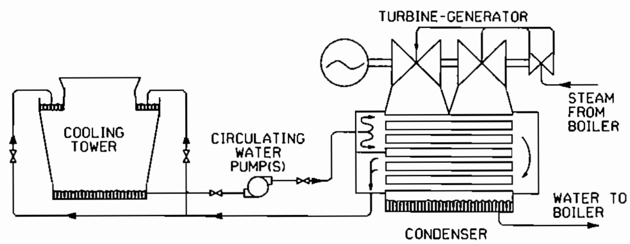

Figure 1. Closed Cooling Water System Schematic for a Typical Generating Unit.

Recently, thermal pollution regulations have necessitated the inclusion of cooling towers in the other two approaches to heat dissipation. The lessons learned from optimizing closed cooling systems can be applied to these "once-through" cooling systems.

COOLING SYSTEM CHARACTERISTICS

As is required by most sucessful design procedures, an understanding of the entire process is necessary to appreciate the specific constraints on the pumping system. In this case, the process is the cooling system including the characteristics of the heat source and heat sink. The essence of economically optimizing an efficient circulating water system is in understanding and matching the diverse operating characteristics of the turbine input and the atmospheric heat sink, and in controlling the cost of circulating the cooling water.

Turbine Input

One parameter of steam turbine performance is the turbine exhaust pressure or back pressure imposed by the condenser. This back pressure is optimized during the turbine selection process. Typically, it is desirable to maintain a constant back pressure during normal operation. Therefore, the condensing temperature remains constant, and heat rejection from the turbine is relatively proportional to turbine load. This condensing temperature becomes the upper limit of the thermal gradient or head available to drive the transfer of heat. Large central station turbines usually are designed with back pressures ranging from 2.0 to 3.5 in of mercury.

Cooling Tower Design

A cooling tower, shown in Figure 2, is a direct contact, water to air heat exchanger that is limited primarily by relative humidity. Warm water entering the cooling tower cascades over the fill that breaks up the water flow, maximizing the surface area of the water relative to the mass of the water. Air drawn through the fill by the fan(s) cools the water by evaporating a small portion of the water.

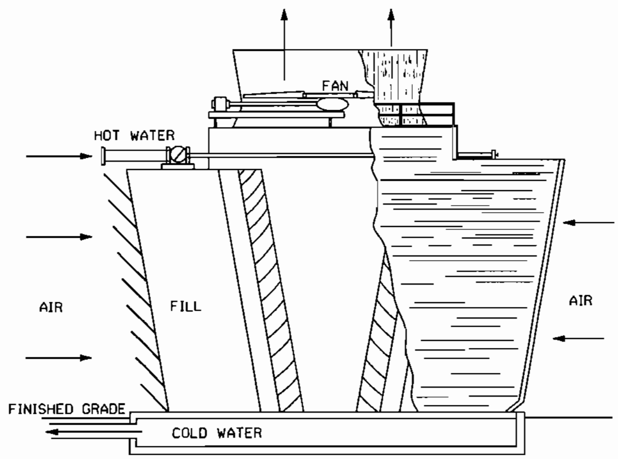

Figure 2. Section of a Typical Crossflow Cooling Tower.

Since the water cooling process is primarily dependant on evaporation, the ambient wet bulb temperature becomes a measure of the cooling ability of the air. Out of necessity, the tower is designed to transfer the maximum turbine heat rejection with the worst potential wet bulb temperature. The maximum wet bulb temperature becomes the "design" lower end of the thermal gradient available to drive the heat transfer process. As a result, the tower is considerably oversized for average weather and full turbine load operation. This margin is even more exaggerated at average turbine load, particularly in arid regions where the maximum wet bulb temperature is well above the norm. Many generating units located in the Southwest and equipped with cooling towers are connected to electrical systems that experience summer air conditioning and/or farm irrigation peak loads. Winter operation of these units is frequently at lower loads and low wet bulb temperatures.

The maximum load capability of the tower can become a liability during winter weather. In subfreezing weather, insufficient thermal load for a given water flow permits ice to form on the exterior of the cooling tower, potentially overloading the tower structure. In the case of concrete structures, the freezing causes the water soaked concrete to spall.

