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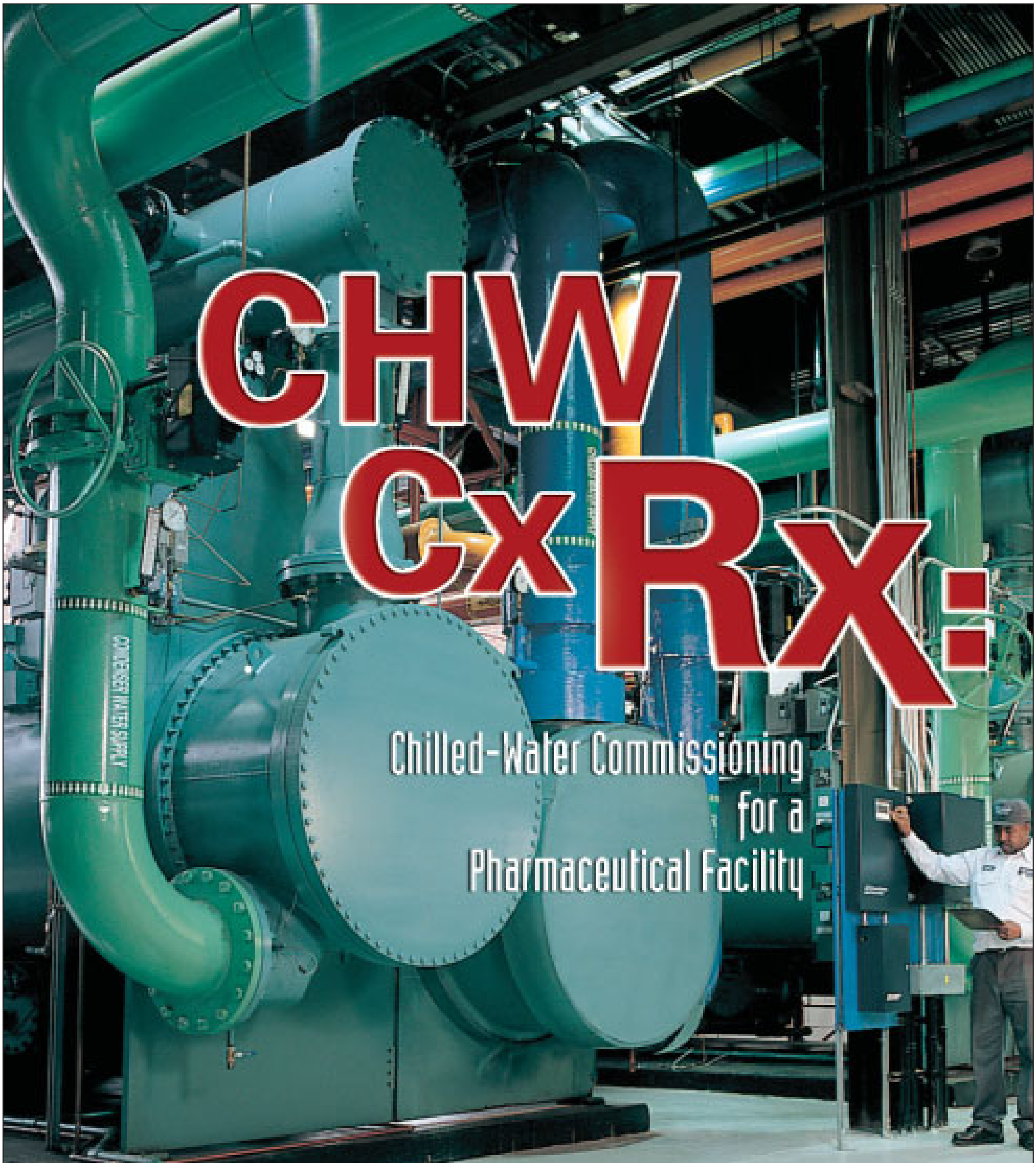

CHW CxRx:

Chilled-Water Commissioning for a Pharmaceutical Facility

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Commissioning tips for large chillers, pumps, and cooling towers at pharmaceutical plants and similar campus-type facilities.

Commissioning

Chilled Water Plants

On Large Campus Settings

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Often, the prospect of commissioning a large mechanical project can be daunting. We all know that daunting tasks have a tendency to be glossed over or given less attention than required. When pertaining to a chilled water plant, this oversight can be costly.

The purpose of this article is to cut through the fog associated with commissioning a chilled water plant on a large campus-type corporate setting.

A good place to start is with the term commissioning. There are many definitions of commissioning from very reputable professional organizations. However, it is wise to reread these definitions repeatedly in order to fully grasp their meaning as they are typically written in general terms intended for universal applicability. Let's simplify the definition for our purposes:

"Commissioning is the documented process to ensure that a system is working effectively, efficiently, and according to the client's expectations."

So, the three key components of commissioning are achieving an effective, efficient system that meets the client's expectations. In this case, effective means that the chillers and cooling towers are maintaining their leaving water temperature, the pumps are capable of maintaining proper flow rates, and the control system is managing the staging of components. Efficiency adds another level of value. This is judged not by a comparison to the minimum requirements but to the maximum potential. Has the chiller been selected and is it operating to maximize its peak or part-load performance efficiency (kilowatts per ton for electric chillers)? Has the pump been selected close to its best efficiency point and is it operating on its curve? Has the piping been installed to prevent needless pressure loss? Etc.

The final component and most important is the client's expectations. It is clearly possible to commission a system that is both effective and efficient, and yet miss the mark. Take a primary-only process chilled water plant. We know that staging of a primary-only plant results in periods when the system's

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Photo courtesy of Baltimore Air Coil

Multiple cooling tower cells for a central chilled-water plant.

supply water temperature will be elevated during the staging of chillers. This is due to the short-term bypass of "warm" return-water through a chiller whose refrigeration has not yet ramped-up. If the process cannot tolerate this sometimes 5-10 min. "jump" in chilled water leaving temperature, then the project has failed.

These aforementioned pitfalls can be avoided by a properly executed commissioning process. While it is true that they can also be avoided by properly executing the engineering and construction process, in reality, the commissioning agent is typically the last line of defense.

The basic commissioning process is as follows:

• Administrative phase: Plan the process and learn the client's expectations.

• Design phase: Review the engineering concepts and plans to ensure the system and its components are commissionable, maintainable, and meet the client's expectations.

• Construction phase: Ensure the system was installed properly, and is operating properly (start-up, testing and balancing).

• Acceptance phase: Ensure that the system in functioning according to the sequence of operations and its components are operating at peak efficiencies.

Certain experiences demonstrate the importance of the aforementioned commissioning process. Unfortunately, if these items are not discovered during the design phase of commissioning, then the only recourse is to inform the client of the bad news.

BALANCING THE PUMPS

We are often told that the final step in the process for balancing of a pump's flow should be to trim the impeller. However, the final step usually taken in the balancing of a pump's flow is the induction of pressure through the partial closing of the discharge balancing valve. Is this the right choice for your clients?

The answer, as always, relates to money—first-cost and operational. The evaluation of both together leads us towards an answer to this question. Which is better in constant speed applications: using the discharge balancing valve to throttle the flow, trimming the impeller, or providing the pump motor with a variable frequency drive?

Whenever the balancing valve is partially closed, it tells us that the actual pressure loss in the system is less than expected according to the engineering calculations. Since pumping power, and subsequently pumping energy, is proportional to its operating differential pressure, the pump will be operating at a higher power requirement than required.

By trimming the impeller, or reducing the speed of the pump (which has the same effect as trimming the impeller), the system can operate at the actual pressure required. Again, since pumping horsepower and energy are proportional to pump differential pressure, and assuming similar efficiencies (which is reasonable), the pump will consume less energy. However, while the operational energy will be less, it will cost more money upfront to trim the impeller or to install a VFD than to use the balancing-valve technique.

For clarity, let's look at a simplified example. Assume a constant-flow pump is required for a primary-only chilled water system that consists of 1,000 tons. The engineer has specified a horizontal split-case pump for 2,400 gpm at 180 ft. at 1,785 rpm. The vendor submittal presents a pump capable of the aforementioned design criteria with a 14.2-in. impeller, an operating efficiency of 85percent, and a brake horsepower of 130.