System Optimization

Optimization of the basic cooling system is a two-step process. The first step weighs the cost of turbine efficiency against the increased cost of the turbine and the estimated cost of the cooling system. More efficient, lower back pressure operation generally requires a larger low pressure turbine. The lower condensing temperature becomes less available thermal head for a given location, and requires a more efficient, more expensive condenser, cooling tower and/or circulating water system.

Once the turbine design is set, the optimization process concentrates on a balance between the first cost of the condenser, the first cost of the cooling tower, and the cost of the horsepower to pump cooling water. This balance can be broken down into three interactive relationships that depend on how the thermal head is subdivided, as shown in Figure 3.

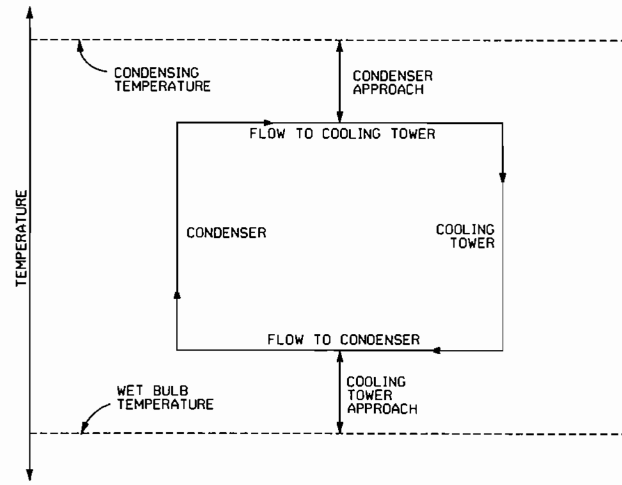

Figure 3. Closed Cooling System Thermal Gradient.

- The relationship between condenser approach temperature and the effective surface area of the condenser tubing. Condenser approach temperature is the difference between condensing temperature and the temperature of the hot water exiting the condenser.
- The relationship between cooling tower approach temperature and the tower size. The tower approach temperature is the difference between wet bulb temperature and the temperature of the cold water leaving the tower.
- The relationship of circulating water flow to the temperature difference between the hot and cold circulating water, known as cooling tower range or condenser rise.

The last of these relationships is dependent on the flexibility of the circulating water pumping system. With a constant flow system, horsepower consumption is high and unaffected by turbine load. It is more economical to obtain capacity and reduce flow by spending extra capital to decrease the design approach temperatures of the cooling tower and/or condenser. Conversely, if circulating water flow can be modulated to maintain load on these system heat exchangers, then the average pumping horsepower can be reduced. Taken to an extreme for a given load and location, setting the average power consumption of a modulating flow system equal to the power consumption of a constant flow system significantly reduces the size of the heat exchangers at the expense of buying larger pumps.

A survey of recently completed generating units was conducted to obtain a perspective of actual industry trends. Cooling tower approaches from 9 to 14 F degrees and ranges from 12°F to 25°F degrees were noted. The vast majority of units surveyed use two-half capacity pumps.

System Head Curve

The flow of a circulating water system, roughly from 35 to 75 times the turbine exhaust flow, can approach a million gallons per minute for the largest central stations. Typically, total system heads at full flow range from 65 to 110 feet. Static head, primarily imposed by the height of the cooling tower, accounts for roughly half of the total system head. The pressure drop across the condenser tubes is the major contributor to dynamic head. A system head curve for a nominal 550 MW electric generating unit is shown in Figure 4.

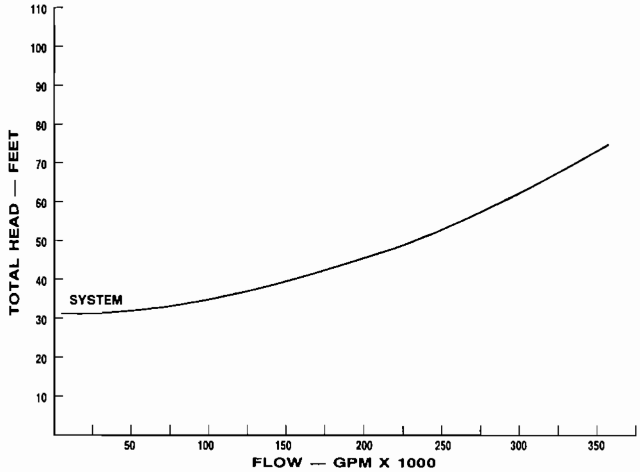

Figure 4. System Head Curve for a Closed Cooling System.