Once the system has been installed, and the balancing contractor has performed his work scope, we inspect the installation and perform our testing and balancing verification by "spot-check" measurements. All readings are within 10 percent of design flow and the pump test sheet is in order. However, there is a partially closed pump discharge balancing valve. Based upon review of the valve manufacturer's data, it appears as if the balancing valve is producing a pressure drop of 35 ft. Since a full-open balancing valve has a pressure drop of 5 ft, we know that the balancing valve is creating 30 ft of unnecessary pressure drop in the system.

Without trimming the impeller or slowing the speed down through the use of a VFD, this valve must remain in its position to maintain design flow. However, if this pump's impeller was trimmed to 13.2 in., the pump would now be delivering 2,400 gpm at 150 ft. The pump would then have a break horsepower of 108, which would result in a savings of $2,500 per year (assumes 1,500 operational hours, 10 cents per KWH levelized electrical rate and a 95 percent efficient motor). For these purposes, we will assume the savings will be similar for the VFD (which would allow the pump speed to be reduced to 1,650 rpm), by neglecting the small inefficiency of the VFD, and the reduced efficiency of a motor operating at reduced speeds.

Pump efficiency [h]	$ per ft: per 100 gpm	
	1 year	20 years
50.0%	$5.95	$119
52.5%	$5.67	$113
55.0%	$5.41	$108
57.5%	$5.18	$104
60.0%	$4.96	$99
62.5%	$4.76	$95
65.0%	$4.58	$92
67.5%	$4.41	$88
70.0%	$4.25	$85
72.5%	$4.11	$82
75.0%	$3.97	$79
77.5%	$3.84	$77
80.0%	$3.72	$74
82.5%	$3.61	$72
85.0%	$3.50	$70

TABLE 1. Operational costs for "wasted" pumping pressure.

These results are obviously significant-but are they significant enough?

Let's look at the first cost differential between the balancing method, trimming the impeller, and utilizing a VFD. Since the balancing method is the base method, and balancing would be required, regardless of which of the above methods were chosen, we will call its relative first-cost $0. Trimming the impeller will take 2 experienced technicians approximately 1 day (removal off-site, trimming, and reinstallation). At $125 per hr, plus some material, this results in an approximate cost to trim the impeller of $2,200. The decision to trim the impeller would have less than a 1 year simple payback ($2,200/$2,500 = 0.9 yrs). Utilizing a VFD will add start-up costs, the cost of an inverter duty motor, the cost of the VFD itself, and slightly more labor to install the VFD housing. If we assume the VFD will be stand-alone (therefore, not tied into the control system), we can exclude the cost of additional controls. In total, the cost of these items is approximately $20,000. At a simple payback of 8 years, this option is clearly less attractive. However, the client would still save $2,500 per year (in today's dollars), resulting in $30,000 over the life of the pump. Still a sound choice if the client can absorb the additional first-cost.

In appears clear that for this example, it does not make sense to use the balancing method, and between the impeller trimming and the VFD, you would recommend trimming the impeller.

We must recognize that this example is specific for a 2,400 gpm pump that was over-designed by 30 ft. By reviewing Table 1, one can see that other values of flow, efficiency, and pressure can be easily evaluated in a similar manner as the above example. Table 1 presents the accumulated savings, in today's dollars, over 1-year and 20-year periods, at varying efficiencies for every 1 ft of "saved" pressure and for every 100 gpm.

For example, with the same 2,400-gpm, 85-percent efficient (at design) pump, but with a balancing valve set to produce 5 ft of "wasted" pressure, go to the 85 percent mark on Table 1 and then locate the value of $70 in the third column, which is per 20 year, per 1 ft, per 100 gpm. Converting it to the given situation, multiply $70 x 5 ft x (2,400/100) = $8,400. In this case, trimming the impeller would still be justified, but using a VFD would not.

Note that this data can also be modified for a situation with more or less operational hours and with a different levelized electrical rate of 10 cents per KWH. With the same example, but for 2,000 hours and for a levelized electrical rate of 7 cents per KWH, multiply $8,400 x (2,000/1500) x ($0.07/$0.10) = $7,840.

In summary, the numbers don't lie. For an oversized system, it will almost always make sense to have the impeller trimmed once the initial balancing is completed. The appeal of this method also is that the decision can be made once you are sure that the system is oversized, whereas the VFD must be decided upon ahead of time, at which point the degree of oversizing is unknown and may not even exist at all once the system is commissioned.

PUMP CAVITATION

Cavitation is typically a concern with either hot-water applications or when the water basin or source is below the pump impeller's centerline. But, avoiding cavitation is not simply ensuring that the math proves that it will work. It includes a careful review of the actual pumping installation.

Cavitation is the boiling of water within a pump. When water boils, it creates water vapor "bubbles." When these bubbles travel through the eye of the impeller to the discharge side of the impeller, the increased pressure causes a violent implosion of the vapor back into a fluid. As you can imagine, this process will create a great deal of noise, cause reduced performance and will invariably cause erosion of material inside the pump. At times, the damage associated with cavitation can be quick, creating the need for pump replacement in a matter of hours.

The most appropriate method to predict if cavitation will occur is the net positive suction head (NPSH) method. By comparing the NPSH that is available (NPSHA) from the system to that which is required by the pump (NPSHR), one can determine if the installation has been properly engineered to reduce the possibility of cavitation. However, as useful as this tool may be, recognize that it is "man-made" and can only be considered as an approximation. In other words, a pump will cavitate, regardless of the NPSH, when the fluid's local pressure falls below the fluid's vapor pressure. While empirical data from the manufacturer show that this will occur when the NPSHA is less than the NPSHR, cavitation can still occur when the NPSHA is above the NPSHR.

Therefore, when engineering a pumping system, make sure that the NPSHA is greater than the NPSHR by a safe margin (I have seen estimates range from a 6 ft differential to double the NPSHR depending upon the manufacturer and the application), and that the pump and its suction line piping is installed per the manufacturer's installation guidelines. These will typically add rules of inlet straight piping runs, no horizontal elbows directly into the suction flange, flow straighteners, etc. Following this strategy will enhance the likelihood of success on your project.

EVAPORATOR AND CONDENSER PRESSURE LOSSES

Issues with the chiller are often at the forefront of everyone's mind because they consume the most energy in a chilled water plant, often accounting for more than 60 percent of the entire plant's energy usage.

As in all projects, the engineer will specify the chiller's performance requirements through equipment schedules, pre-purchase specifications, etc. The basis of design is often a specific chiller by a specific manufacturer. The determination of this selection is based upon such criteria as available utility (electric, steam, natural gas), full and part-load operational efficiency, turn-down, footprint, etc. Of course, the typical discussion point here is the chiller's operational efficiency, usually discussed in convenient terms of full-load kilowatts per ton (electric powered) or pounds per hour per ton (steam powered). However, an often overlooked piece of information in the evaluation of a chiller selection is the differential pressure across the evaporator and the condenser barrels.

Similar to the previous discussion of wasted pressure induced by balancing valves, unnecessary pressure adds money to the project in terms of operating energy. While one would think that a higher pressure-drop chiller would be a "cheaper" selection, this is not always the case. Therefore, at times, it doesn't even require a comparative evaluation between first-cost and operational costs as they are both lower than other selections. Then why is there "wasted" pressure?

The first scenario is when an initial selection has a higher than necessary pressure drop, meaning that an alternate selection exists with lower pressure drops that would still meet the other selection criteria. This can only be caught during the design phase of the project. For the commissioning agent, this can be caught during initial review of the drawings. In these cases, the cause is simply a less than ideal selection of numerous variables, such as tube diameter, number of passes, the absence or presence of internal rifling, etc. The only way to avoid this problem, which cannot be solved once the system is installed, is for the project team to obtain multiple selections and comparatively evaluate the selections.

The second scenario is that the chiller was selected properly, but is operating at a higher than predicted differential pressure once it is installed (assuming no factory testing was procured when it could be determined by the manufacturer). In this case, the first, and clearly the most important questions are whether the chilled water pumps and condenser water pumps have sufficient capacity to absorb this unexpected additional pressure. Assuming it is determined that the pumps have the spare capacity, the discussion typically ends, but should it? It depends.