THE PUMPING SYSTEM

The high flow verses low and constant head nature of the system curve complicates the task of modulating flow. The difficulty of modulation grows with unit size because the design flow of the system is proportional to unit size while the head is held constant by component characteristics.

Traditional Systems

Most early systems were effectively single pump systems even though a second or alternate pump may have existed for reliability purposes. Typically, these pumps were motor driven. Since early generating units also tended to be smaller, a single pump did not necessarily infer a high specific speed pump. Turndown in these systems generally could be accomplished by manually isolating a portion of either the condenser or the cooling tower, but typically was done only for maintenance purposes. Partial isolation of the tower was also a means of concentrating thermal load to prevent ice formation. Because unit sizes were relatively small and energy prices were relatively low, encumbering these pumping systems with turndown capability was not warranted.

The use of two half-capacity pumps became vogue as the practicality of building very large, full capacity pumps waned; and as the economic incentives to reduce capital costs and save energy consumption increased. The ability of one pump to carry approximately 60 percent of design flow made it practical to omit having a third, spare pump. As indicated in the above noted survey, this arrangement is the current standard of the industry.

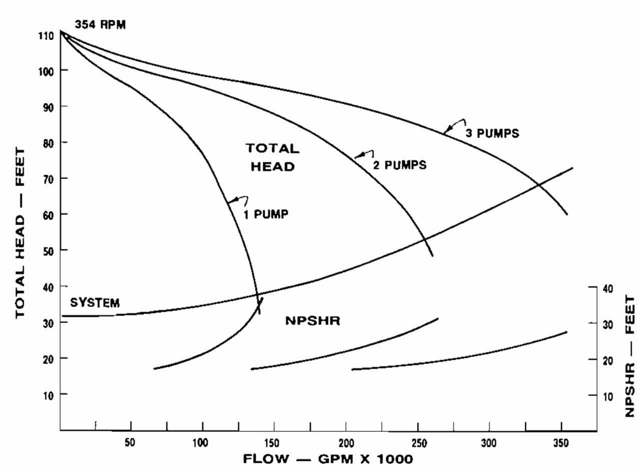

Figure 5. System Operating Curves for a Three Single-Speed Pumping System as Applied to a 550 MW Generating Unit.

Additional attempts to reduce the escalating cost of energy have given cause to extrapolate from the sucessful two half-capacity schemes to three third-capacity and even some four fourth-capacity pump arrangements. In addition to having greater turndown, each pump has a lower specific speed, giving rise to at least a theoretical efficiency advantage [1]. These schemes work, but are marginal because:

- During one pump operation, the single pump flow increases considerably past the pump design flow even though the system head curve has considerable static contribution. In the case of a three pump arrangement, single pump flow is really midway between one-third and one-half system design flow as shown in Figure 5.
- The required suction head increases dramatically as the actual pump flow increases beyond design flow. This characteristic is also shown in Figure 5.

Variable Speed Operation

Generically, the above described limitations to multi-pump schemes can be minimized by adding variable speed operation as a supplemental means of controlling flow. Desired flow reductions are achieved by reducing speed as well as the number of pumps operating. Lower speed operation alleviates the exaggerated NPSH requirements that occur during single pump operation.

Variable speed schemes, while common on other pumping systems within a generating station, have been considered too complex for circulating water systems. The flat nature of the system head curve is perceived as limiting the range of speed variation. However, the specific characteristics of large circulating water systems help to mitigate these reservations. Pumps used in circulating water service, whether horizontal or vertical, are low speed pumps, typically in the 250 to 400 cpm range. There are six possible synchronous speeds in this range, making the use of either two motors or two-speed motors practical. This relatively simple approach avoids the complexity of other speed variation techniques. Depending on the shape of the pump curve, a change of either one or two synchronous steps can significantly change flow without leaving the usable range of the pump.

Specific Case

A system using three pumps equipped with two-speed motors can achieve four relatively equal flow steps. The curves for such a system as installed on a 550 MW generating

Table 1. Characteristics of a Three Pump Cooling System Equipped with Either Single Speed or Two-Speed Motors as Applied to a 550 MW Generating Unit.

Pumps On	Speed, CPM	System Flow, GPM	System Flow, %	Pump Flow, GPM	Total Head, Ft	Pump Eff, %	Pump Shaft HP
THREE SINGLE-SPEED PUMPS							
3	354	334,500	100.00	111,500	69	89.0	2,161
2	354	254,000	75.93	127,000	53	80.3	2,109
1	354	138,000	41.26	138,000	38	52.0	2,541
THREE-TWO SPEED PUMPS							
3	354	334,500	100.00	111,500	69	89.0	2,161
3	295	254,000	75.93	84,667	53	91.0	1,241
2	295	194,000	58.00	97,000	44	88.9	1,212
1	295	107,000	31.99	107,000	36	82.5	1,160

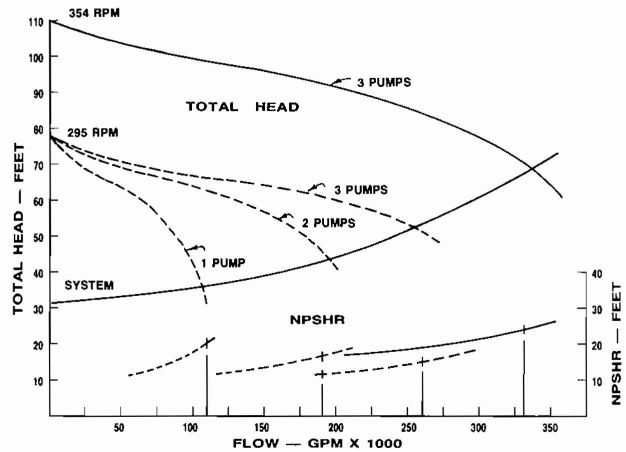

Figure 6. System Operating Curves for a Three Two-Speed Pumping System as Applied to a 550 MW Generating Unit.

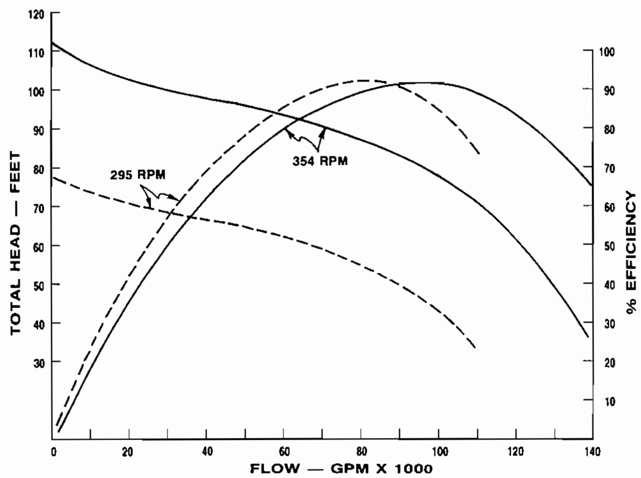

Figure 7. Curve for the Pump Used in the Systems Shown in Figures 5 and 6.

unit are shown on Figures 6 and 7. The different partial load characteristics of the three pump two-speed system are compared to the same three pumps equipped with only single speed drives (Table 1). The accuracy of the system evaluation depends on the system designer's ability to reasonably estimate the time the pumping system will operate at each flow step. The operating times for each of these systems based on actual load factor data from a base-loaded, coal-fueled generating station and average monthly wet bulb temperatures from the Southwestern Plains are shown in Tables 2 and 3.

The operating data from Tables 2 and 3 are used in computing the energy savings shown in Table 4. The energy cost is based only on the cost of input energy to the generating station. In this case, the energy savings would offset the capital investment in a little over two years.