Let's look again at a specific example. An engineer has selected a 1,650 ton chiller. On the evaporator side, the design flow rate is 3,940 gpm at a differential pressure of 27.3 ft, whereas the condenser flow rate is 4,615 gpm at 11.9 ft of differential pressure. When the commissioning agent is executing the capacity performance testing, it is determine that the actual differential pressures are 35 ft and 17 ft for the evaporator and condenser, respectively—a difference of 7.7 ft and 5.1 ft. By knowing that the chilled water and condenser water pumps are operating at 80 percent efficiency, we obtain a value of $74. Therefore, for the evaporator, over a 20-year period, this will result in additional expenditures of $74 x 7.7 ft x (3,940/100) gpm = $22,450. For the condenser, over the same period, this additional pressure will result in additional expenditures of $74 x 5.1 ft x (4,615/ 100) gpm = $17,420. In total, this results in almost $40,000 of approximate additional expenditures over the next 20 years. Perhaps enough evidence to obtain a reasonable credit on the chiller purchase. If in your application, the pump efficiency is worse, or the pressure discrepancy is worse, the problem may be more severe than the example presented. With any discrepancy of this nature, our value to the client is in presenting them with the information they need to make the appropriate decisions.

COOLING TOWER ENTERING AIR CONDITIONS

A more subtle problem can be found in the selection of the open evaporative cooler, commonly known as the cooling tower. In the selection of this major component, we know the most relevant data to the selection process is the entering air wet-bulb temperature, condenser water flow rate, and leaving and entering water temperatures (LWT, EWT). The latter information is typically determined in an iterative process with the chiller selections in order to determine the most proper selection for the plant. The former is more straight-forward. Typically, the entering air wet-bulb temperature is assumed to be the outside air peak wet-bulb temperature, which can be obtained from statistical weather data from one of many sources. However, depending upon the actual installation, this assumption can lead to insufficient performance.

While it is true that under most circumstances, the entering and ambient wet-bulb temperatures are the same, in some cases they are not. Where the installation can cause moist discharge air from the cooling tower to mix with entering (and drier) outside air, the entering wet-bulb temperature will be elevated above the actual outside air wet-bulb temperature.

This problem has two reasonable solutions. The first is to plan for reduced performance from the cooling tower during periods when the outside air wet-bulb temperature is at its highest. The reduced performance will cause a higher leaving water temperature, which will reduce the performance of the chillers as well. The other option is to account for this recirculation by recommending a selection that includes a higher entering wet-bulb temperature. This selection will typically have a higher first-cost. Both of these assumes that the problem is anticipated. But what if it is not? What is the effect on the chilled water plant?

A certain cooling tower has been selected to serve a nominal 2,000-ton chiller. This cooling tower is selected at 3,800 gpm with an 85 F LWT and a 100 F EWT. Assume that when the cooling tower is undergoing its performance test, the outside wet-bulb temperature is measured at design conditions (77 F in my geographical area), and the commissioning agent measures an entering wet-bulb temperature of 78 F, representing a 1F recirculation effect. In this case, assuming the chillers are fully loaded, this selection is now only capable of producing 85.7 F. Using age-old industry rules of thumb, this would result in an approximate 0.7 percent decrease in chiller performance (there will be approximately 1 percent decrease in performance per degree of condenser water temperature increase). At 0.55 KW per ton, the reduction would equate to 0.004 KW per ton or 8 KW during peak operation. Since this 77 F condition is only expected to be exceeded a small amount of the year (32 hours), the predicted energy "waste" would only be 256 KWH, equating to $25 per year. However, this does not include the fact that due to this recirculation the tower will continually need to work "harder" to maintain the design LWT at all outside air conditions. It does also not include that the chiller, if fully loaded, will begin to deliver a higher than designed chilled water supply temperature.

During your commissioning review of the preliminary engineering documents, pay special attention to installation issues that could cause this recirculation. Having the manufacturer review these drawings would also be a good solution. While we know that the manufacturer is typically involved heavily in the specification of the cooling tower, they are sometimes left out of the loop during installation. However, prior to this step, you can add value by reviewing the installation guidelines from the manufacturer, which will typically point out items such as the following:

• Ensure that if the towers must be installed near an adjacent wall or building that the discharge of the towers are at least as high as the adjacent wall or building.

• If barriers are required due to aesthetics (sometimes driven by local ordinances), ensure that the net free area of the enclosure louvers are sufficient for design airflow (typically design velocities across the louver's net free area should be no higher than 600 fpm).

• Ensure that the primary louver faces (intakes) are parallel to the prevailing wind associated with the highest outside air wet-bulb temperature.

As an added measure of protection, I also recommend having the tower selected with a small degree of conservatism by selecting the entering air wet-bulb temperature to be 1 F higher than the design outside air temperature.

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