A reduced potential of cooling tower freezing is a side benefit of the two-speed system. The lower flow capability of this pumping system, when combined with a cooling tower designed in totally isolable modules, allows smaller sections of the tower to operate under higher thermal load.

DISCUSSION

In the example, the two-speed pumping system is designed to operate most efficiently at the lower speed until the full compliment of pumps is needed to operate at the higher speed to achieve full flow. As can be seen on Figures 5 and 6, partial flow operation moves to the right on the pump curve. Selecting pumps that set full capacity operation slightly to the left of the best efficiency point can move partial load

Table 2. Operating Times of a Single Speed Three Pump Cooling System as Applied to a 550 MW Generating Unit.

PUMPS ON	3	2	1
SPEED	354	354	354
MONTH	HOURS PER YEAR		
JAN	0	0	681
FEB	0	0	682
MAR	0	0	682
APR	0	185	496
MAY	0	427	254
JUN	0	546	136
JUL	2	593	88
AUG	0	591	88
SEP	0	470	211
OCT	0	297	385
NOV	0	0	682
DEC	0	0	681
TOTAL	2	3109	5066

Table 3. Operating Times of a Two-Speed Three Pump Cooling System as Applied to a 550 MW Generating Unit.

PUMPS ON	3	3	2	1
SPEED	354	295	295	295
MONTH	HOURS PER YEAR			
JAN	0	0	1	681
FEB	0	0	185	495
MAR	0	0	376	306
APR	0	0	470	211
MAY	0	0	546	136
JUN	0	376	306	0
JUL	2	427	254	0
AUG	0	427	254	0
SEP	0	185	496	0
OCT	0	0	470	211
NOV	0	0	376	306
DEC	0	0	1	681
TOTAL	2	1414	3734	3026

Table 4. Economic Comparison of a Three Pump Cooling System Equipped with Either Single Speed or Two-Speed Motors as Applied to a 550 MW Generating Unit.

ANNUAL HORSEPOWER SAVINGS *

Pumps On	Speed, CPM	System Flow, GPM	Total Head, Ft	Eff, %	Total Shaft HP	Hours per Year	Capacity Factor %	Energy Cost, $
				THREE SINGLE-SPEED PUMPS				
3	354	334,500	69	89.0	6,483	2	0.02	201
2	354	254,000	53	80.3	4,218	3,109	23.09	203,537
1	354	138,000	38	52.0	2,541	5,065	22.66	199,767
TOTAL						8,176	45.77	403,505
				THREE TWO-SPEED PUMPS				
3	354	334,500	69	89.0	6,483	2	0.02	201
3	295	254,000	53	91.0	3,722	1,414	9.27	81,686
2	295	194,000	44	88.9	2,425	3,734	15.94	140,540
1	295	107,000	36	82.5	1,160	3,026	6.18	54,502
TOTAL						8,176	31.41	276,929
						Annual Savings		126,576

CAPITAL COST INCREASE

				Single-Speed	Two-Speed
Motor HP	2,250	$/HP		30	70
Installed HP	6,750	$ Installed		202,500	472,500
			Capital Increase		270,000

* Using 20.8 mils/kwh bus bar cost

operation to a more advantageous position on the curve. This advantage can be assured if the final selection of the pump impeller diameter is made after the actual system components with their respective flow resistances are committed. Close coordination between the station and pump designers make this iterative step practical.

The inability of a two-speed system to infinitely adjust flow does not present a particular problem. Process requirements for infinite speed variation are mitigated by three factors.

- The number of cooling tower fans in operation is adjustable.
- Small decreases in flow are offset by proportionately small increases in condensing temperature that increase the thermal head available to drive the condenser and cooling tower. In the subject range, a one degree Fahrenheit condensing temperature change corresponds to a change of less than one tenth of an inch of mercury back pressure.
- Flow interruptions due to speed changes have little effect on turbine back pressure because of the volume of the circulating water system and the ability to change speeds quickly on one pump at a time.

Additionally, the actual operating points are on a relatively efficient section of the pump efficiency curve (Figure 7). An infinitely variable system would only be able to locally optimize flow along the curve, marginally improving efficiency. In both cases, the requisite number of pumps would have to be operating to maintain best efficiency.

The use of two-speed motors is not new and the techniques required to reliably switch speeds are well developed. The idea of driving pumps at two or more synchronous speeds to refine flow control was presented almost thirty years ago for public works applications [2]. The efficiency penalty paid for selecting a two-speed motor instead of a single speed motor is anywhere from zero to two percent or three percent, depending on the motor design. A two-speed motor can be applied to either horizontal or vertical pumps.

Two single speed motors do not suffer a driver efficiency penalty, but they are marginally more expensive and require more room than a single two-speed motor and are relatively awkward in vertical pump applications.

In the course of designing a two-speed system, the speed switching sequence should be carefully analyzed. Reverse pump rotation can overload the motor when it is renergized.

Alternate Pump Drivers

A number of other variable speed driving techniques are available, and most offer infinitely adjustable speed for the reasons stated above, only small pumping and heat transfer efficiency improvements are achievable. Conversely, other characteristics of these systems host a variety of liabilities.

- Steam turbines, the most traditional multi-speed drivers in electric generating stations, are hampered by the popular practice of locating the circulating pumps adjacent to the cooling tower basin to increase available NPSH. Drive steam would have to be routed significant distances and would not necessarily be available during startup conditions. Steam turbines of the size required by these pumps and designed for modern central station boiler pressures and temperatures are prohibitively expensive. The typical turbine sold in this power range is designed for lower pressures and temperatures. Such a turbine would require pressure breakdown and desuperheating stations, and is 15 percent to 20 percent less efficient than large generator drive turbines. No turbines are currently marketed to drive horizontal, much less vertical, circulating pumps without gearing. After including the requisite accessories, a turbine drive system would be twice as expensive as a two-speed motor system, and not as efficient.
- Torque transmission speed adjusting devices such as hydrodynamic and eddy current speed varying couplings are less costly than turbines, but have an efficiency approximately proportional to the ratio of output speed to input speed. A pump operating with a nominal speed range from 360 to 300 cpm would suffer efficiency losses from three percent to 17 percent through the coupling alone. The low speed ranges of circulating water pumps correspond to relatively high torques for the required horsepower, and in turn, relatively high capital investments for the couplings. Gearboxes can be used to reduce the capital costs of low speed motors and couplings, but at the expense of the gearbox and attendant gear losses. Both of these drive types can be applied to both horizontal and vertical pumps. For vertical applications, "reed frequency" resonance should be checked to avoid future vibration problems. These systems cost roughly the same as a two-speed motor system, and would not be as efficient.
- Adjustable frequency systems are relatively new in the subject power ranges. This technology uses a relatively standard motor. The motor is coupled to the pump in the same fashion as a single speed motor drive. The speed of the pump is then controlled by varing the frequency of the motor input power, using solid state switching. Adjustable frequency systems drop the basic motor efficiency by one to two percent, but are comparable in cost to turbine drive systems.

Other Circulating Water System Applications

Although this discussion deals primarily with circulating water systems that incorporate mechanical draft cooling towers in a closed cycle, the same design and evaluation technique is applicable to other cooling systems suggested in the introduction.

Generating units equipped with natural draft cooling towers would likely see more flow variations relative to turbine load to compensate for weather fluctuations and the lack of draft fans. These towers typically have a higher static pumping head requirement, but higher pump speeds do not result, since these towers typically are applied to larger generating units.

CONCLUSION

Circulating water systems are not in the limelight of generating station design, as demonstrated by the prevalent lack of operating flexibility. In comparison to currently popular trends in circulating water pumping schemes, significant efficiency and operational improvements are available without sacrificing reliability. To "design in" these improvements, a knowledge of the entire cooling process including potential operating characteristics and liabilities is required. This background, along with knowing the limitations of applicable pumps and available drives, is the controlling factor in evaluating the practicality of choosing more energy efficient pumping systems. Understanding these idiosyncracies is likely to result in more cost effective cooling system designs.

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